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by E. N. Bamberger, E. V. Zaretsky, and H. Signer
Lewis Research Center
Cleveland, Ohio 44135

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by E. N. Bamberger,[1] E. V. Zaretsky,[2] and H. Signer[3]
NASA-Lewis Research Center

ABSTRACT

Groups of thirty 120-mm bore angular-contact ball bearings were endurance tested at a speed of 12 000 and 25 000 rpm ($1.44x10^6$ and $3.0x10^6$ DN) and a thrust load of 66 721 N (5000 lb). The bearings were manufactured from a single heat of VIM-VAR AISI M-50 steel. At $1.44x10^6$ and $3.0x10^6$ DN, 84 483 and 74 800 bearing test hours were accumulated, respectively. Test results were compared with similar bearings made from CVM AISI M-50 steel run under the same conditions. Bearing lives at speeds of $3x10^6$ DN with the VIM-VAR AISI M-50 steel were nearly equivalent to those obtained at lower speeds. A combined processing and material life factor of 44 was found for VIM-VAR AISI M-50 steel. Continuous running after a spall has occurred at $3.0x10^6$ DN can result in a destructive fracture of the bearing inner race.

INTRODUCTION

Rolling-element bearings for advanced technology aircraft engines for the 1980's are expected to operate at speeds to 3 million DN (DN is a bearing speed parameter

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[1]General Electric Co., Evendale, Ohio

[2]Member, ASME; NASA Lewis Research Center, Cleveland, Ohio.

[3]Industrial Tectonics, Inc., Compton, CA.

and is equal to the product of the bearing bore in millimeters and the shaft speed in rpm). Bearing temperatures for these advanced engines are expected to range from 492 to 589 K (425° to 600° F). Current production engine bearings operate at speeds less than 2.3 million DN and at temperatures generally less than 492 K (425° F). Because compressor or turbine blade tip speeds and disk burst strengths begin to limit the maximum speed of rotating components, a bearing speed of three million DN is equivalent to the practical limit of aircraft engine operation.

At high speed, the effect of centrifugal loading of the rolling elements against the outer race of the bearing becomes extremely important. Theoretical life calculations for a 150-mm bore angular-contact ball bearing operating at 3 million DN (20 000 rpm) predict that this bearing has approximately 20 percent AFBMA-calculated life [1]. This decrease in predicted life is due to the increased stress in the outer race caused by centrifugal effects. The expected final result is extremely short bearing life at speeds much above 2 million DN both in actual running time (hr) and in total bearing inner-race revolutions

Another problem associated with operating bearings at high speed is the need to adequately cool the bearing components because of excessive heat generation. A method which has been used successfully to 3 million DN is to apply cooling lubricant under the race [2]. In this method 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. As a result, both the cooling and lubricant function is accomplished.

A parametric study was performed to bearing speeds to 3 million DN with 120-mm bore angular-contact ball bearings under varying thrust loads, bearing and lubricant temperatures, and cooling and lubricant flow rates [3-5]. Contact angles were nominally 20° and 24°. The bearings were made from vacuum-induction melted, vacuum-arc remelted AISI M-50 steel. It was found that bearing operating temperature, differences in temperatures between the inner and outer races, and bearing power consumption can be tuned to any desirable operating requirement by varying 4 parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race and oil inlet temperature. These preliminary endurance tests at 3 million DN and 492 K (425° F) indicated that long term bearing operation could be achieved with a high degree of reliability.

It was found that 120-mm bore angular-contact ball bearings which were jet lubricated were limited to speeds less than 2.5 million DN [6]. Under-race lubrication produced, under all conditions of operation, lower bearing temperatures than jet lubrication with no apparent bearing speed limitation [7].

The research reported herein was undertaken to investigate the endurance and failure characteristics of main-shaft jet engine bearings at 3 million DN. The objectives were to (a) determine the life of under-race lubricated 120-mm bore angular-contact ball bearings made from vacuum-induction melted, vacuum-arc remelted AISI M-50 steel, (b) compare the bearing fatigue life at 3 million DN to that obtained at 1.44 million DN under identical conditions and (c) define the bearing failure mode at 3 million DN.

HIGH-SPEED BEARING TESTER

A schematic of the high-speed, high-temperature bearing tester used in these tests is shown in Fig. 1. This tester was initially described in [7, 8]. The tester consists of a shaft to which two test bearings are mounted. Loading is supplied through a system of ten springs which apply a thrust load to the bearings. A flat belt drives the test spindle from a 75 kW (100 hp) fixed speed electric motor. The drive motor is mounted to an adjustable base, so that drive pulleys for 12 000 to 25 000 rpm can be used with the same drive belts. The drive motor is controlled by a reduced voltage autotransformer starter which permits a selection of the motor acceleration rate during startup. This control protects the bearings efficiently from undesirably higher sudden acceleration during startup.

The lubrication system of the test rig delivers up to $2.8\text{x}10^{-2}$ cubic meter (7. 5 gal) per minute. There are three lubricant loops in the system. The oil flow in each loop is metered by adjustable flow control valves and can be individually measured by a flow rate indicator without interruption to the machine operation. Two of these loops are shown in Fig. 2. The first of these loops (C_o) supplies cooling oil to each bearing outer race. The second loop is divided by a lubricant manifold which feeds individual annular grooves or channels at the shaft internal diameter and proportions the amount of oil which is to lubricate and/or cool each bearing inner race. L_i designates the oil flow to the bearing through a plurality of radial holes in the center of the split inner race. C_i designates the lubricant used to cool the bearing inner race and lubricate the contact of the cage with the race land through radial holes in the inner-race shoulder. The lubricant system permits a selection of various lubricant schemes, including bearing lubrication through the inner-race split, lubrication of the cage-race shoulder contact region, the application of inner and/or outer-race cooling, and a selection of any desired flow ratio for cooling and lubrication. The third lubricant loop is fed into the slave bearing which supports the shaft (not shown in Figs. 1 and 2). By the system of valves and manifolds previously discussed an unlimited number of combinations of oil flows can be achieved to evaluate various conditions. Consequently, values of L_i, C_i, and C_o can be independent of each other.

In place of under-race lubrication and inner- and outer-race cooling, lubrication can also be provided to the test bearings through a recirculating jet feed lubrication system (not shown in Figs. 1 and 2). Two lubricant jets positioned 180^o apart were used for each bearing for the 12 000 rpm tests. Each lubricant jet had a single orifice having an orifice diameter of 2.23 mm (0.088 in.). The lubricant jets were positioned 11 mm (0.43 in.) from the bearing face with the lubricant stream aimed at the upper edge of the separator so that a fan pattern sprayed lubricant at the bearing between the separator and the inner and outer races.

The machine instrumentation includes protective circuits which shut down a test when a bearing failure occurs, or if any of the test parameters deviate from the programmed conditions. Measurements were made of bearing inner-race speed, bearing cage speed, test spindle excursion, oil flow, test bearing inner- and outer-race and lubricant temperatures, and machine vibration level. The speed and spindle measurements were made with proximity probes and displayed by digital read-out and oscilloscope, respectively. The oil flow was established by a flowmeter, and bearing outer-race lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded. The inner-race temperature of the front test bearing was measured with an infrared pyrometer.

TEST BEARING

The test bearings were ABEC-5 grade, split inner-race 120-mm bore ball bearings. These bearings were identical to those used in the investigations reported in [3-5]. The inner and outer races, as well as the balls, were manufactured from one heat of double vacuum-melted (vacuum-induction melted, consumable electrode vacuum remelted) AISI M-50 steel. The chemical analysis of the particular heat is shown in Table 1. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15 balls, 2.0638 cm (13/16 in.) in diameter. 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 gm-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°. 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 microin.) AA, and the inner and outer raceways were held to a 5 μcm (2 microin.) 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.

LUBRICANT

The oil used was a 5-centistoke neopentylpolyol (tetra) ester. This is a Type II oil, qualified to MIL-L-23699 as well as the oil specifications of most major aircraft-engine producers. The major properties of subject oil are presented in Table 2 and a temperature-viscosity curve is shown in Fig. 3.

TEST PROCEDURE

Prior to installation in the test rig, the test bearings were inspected and major dimensional characteristics such as radial and axial play and mounted contact angle were recorded. The shaft containing the two test bearings was then assembled into the test rig and the load was applied through the load springs. The load was also checked by a load gage incorporated into the loading system. Since the bearing oil supply was independent of the test rig drive supply, lubricant flow was initiated prior to turning over the test bearings. The bearings were then brought up to speed, using the reduced voltage auto-transformer which permitted a uniform acceleration (approx. 15 seconds from start-up to 25 000 rpm) for each test. Upon reaching the test speed, shaft run-out was checked using two proximity probes. If this run-out exceeded a pre-set limit, the test was stopped and corrective adjustments were made. Once test conditions were established, the oil/water heat exchangers were trimmed to achieve the desired test bearing temperature.

Test conditions were a speed of 12 000 or 25 000 rpm (1.44×10^6 or 3.0×10^6 DN) and a thrust load of 66 721 N (5000 lb). For the 12 000 rpm tests, the maximum Hertz stresses at the outer and inner races were 1731×10^6 and 2048×10^6 N/M^2 (251 and 297 KSI), respectively. At 25 000 rpm, the maximum Hertz stresses at the outer and inner races were 2096×10^6 and 1965×10^6 (304 and 285 KSI), respectively.

For the 25 000 rpm test condition, the bearings were lubricated through the inner race. Lubricant flow to the inner race (L_i) was 1.2×10^{-3} cubic meter per minute (0.313 gpm) and inner-race cooling flow (C_i) was approximately 3.6×10^{-3} cubic meter per minute (0.94 gpm). The cooling flow rate per bearing to the outer race (C_o) was 2.8×10^{-3} cubic meter per minute (0.75 gpm). The oil inlet temperature was approximately 425 K (305° F). The oil outlet temperature was approximately 478 K (400° F). The bearing temperature was 492 K $\pm$ 6 K (425° F $\pm$ 10° F).

The 12 000 rpm tests were lubricated by oil jet lubrication. These tests were conducted to provide a baseline for comparison with the 25 000 rpm tests and also to compare the life of these test bearings with that previously obtained with oil jet lubrication [10]. For the 12 000 rpm tests, the oil in temperature and flow rate were controlled to provide the same bearing temperature as the 25 000 rpm tests. The oil in temperature was approximately 458 K (365° F) and the oil out temperature was approximately 486 K (415° F). As with the 25 000 rpm tests, the bearing temperature was 492 K $\pm$ 6 K (425° F $\pm$ 10° F). The lubricant flow to each test bearing was 5.46×10^{-3} cubic meter per minute (1.4 gpm). The oil jet velocity at the exit was 1128 cm/sec (444 in./sec).

For both the 12 000 and 25 000 rpm tests, testing was terminated upon reaching 3000 hours or failure, whichever occurred first. Upon failure, the mating bearing was thoroughly inspected and if no debris or other damage was observed, it was returned to service in a subsequent test. The data were analyzed using the methods of [11].

Based upon previous experience [3, 10], approximately 1.9×10^{-3} cubic meter (0.5 gal) of oil was replenished in every 24 hours of operation. This amount of oil was removed from the sump and replaced with an equal amount of fresh oil. The rate of replenishment was approximately equal to 0.3 percent per hour of the entire sump capacity. This operation was performed without interruption of the test. By closely monitoring the oil replenished, it has been shown that no significant increases in either viscosity or acid number occur.

The rationale for this replenishment is that it serves to maintain the lubricant properties, specifically viscosity and acid number, at a relatively constant level. If no oil changes were made, the lubricant would present a continuously changing parameter with indeterminate effects on bearing performance. The selected replenishment rate is based on actual oil consumption rates in turbojet engines.

RESULTS AND DISCUSSION

Fatigue Life Results - Two groups of 120-mm bore angular-contact ball bearings made from vacuum-induction melted, vacuum-arc remelted (VIM-VAR) AISI M-50 steel 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 (1.44×10^{6} or 3.0×10^{6} DN) and a bearing thrust load of 66 721 N (5000 lb). The fatigue-life results of these tests are shown in Fig. 4 and are summarized in Table 3. At 1.44×10^{6} and 3.0×10^{6} DN, 84 483 and 74 800 bearing test hours were accumulated, respectively.

Only one bearing failed at 1.44×10^{6} DN (12 000 rpm). As a result, the bearing life distribution presented in Fig. 4 for this speed is based upon the results at 3.0×10^{6} DN (25 000 rpm) and is only an estimate.

For comparison pruposes, the experimental life results are compared to the predicted life obtained by the methods of Lundberg and Palmgren [12, 13] considering the centrifugal effects at 25 000 rpm but not elastohydrodynamic effects. The predicted life was 21×10^{6} inner-race revolutions at 3.0×10^{6} DN. The life distribution based upon the experimental life previously obtained in [10] adjusted for differences in contact angle, thrust load, and speed is also given in Fig. 4 for comparative purposes. This life based on [10] coincides with current design practice which would use a material factor of five multiplied by the predicted life to obtain a design life of 105×10^{6} inner-race revolutions.

The major difference, besides contact angle, between the bearings tested in the current study and those reported in [10] 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 used in [10]. However, using material factors for CVM AISI M-50 steel from the ASME design guide [14] and a high-speed bearing computer program which considers elastodydrodynamic effects based upon the analysis of Harris [15], an ASME predicted bearing life at a 90-percent probability of survival, L_A, was calculated. The ASME predicted bearing life distribution is shown in Fig. 4. The L_A life was 303 million inner-race revolutions. The ASME life prediction at 1.44×10^{6} DN (12 000 rpm) is approximately 1.2 times greater than the L_A life at 3.0×10^{6} DN or 363 million inner-race revolutions. An average life ratio of 7.6

between the experimental life results and the ASME prc cted lives was obtained from these data (see Table 3). This life ratio can be attributed to the use of the double-vacuum melted (VIM-VAR) AISI M-50 steel.

The ASME Design Guide [14] currently recommends a "Materials Factor D" of 2 for AISI M-50 and a "Processing Variable Factor-E" of 3 for consumable electrode vacuum meltir.g. The resultant life modifying factor would be 2×3 or 6. This is the value used for the ASME life predictions previously referred to. Based upon the experimental data presented herein the "Processing Variable Factor-E" for VIM-VAR processing would be 22. Hence, the life modifying factor would be 2×22 or 44 for VIM-VAR AISI M-50 steel.

<u>Speed Effects</u> - As previously discussed, theory [15-17] indicates that centrifugal forces would reduce bearing life as measured in inner-race revolutions as speed is increased. Omitting elastohydrodynamic effects, the theoretical reduction in bearing life between 1.44×10^6 DN and 3.0×10^6 DN is approximately 28 percent for the test conditions. Factoring elastohydrodynamic effects into the calculations, the theoretical reduction in bearing life is approximately 17 percent. The experimental life reduction is estimated to be approximately 9 percent. There is a good correlation between the theoretical analysis and the experimental results showing the effect of speed on bearing life.

Using the analysis of Harris [15] and a representative load-speed duty cycle shown in Table 4 which can be anticipated for the main shaft thrust bearing for an advanced turbojet engine, a bearing life in terms of engine operating hours for the specific bearing design used in this program was calculated. A combined processing and material life modifying factor of 44 was used for the VIM-VAR AISI M-50 steel. The resultant duty cycle life for the bearing would be approximately 7160 hours at a maximum bearing speed of 3.0×10^6 DN. For purposes of comparison, the bearing life was recalculated for a speed of 1.5×10^6 DN using a life modifying factor of 6 for CVM AISI M-50 obtained from the ASME Design Guide [14]. The duty cycle life would be approximately 8800 hours. In terms of absolute hours the life with the VIM-VAR material at 3.0×10^6 DN is 81 percent the life at 1.5×10^6 DN with the CVM material. Hence, with improved but currently available material technology and improved lubrication concepts it is possible to obtain bearing lives (in absolute hours) at speeds of 3×10^6 DN nearly equivalent to those obtained at current production engine bearing speeds.

If the VIM-VAR material processing factor were not applied to the bearings operating at 3.0×10^6 DN, the resultant life using the CVM material processing factor at this speed would be approximately 976 hours. Thus, the life reduction ratio in going from 1.5×10^6 to 3.0×10^6 DN would be approximately 9 to 1. Normally, if the life was solely a function of speed, the life ratio would be 2. However, because of centrifugal forces at the outer race at the higher speed, the outer-race stresses do not change significantly with applied bearing thrust load (see Table 4). Hence, the bearing life remains relatively low at the high speed conditions over the duty cycle.

Bearing Failure Analysis - For the 12 000 rpm (1.44×10^6) test, the single failure which occurred was on one ball. For the 25 000 rpm (3.0×10^6 DN) tests, of the six failures which were encountered, three failures occurred on the inner race and the remaining three were single ball failures.

Figure 5 shows typical ball and inner race fatigue spalls. Metallurgical analysis of the failed bearings established that the failures were initiated by classical subsurface rolling-element fatigue. In this failure mode, a small subsurface crack forms, normally associated with a stress riser such as a carbide [18]. The crack propagates radially outward from this point of initiation and upon reaching the surface forms a spall. Quite often this crack will bifurcate and propagate radially inward from the same point of origin, although such growth normally does not reach significant depths.

Good design practice for aircraft bearings dictates that bore/shaft fit-ups are so selected that inner-race mounting or hoop tensile stresses are limited to small values. Consequently, the potential driving force to continue rolling-element fatigue crack growth inward is essentially eliminated. However, at high bearing speeds such as 3.0×10^6 DN, considerable hoop tensile stresses can be introduced into the rotating inner races. Such stresses can be approximated by rather simple relationships to identify the threshold stress range for rapid fracture failure modes. Since aircraft engine bearing inner races tend to be rings with inner to outer diameter ratios of 1.07 to 1.10, it is fairly accurate to treat them as thin rings and to calculate an average tangential hoop stress. For the test bearings run at 3.0×10^6 DN the average hoop stress was calculated to be approximately 2.3×10^8 N/M^2 (34 000 psi).

Using the methods of [19, 20], a critical stress intensity factor K_{Ic} of 17.6×10^3 N M$^{-2}/\sqrt{M}$ (16 KSI/$\sqrt{IN}$) was calculated for the AISI M-50 material. This value is in good agreement with the experimental critical stress intensity factor of 12.7×10^3 N M$^{-2}/\sqrt{M}$ (14 KSI/$\sqrt{IN}$) for AISI 52100 steel [21].

When the bearing races are operated at low hoop stress values, incipient cracks formed in the raceway by rolling-element stressing do not propagate and/or the propagation rate is sufficiently slow that normal spalling type fatigue will occur. While spalling is undesirable, it is a relatively gradual failure process that can be detected by vibration monitors, chip detection devices and/or oil system monitors. Consequently, the affected bearing can usually be removed before serious secondary damage is incurred. However, if the hoop stresses are high, the onset of a spalling failure can cause the stress intensity factor within the raceway to reach its critical value resulting in a destructive fracture of the race [19].

To test the above hypothesis, two inner races from bearings suspended with no failure were prepared with an artificial defect, designed to initiate a normal spalling fatigue sequence. The defects were generated by electrochemically machining a slot into the center of the ball track. The rectangular slot which is illustrated in Fig. 6 was approximately 0.005 cm (0.002 in.) by 0.038 cm (0.015 in.) entering the raceway at a 30° angle and extending to a maximum depth of 0.015 cm (0.006 in.). The 0.038 cm (0.015 in.) dimension of the slot was in the axial direction and encompassed 10 percent of the ball track width. The calculated depth of the maximum subsurface shearing stress for this bearing under the test conditions was 0.025 cm (0.010 in.). Consequently, the deepest penetration of the slot was well above the maximum shearing stress. The sides and bottom of the slot were rounded by electrochemical polishing to reduce their effect as stress-raisers. The relative motion of the inner race to the ball is such as to stress the defect from A to B as shown in Fig. 6.

The first test with an artificial defect was run with all tester shutoff systems operative. The bearing ran for 2.75 hours at which time it was terminated by the vibration shutoff. Upon disassembly, the inner race was found to have a typical spalling failure approximately 3 inches long (see Fig. 7). The induced defect was approximately 0.953 cm (0.375 in.) behind the leading edge of the spall. This was as expected. It has been shown that while the general spall propagation is in the direction of A to B in Fig. 6, some spalling continues to take place in the opposite direction. Having established that the induced defect could initiate a normal spalling sequence, the second such bearing was installed in the same position on the tester. This time, the shutoff systems were disconnected. After 6 hours 17 minutes, vibration, temperature, and noise level indicated a fatigue spall failure. Approximately 7-1/2 minutes later an obviously severe failure occurred which terminated the test.

The failed bearing is shown in Fig. 8. Examination of the bearing revealed that the inner race had fractured into eight discrete segments. Reconstruction of the race showed that a fatigue spall approximately 8.9 cm (3-1/2 in.) long had occurred and that one of the fractures (presumed to be the initial one) had occurred toward the forward end of this spall. The mating unloaded inner race, balls and outer race showed some debris damage but otherwise were in good condition. The cage was intact except that it was plastically deformed (coned) on the side of the inner-race fracture. This indicates that the initial race fracture had been contained by the retainer and that the additional breakup of the race took place while the failed part was still in containment. It is possible that the initial fracture had occurred early in the 7-1/2 minute destructive cycle.

The results of this test make it obvious that race fracture at very high speeds can be a serious problem. Its solution must incorporate both fracture mechanics methodology as well as materials development, aimed at improving the fracture strength of the high-speed bearing steels.

SUMMARY

Groups of thirty 120-mm bore angular-contact ball bearings were endurance tested at a speed of 12 000 and 25 000 rpm (1.44×10^6 and 3.0×10^6 DN) and a thrust load of 66 721 N (5000 lb). The bearings were manufactured from a single heat of vacuum-induction melted, vacuum arc remelted (VIM-VAR) AISI M-50 steel. For the 12,000 rpm tests, the maximum Hertz stresses at the outer and inner races were 1731×10^6 and 2048×10^6 N/M^2 (251 and 297 KSI), respectively. At 25 000 rpm, the maximum Hertz stresses at the outer and inner races were 2096×10^6 and 1965×10^6 (304 and 285 KSI), respectively. The bearing temperature was 492 K (425° F). The bearings were lubricated by a tetraester lubricant meeting the MIL-L-23699 specification. At 1.44×10^6 and 3.0×10^6 DN, 84 483 and 74 800 bearing test hours were accumulated, respectively. Test results were compared with similar bearings made from consumable-electrode melted (CVM) AISI M-50 run under the same test conditions. The following results were obtained:

1. With improved but currently available material technology and improved lubrication concepts it is possible in terms of absolute hours to obtain bearing lives at speeds of 3.0×10^6 DN nearly equivalent to those currently being obtained at lower speeds.

2. A combined processing and material life modifying factor of 44 was found for VIM-VAR AISI M-50 steel as compared to a factor of 6 for CVM AISI M-50 steel.

3. Initial bearing failure at both 12 000 and 25 000 rpm was by classical subsurface rolling-element fatigue of either the inner race or a ball. However, continuous running after a spall has occurred at 3.0×10^6 DN can result in a destructive fracture of the bearing inner race because of hoop stresses present due to centrifugal forces.

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Table 1 Chemical analysis of vacuum induction, consumable-electrode vacuum remelted AISI M-50 bearing steel

Element	Composition, wt. % Races and balls
Carbon	0.83
Manganese	0.29
Phosphorus	0.007
Sulfur	0.005
Silicon	0.25
Chromium	4.11
Molybdenum	4.32
Vanadium	0.98
Iron	Balance

Table 2 Properties of tetraester lubricant

Additives	Antiwear, oxidation inhibitor, antifoam
Kinematic viscosity, cS, at —	
311 K (100 deg F)	28.5
372 K (210 deg F)	5.22
477 K (400 deg F)	1.31
Flash point, K (deg F)	533 (500)
Fire point, K (deg F)	Unknown
Autoignition temperature, K (deg F)	694 (800)
Pour point, K (deg F)	214 (−75)
Volatility (6.5 hr at 477 K (400 deg F)), wt. %	3.2
Specific heat at 477 K (400 deg F), J/(kg)(K) (Btu/lb)(deg F))	2340 (0.54)
Thermal conductivity at 477 K (400 deg F), J/(m)(sec)(K) (Btu/hr)(ft)(deg F))	0.13 (0.075)
Specific gravity at 477 K (400 deg F)	0.850

Table 3 Summary of endurance tests with 120-mm bore angular contact ball bearings [material, VIM-VAR AISI M-50 steel; temperature, 492 K (425° F); contact angle, 24°]

Speed, rpm (DN)	Thrust load, N (lb)	Maximum Hertz stress, N/M² (KSI)		Predicted life [12, 13], [a] rev. $\times 10^{-6}$ (hrs)		Experimental life, rev. $\times 10^{-6}$ (hrs)		Experimental Weibull slope	Failure index[c]
		Inner race	Outer race	L_{10}	L_{50}	L_{10}	L_{50}		
12 000 (1.44×10^6)	66 721 (5000)	2048×10^6 (297)	1731×10^6 (251)	29 (40)	157 (216)	[b]2700 (3750)	[b]6975 (9687)	[b]2.1	1 out of 30
25 000 (3.0×10^6)	66 721 (5000)	1965×10^6 (285)	2096×10^6 (304)	21 (14)	113 (76)	2400 (1600)	6200 (4133)	2.1	6 out of 30

[a]Centrifugal effects included.

[b]Estimated.

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

Table 4 Engine bearing duty cycle for 120-mm bore angular contact ball bearing

Speed, rpm (DN)	Percent of speed	Percent of time at speed and load	Thrust load at speed, N (lb)	Life at speed and load, hr	Maximum Hertz stress, N/M² (KSI)		Bearing life, [c] hr
					Inner	Outer	
[a]12 500 (1.5×10^6)	60	10 (Idle)	3114 (700)	183×10^3	1082×10^6 (157)	1013×10^6 (147)	8.8×10^3
[CVM AISI	100	20 (Take-off)	11 121 (2500)	3.5×10^3	1627×10^6 (236)	1489×10^6 (216)	
M-50]	100	70 (Cruise)	6672 (1500)	12.8×10^3	1365×10^6 (198)	1358×10^6 (197)	
[b]25 000 (3.0×10^6)	60	10 (Idle)	3114 (700)	203×10^3	1048×10^6 (152)	1331×10^6 (193)	7.2×10^3
[VIM-VAR AISI M-50]	100	20 (Take-off)	11 121 (2500)	4.3×10^3	1572×10^6 (228)	1944×10^6 (282)	
	100	70 (Cruise)	6672 (1500)	7.5×10^3	1331×10^6 (193)	1855×10^6 (269)	

[a]ASME life modifying factor for CVM AISI M-50, 6 [14].

[b]Advanced material VIM-VAR AISI M-50 life modifying factor, 44.

[c]Bearing life based on Miner's rule.

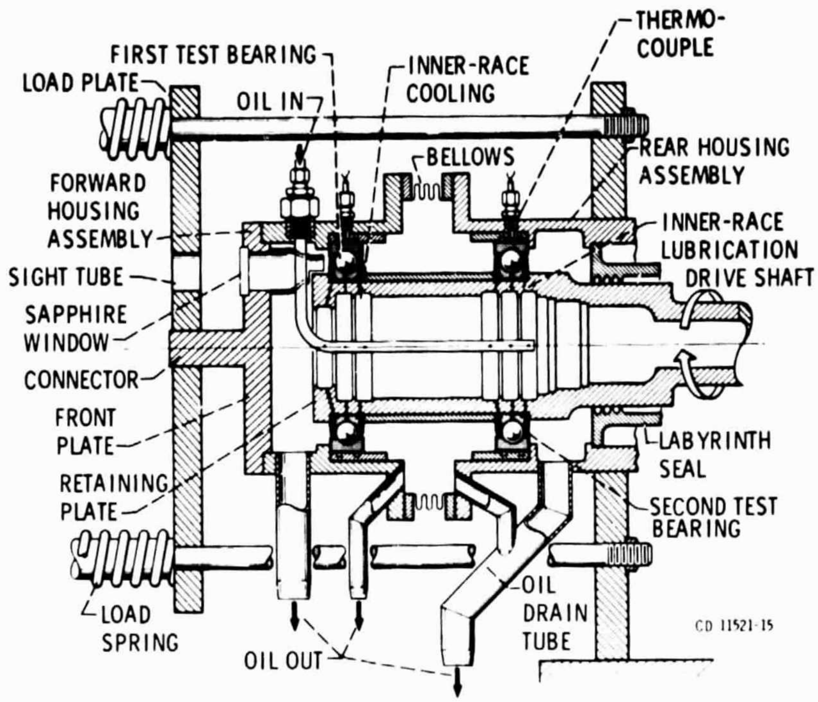


Figure 1. - High-speed, high-temperature bearing test apparatus.

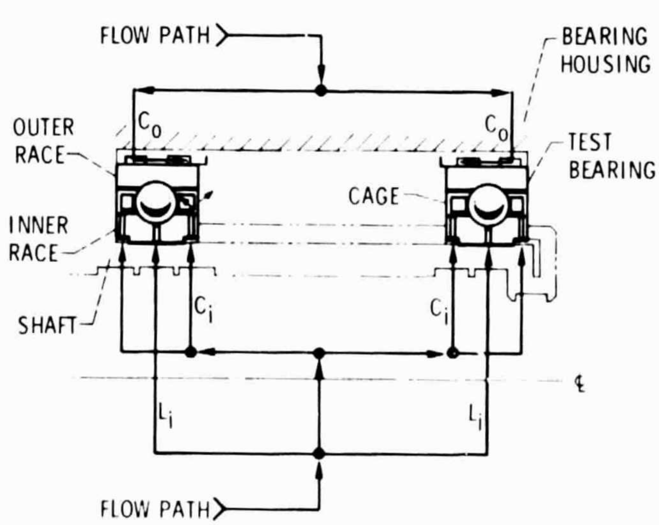


Figure 2. - Lubricant system for test bearings.

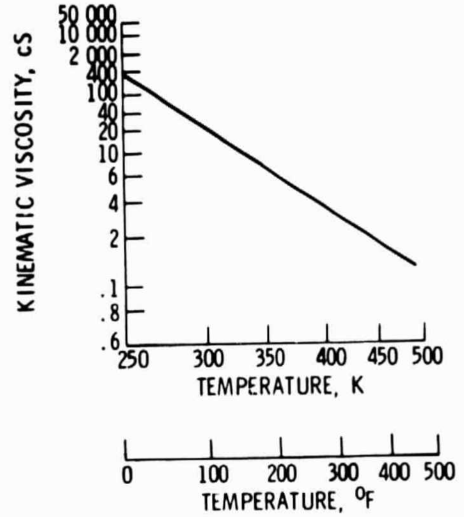


Figure 3. - Viscosity as function of temperature for tetraester (type II) lubricant.

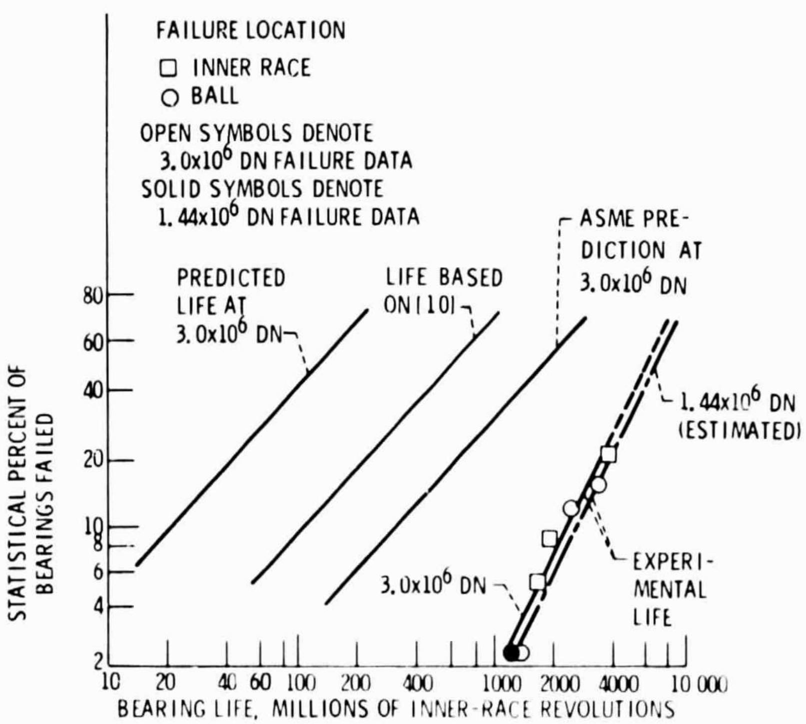
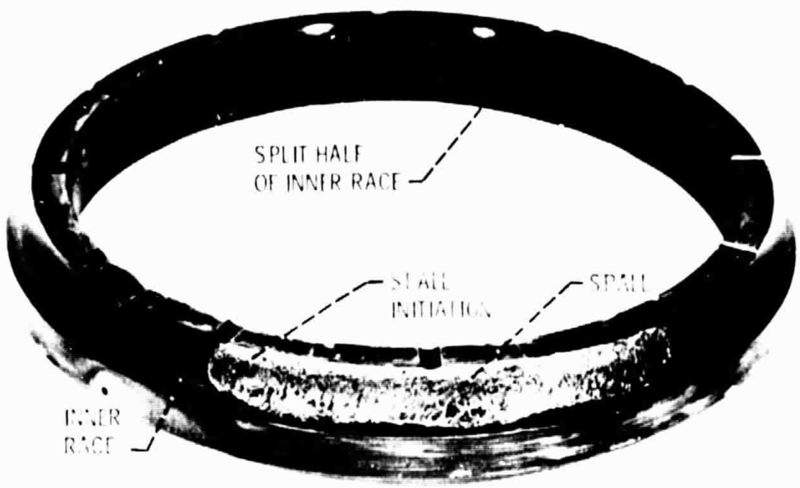
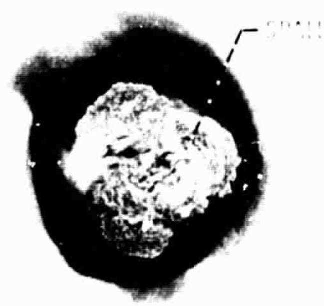


Figure 4. - Endurance characteristics of 120-mm bore angular-contact ball bearings. Thrust load, 22 241 N (5000 lb); temperature, 492 K (425^{o} F); material, AISI M-50 steel; lubricant, tetraester.



(a) Representative race failure. Time to failure, 1116 hours.



(b) Representative ball failure. Time to failure, 1674 hours.

Figure 5. - Representative rolling-element fatigue failures of ball and inner race of 120-mm bore ball bearing made of VIM-VAR AISI M-50 steel. Thrust load, 66 721 N (5000 lb) speed, 25 000 rpm; lubricant, tetraester.

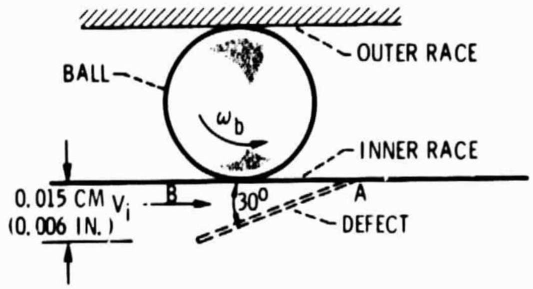


Figure 6. - Induced defect in bearing inner race.

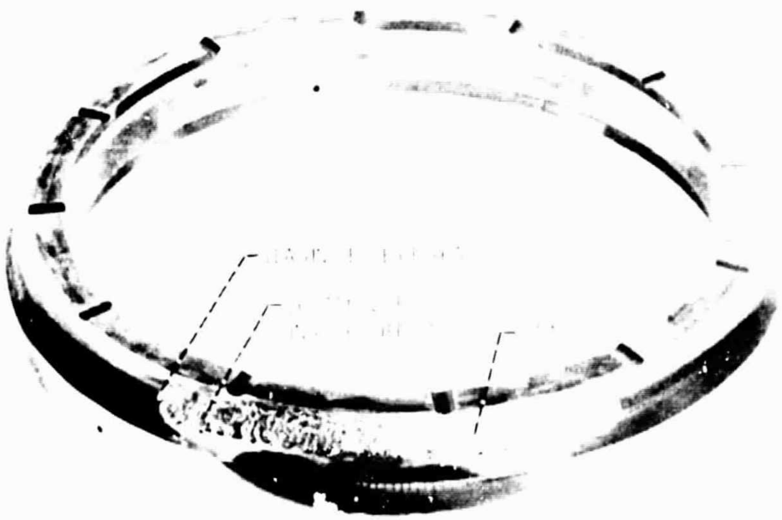


Figure 7. - Fatigue spall initiated from induced defect in inner race.

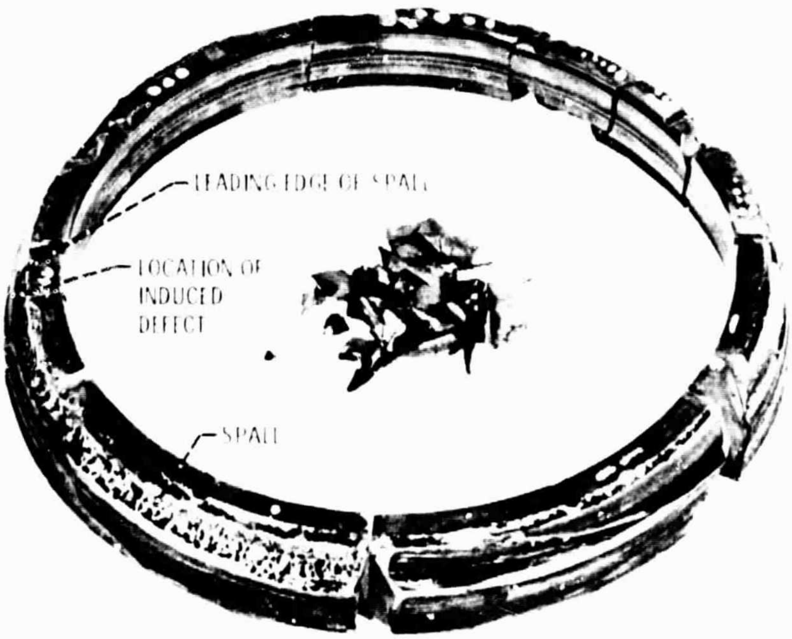


Figure 8. - Fractured bearing inner race caused by initiation of a rolling-element fatigue spall.