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Endurance Test with Large-Bore Tapered-Roller Bearings to 2.2 Million DN

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ABSTRACT

Endurance life tests were run with standard design and optimized high-speed design 120.65-mm-(4.750-in.-) bore tapered-roller bearings at shaft speeds of 12 500 and 18 500 rpm, respectively. Standard design bearings of vacuum melted AISI 4320 and CBS-1000M, and high-speed design bearings of CBS-1000M and through-hardened AISI M-50 were run under heavy combined radial and thrust load until fatigue failure or until a preset cutoff time of 1100 hours was reached. Standard design bearings made from CBS 1000M material ran to a 10-percent life approximately six times rated catalog life. Twelve identical bearings of AISI 4320 material ran to ten times 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 CBS 1000M high-speed design bearings ran to twenty-four times rated catalog life without any spalling, cracking or fracture failures.

INTRODUCTION

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. For this application, tapered-roller bearings have greater load capacity for a given envelope or for a given bearing weight than the more commonly used ball and cylindrical roller bearings.

Research directed toward high speed applications of tapered roller bearings has extended their high speed capabilities. Stable operation at speeds as high as 2.4 million DN have been attained under combined radial and thrust loads with computer-optimized, high-speed design tapered-roller bearings [1][1]. These bearings showed a 33 percent improvement in speed capability and a 16 percent decrease in heat generation over a modified baseline design bearing.

Under thrust-load-only conditions, for applications such as turbine engine main-shaft thrust bearings, speeds as high as 3.5 million DN have been attained [2]. Life tests with these specially designed bearings at 3 million DN yielded an estimated experimental life of approximately 3 times the unadjusted manufacturer's catalog life.

Advanced helicopter transmissions which require higher speed capability of tapered-roller bearings also require higher temperature capability [3]. Thus, materials with temperature capabilities higher than the conventional carburizing steels are required to withstand the higher operating temperatures at maximum conditions.

Several carburizing type steel have been developed for higher temperature use, primarily through the addition of alloying elements such as Cr and Mo. CBS-1000M, for example, has been developed for continuous service up to 589 K (600° F). Another candidate material for higher temperature transmission bearing use is AISI M-50, the through-hardened material used for nearly all turbine engine main shaft bearings.

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[1]Numbers in brackets denote reference at end of paper.

The objective of the research reported herein was to determine an experimental life estimate for 120.65-mm-(4.750-in.-) bore tapered roller bearings of two designs under combined radial and thrust loads. A modified standard bearing design [4] was life tested at 12 500 rpm. A computer-optimized, high-speed design was life tested at 18 500 rpm. Both designs were tested at a combined load of 26 700 N (6000 lb) radial load and 53 400 N (12 000 lb) thrust load.

APPARATUS AND PROCEDURE

High-Speed Tapered Roller Bearing Test Rig

Three test rigs of the type used in the performance tests of [1 and 4] were used for these life tests. Two bearings are tested simultaneously in each test rig, one of which is shown in figure 1. For these endurance tests, the cone-face temperature was measured with an infrared pyrometer aimed through an air-purged sight tube. The test rig vibration levels were measured with piezoelectric accelerometers which automatically shut down the test when vibration due to a bearing failure exceeded a predetermined level. Chip detectors were located in the oil drain lines from each test bearing for additional failure detection. The test rigs are described in detail in [4].

Test Bearings

Both designs of the tapered-roller test bearings had a bore of 120.65 mm (4.750 in.). Other significant geometry and dimensions are given in table 1.

The standard bearings were catalog design with modification of the roller ends which were ground to a spherical radius equal to 80 percent of the apex length. The cone was also modified with forty oil holes, 1.016 mm (0.040 in.) in diameter, drilled through from a manifold on the cone bore to the undercut at the cone-rib surface. The cage was AISI 1010 steel which was silver plated and of the standard roller riding design. Two groups of bearings of this design were used: one group was made from consumable-electrode, vacuum-melted (CVM) AISI 4320 steel, and the other from CVM CBS 1000M. Both were case carburized and finished to the specifications shown in table 2.

The AISI 4320 material is representative of the conventional carburizing steels with high industrial usage. CBS 1000M is a material alloyed for hardness retention to service temperatures up to 589 K (600° F). The particular heat of CBS 1000M used for both the standard and the high speed design bearings was the same as that used for the bearings tested in [2]. Chemical compositions of the materials used for cups, cones, and rollers are shown in table 3.

The selection of the computer optimized high-speed bearing design was based on bearing fatigue life, total heat generation, and cone-rib contact stress and heat generation at speeds up to 20 000 rpm as described in [1]. Major differences from the standard bearing design were smaller cup angle, smaller pitch and outside diameters, and fewer rollers. The cage of the high speed design bearing was made of silver-plated AISI 4340 and designed to be guided by lands on the cone. The cone contained 48 oil holes, 1.016 mm (0.040 in.) in diameter at each end, drilled through from manifolds on the cone bore to the undercuts at each raceway end. In addition, six oil holes of the same size were drilled at each end to lubricate the cage-land riding surfaces.

Two material groups of high-speed design bearings were used. One group was made of case-carburized CVM CBS 1000M and the second group from through-hardened, double vacuum melted AISI M-50. (Double vacuum melting refers to vacuum induction melting followed by vacuum arc remelting or VIM-VAR). The specifications on hardness, case depth and surface finishes for the CBS 1000M bearings are shown in table 2. The AISI M-50 bearings were through-hardened to Rockwell C 61 to 63 and were ground and lapped to the same finishes as the CBS 1000M bearings.

The basic dynamic load ratings and AFBMA dynamic load ratings of both bearing designs are given in table 4. The high-speed design has approximately 10 percent less capacity than the standard bearing due to its optimization of performance at higher speeds and the fewer number of rollers to accomodate the higher strength cage.

Life Testing Procedure

A set of two test bearings was examined visually on all contacting surfaces to assure that no defects which could lead to early failure were present. During life testing, the test bearing cup and support-bearing outer-ring temperatures, and the lubricant in and out temperatures were continuously recorded. The test bearing cone temperature, lubricant flow rates to test bearings and support bearings, spindle rotational speed, test rig vibration level, and load system pressures were monitored and recorded twice each day.

Testing continued until a failure was indicated or until a predetermined cutoff time of 1100 hours was reached. When one of the two bearings on test failed, the other was later mated with another suspended bearing and continued on test until failure or 1100 hour cutoff.

Lubrication of the standard bearing was accomplished with a combination of flow through holes in the cone directly to the cone-rib-roller-end contact and jet flow at the roller small end as shown in figure 2(a). For the high-speed design, all lubricant was supplied through holes in the cone at both ends of the roller and through the cage-riding lands as shown in figure 2(b). Flow rates, including cup cooling, were based on the results of parametric studies with each bearing design in [1 and 4].

The test conditions for these life tests are given in table 5. The test speeds were selected based on the results of the performance tests of [4] for the standard design bearings and [1] for the high-speed design bearings. The selected speeds were, in each case, less than the maximum speed capability with each design and at dynamically smooth operating speeds of the test rig.

During initial testing with the standard bearings, superficial surface peeling failures occurred rather than spalling fatigue failures. Several changes in lubricant temperatures and flow rates were made as will be discussed in a later section. The conditions listed in table 5 are those final conditions after the peeling failures were eliminated.

RESULTS and DISCUSSION

Standard Design Bearings

Life testing was initiated with standard design bearings of AISI 4320 material at 12 500 rpm, a thrust load of 53 400 N (12 000 lb) and a radial load of 26 700 N (6000 lb). Lubricant flow rates were a cone-rib flow rate of 0.0019 m^3/min. (0.5 gpm), a jet flow rate of 0.0038 m^3/min (1.0 gpm) and a cup cooling flow rate of 0.0026 m^3/min (0.7 gpm). The lubricant-in temperature was 366 K (200° F). These conditions were selected to maintain cup and cone temperatures less than 433 K (300° F) based on results of performance tests with identical bearings reported in [4]. Measured cup and cone temperatures ranged from 416 to 422 K (290° to 300° F) in these initial tests.

Peeling surface distress. - Inspection of the test bearings from the first two tests, which were stopped due to test facility malfunction and support bearing failure after 330 and 569 hours, revealed that a shallow surface distress was occurring on the raceway and roller taper surfaces. This type distress is called peeling [5] and appears as a very shallow area, uniform in depth. Typical peeled areas in these bearings, approximately 0.008 mm

(0.0003 in.) deep, are shown in figure 3. The peeling tended to initiate at minor surface defects such as the deeper surface scratches or indentations.

The peeling also tended to be concentrated near the axial center of the raceways and roller with a slight bias toward the roller large end. This effect may be expected since the rollers are slightly crowned and the contact stress is somewhat higher in the center of the raceway. Profile traces across the raceways and along the roller tapers of the initial test bearings revealed that the roller crown radius had been decreased to approximately one-third of its original value. This exaggerated crown caused further stress concentration at the center of the roller and further aggravated the peeling.

This exaggerated crown was also observed in similar tests reported in [3] and was attributed to an uneven transformation of retained austenite. The level of retained austenite in the case of the AISI 4320 bearings is approximately 30 percent. Austenite is a relatively unstable phase and transforms to martensite at a rate that depends on temperature and stress conditions. As it transforms, a growth of the material occurs. Since stresses and temperatures are higher near the center of the roller raceway contact, greater transformation, and thus growth occurs there, and it becomes a self aggravating condition. Some measurements of retained austenite on the rollers from the initial test bearings indicated that nearly all of the austenite had transformed near the center. It is believed that the growth from this transformation could account for the exaggerated crown that was measured. The cup and cone raceways also experienced some crown increase but to a lesser extent than the rollers.

The major cause of peeling type surface distress is believed to be due to an inadequate lubricant film parameter, which is the ratio of the elastohydrodynamic film thickness in the roller-raceway contact to the composite surface roughness. The composite surface roughness is the square root of the sum of the squares of the RMS roughnesses of the two surfaces.

Lubricant flow rates and lubricant-in temperature were varied to improve the EHD film thickness conditions. Cup and cone temperatures were reduced to less than 394 K (250° F), but the peeling failures continued to occur. Identical tests were also run with CBS 1000M bearings, and the results were the same.

Improved surface finish. - Further reduction of bearing temperatures to improve the lubricant film parameter were considered to be impractical since lower lubricant-in temperatures would not be representative of helicopter transmissions, and further increase in flow rate would not result in significantly lower bearing temperatures based on the work of [4]. The alternate means of increasing the film parameter is improving the surface roughness of the raceways and/or rollers. The remaining untested bearings were returned to the manufacturer, and the raceways and roller tapers were honed to improve the surfaces from the values given in table 2 to 0.10 μm (4 μin.) or better. After honing, the measured roughness of the raceways and roller tapers were typically 0.09 μm (3.5 μin.) and 0.06 μm (2.5 μin.), respectively. Also, the number of deeper surface scratches appeared to be minimized.

After honing, the bearings were reassembled, and testing was resumed at previous conditions selected to maintain bearing temperatures at 394 K (250° F) or less. A shutdown for test bearing inspection after 640 hours revealed no peeling or distress on any surface of the bearings. The life test conditions were thus established for the remaining tests with the standard bearings of both materials. A summary of the effect of the lubricant film parameter on peeling surface distress is shown in table 6.

Life tests at 12 500 rpm. - The remaining test bearings of both materials, after re-honing, were run at the test conditions shown in table 5. Twelve bearings of the AISI 4320 material ran to the 1100 hour cutoff without failure of any type. At these conditions, the rated catalog life of this bearing design is 102 hours, so that the experimental 10-percent life is greater than ten times the catalog life. (The 10-percent life is the life within which 10 percent of the bearings can be expected to fail by rolling-element fatigue spalling; this 10-percent life is equivalent to a 90-percent probability of survival.)

Sixteen re-honed bearings of the CBS 1000M material were run to spalling fatigue failure or to the 1100 hour cutoff. Twelve of the bearings ran to 1100 hours without failure. Three bearings experienced spalls on the cup or cone raceways. One bearing was suspended at 820 hours without spalling failure, since it had received surface damage on the cone-rib due to an obstruction in a lubricant orifice. These data are plotted on Weibull coordinates according to the procedures of [6] in figure 4. A least squares line drawn through the three failure points shows an estimated 10-percent life of approximately 600 hours, or about 6 times the rated catalog life.

A comparison of the results with the two materials show that the CBS 1000M bearing life is less than the AISI 4320 bearing life. However, a quantitative estimate of the difference is not made from these results since no fatigue failures occurred with the AISI 4320 bearings.

High-Speed Design Bearings

AISI M-50 bearing tests. - Life testing with the computer optimized, high-speed design tapered-roller bearings was initiated with the AISI M-50 bearings at 18 500 rpm. The externally applied load was identical to that for the standard design bearings, that is, a thrust load of 53 400 N (12 000 lb) and a radial load of 26 700 N (6000 lb). Lubricant flow was through the cone with 0.0055 m^3/min (1.45 gpm) at each end of the roller. Cup cooling flow of 0.0023 m^3/min (0.6 gpm). With the lubricant-in temperature of 355 K (180° F), the average cup and cone-face temperatures were 419 K (295° F) and 425 K (305° F), respectively. Under these load and speed conditions, the rated catalog life of this bearing design is 46 hours.

Five pairs of the AISI M-50 of bearings were run at these conditions. Two tests (four bearings) were stopped after 36 and 283 hours due to failures in the lubrication system. Another pair of bearings ran to the 1100 hour cutoff without failure. Another test was stopped after 106 hours due to a test rig malfunction, and upon inspection of the test bearings, a very small spall was found on one of the rollers.

After 188 hours, the other test was stopped due to severe rig vibrations. Disassembly revealed that both test bearings had cracked cones. One cone had cracked entirely through the cone section. The crack in the other cone was entirely contained in the load zone of the raceway.

The cone of bearing S/N 78-9 is shown in figure 5. Evidence of an axial crack is seen in the spalled area. The cone was subsequently cut partially through and fractured at that location to reveal the crack. The extent of the crack is shown in figure 6. The dimensions are approximately 9.5 mm (0.37 in.) long and 1 mm (0.04 in.) deep.

The cone of bearing S/N 78-10, with the complete fracture, is shown in figure 7. A portion of the initial crack and advanced stages of fracture are shown in figure 8. Initial crack dimensions are approximately 22 mm (0.87 in.) long and 6 mm (0.24 in.) deep.

Fatigue spalling is noted in figures 5 and 7 adjacent to the cracks. It is believed that the fatigue cracks, related initially to the spalling process, propagated at an accelerated rate due to the presence of the superimposed high hoop tensile stress field in the cones. In the case of bearing serial number 78-10, the crack propagated to its critical value and destructive fracture occurred. Fortunately, the test rig was shut down before complete fragmentation of the cone occurred. The crack in bearing S/N 78-9, although at an advanced stage, had not yet reached the critical fracture stage.

The average tangential tensile hoop stress, based on the assumption of thin rings, was estimated for the 18 500 rpm condition to be approximately 0.145 GPa (21 000 psi). This calculated stress also includes effects of the cone-shaft interference fit. This stress is at the lower end of the range where in [7 and 8], it was shown that the critical crack size can readily be reached in through-hardened AISI M-50, and fracture of inner races (or cones in the case of tapered roller bearings) is probable.

Bamberger [9] reported that fracture of ball bearing inner rings running at 3.0 million DN occurred where hoop stress due to high rotational speed was calculated to be 0.23 GPa (34 000 psi). Fracture occurred only after considerable fatigue spalling. In the present tapered-roller bearing tests, where calculated hoop stresses were somewhat lower, fracture occurred before significant spalling occurred. In the cases discussed by Clark [7], there was a greater tendency toward inner ring cracking without significant spalling with roller bearings than with ball bearings at similar stress levels. This tendency may be related to the relative spall propagation rates with ball and roller bearings. The results with tapered-roller bearings appear to be consistent with this trend.

These results indicate that the use of a through-hardened material such as AISI M-50, to accomodate high temperatures in high-speed tapered-roller bearing applications, involves the same, if not more, severe risks of catastrophic fracture of the cone as reported in [7 to 9] for ball bearing inner rings at high speeds. It is apparent that materials for high-speed tapered-roller bearings must have higher fracture toughness such as that associated with case-hardened materials.

<u>CBS 1000M bearing tests</u>. - Five computer optimized, high-speed design tapered roller bearings made of case-carburized CBS 1000M material were also life tested at conditions identical to those of the AISI M-50 bearings. Four of these bearings ran to the 1100 hour cutoff time without failure. One test resulted in a severe failure of one test bearing after 135 hours as a result of failure detection and shutdown system malfunction. The extent of damage was too great to determine the origin or cause of the failure. However, close observation revealed at least two spalls in the cone raceway. The largest spall is shown in figure 9. No cracks or fracture of the cone were observed although considerable damage to the raceway, cone rib and cage occurred. This result offers some, although admittedly meager, evidence that a high-temperature, case-carburized material such as CBS 1000M can resist fracture or cracking at conditions of high tensile hoop stress after spalling and significant cone raceway damage had occurred.

Too few bearings were run at these conditions, because of availability, to obtain a reasonable statistical estimate of the rolling-element fatigue life. However, since four of the five bearings survived the 1100 hour cutoff (24 times rated catalog life) this bearing design/material combination does show promise for future high-speed tapered-roller bearing applications.

SUMMARY

Endurance life tests were run with two designs of 120.65 mm (4.750-in.) bore tapered-roller bearings of three materials under heavy combined radial and thrust loads. Bearings of a modified standard design were life tested at

12 500 rpm. Computer-optimized design bearings were life tested at 18 500 rpm. Bearing temperatures were maintained in the range from 394 to 433 K (250° to 300° F).

The following results were obtained:

1. Standard design bearings of CBS 1000M material gave an estimated ten-percent life of about six times rated catalog life. Twelve identical bearings of AISI 4320 material ran to a cutoff of ten times rated catalog life without failure.

2. Under conditions of marginal lubricant-film-thickness-to-composite-surface-roughness ratio, peeling surface distress was a dominant mode of failure.

3. Reducing bearing temperatures to increase lubricant film thickness and re-honing the raceways and rollers to improve surface finish effectively eliminated the peeling mode of failure.

4. Cracking and fracture of the cones of AISI M-50 bearings occurred due to the high tensile hoop stresses at 18 500 rpm with the optimized high-speed design bearings.

5. CBS 1000M bearings of the optimized high-speed design did not crack or fracture. Four of these bearings ran to 24 times rated catalog life without failure.

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TABLE I. - TEST BEARING GEOMETRY

Dimension	Standard design	Computer optimized design
Cup half angle	17°	53'
Roller half angle	1°35'	1°35'
Roller large end diameter, mm (in.)	18.29 (0.720)	18.29 (0.720)
No. of rollers	25	23
Total roller length, mm (in.)	34.17 (1.3452)	34.18 (1.3456)
Pitch diameter, mm (in.)	166.8 (6.569)	155.1 (6.105)
Bearing outside diameter, mm (in.)	206.4 (8.125)	190.5 (7.500)
Roller crown radius, mm (in.)	25.4×10^3 (1000)	25.4×10^3 (1000)
Roller spherical end radius, percent of apex length	80	80

TABLE 2. - TEST BEARING SPECIFICATIONS

	Standard design[a]	Computer optimized design (CBS 1000M)[b]
Case hardness, Rockwell C	58 to 64	58 to 64
Core hardness, Rockwell C	25 to 48	25 to 48
Case depth (to 0.5 percent carbon level after final grind), cm (in.):		
Cup and cone	0.086 to 0.185 (0.034 to 0.073)	0.061 to 0.185 (0.024 to 0.073)
Roller	0.091 to 0.201 (0.036 to 0.079)	0.091 to 0.201 (0.036 to 0.079)
Surface finish,[c] μm (μin.), rms:		
Cone raceway	0.15 (6)	0.10 (4)
Cup raceway	.20 (8)	.10 (4)
Cone rib	.18 (7)	.15 (6)
Roller taper	.13 (5)	.05 (2)
Roller spherical	.15 (6)	.08 (3)

[a]Identical specifications for both AISI 4320 and CBS 1000M bearings.
[b]AISI M-50 bearings through-hardened to Rockwell C 61 to 63 and finished to the same specifications as the CBS 1000M bearings.
[c]Measured values.

TABLE 3. - CHEMICAL COMPOSITION OF THE TEST BEARING MATERIALS

	Alloying element, weight percent (bal. Fe)						
	C	Mn	Si	Cr	Ni	Mo	V
AISI 4320	0.17/0.23	0.45/0.65	0.20/0.35	0.40/0.60	1.65/2.00	0.20/0 30	---------
CBS 1000M	0.12/0.16	0.40/0.60	0.40/0.60	0.90/1.20	2.75/3.25	4.75/5.25	0.25/0.50
AISI M-50	0.80/0.85	0.15/0.35	0.10/0.25	4.00/4.25	0.10 max.	4.00/4.50	0.90/1.10

TABLE 4. - LOAD RATINGS AND CAPACITY OF THE TEST BEARINGS

	Standard design	Computer-optimized design
Basic dynamic radial load rating[a], N (lb)	74 700 (16 800)	70 700 (15 900)
Basic dynamic thrust load rating[a], N (lb)	58 700 (13 200)	51 600 (11 600)
AFBMA[b] basic dynamic capacity, N (lb)	288 000 (64 800)	255 000 (57 400)

[a]The load which gives a 10-percent life of 90 million cone revolutions.
[b]Antifriction Bearing Manufacturers Association.

TABLE 5. - LIFE TEST CONDITIONS

	Standard design	Computer-optimized design
Spindle speed, rpm	12 500	18 500
Thrust load, N (lb)	53 400 (12 000)	53 400 (12 000)
Radial load, N (lb)	26 700 (6 000)	26 700 (6 000)
Lubricant flow rate, m^3/min (gpm):		
Jet at small end	0.0076 (2.0)	0
Through cone, small end	0	0.0055 (1.45)
Through cone, large end	.0038 (1.0)	.0055 (1.45)
Cup cooling flow rate, m^3/min (gpm)	0.0019 (0.5)	0.0023 (0.6)
Lubricant-in temp., K (°F)	350 (170)	355 (180)
Avg. cup temp., K (°F)	391 (245)	419 (295)
Avg. cone face temp., K (°F)	391 (245)	425 (305)

TABLE 6. - LUBRICANT FILM PARAMETER EFFECT ON PEELING ON

TAPERED-ROLLER BEARING RACEWAY

Bearing temp., K (°F)	Surface roughness, μm (μin.)			Calculated film thickness, μm (μin.)	Film parameter	Result
	Raceway	Roller	Composite			
416 (290)	0.15 (6)	0.13 (5)	0.20 (7.8)	0.28 (11)	1.4	Peeling
391 (245)	0.15 (6)	0.13 (5)	0.20 (7.8)	0.38 (15)	1.9	Peeling
391 (245)	0.09 (3.5)	0.06 (2.5)	0.11 (4.3)	0.38 (15)	3.5	No peeling

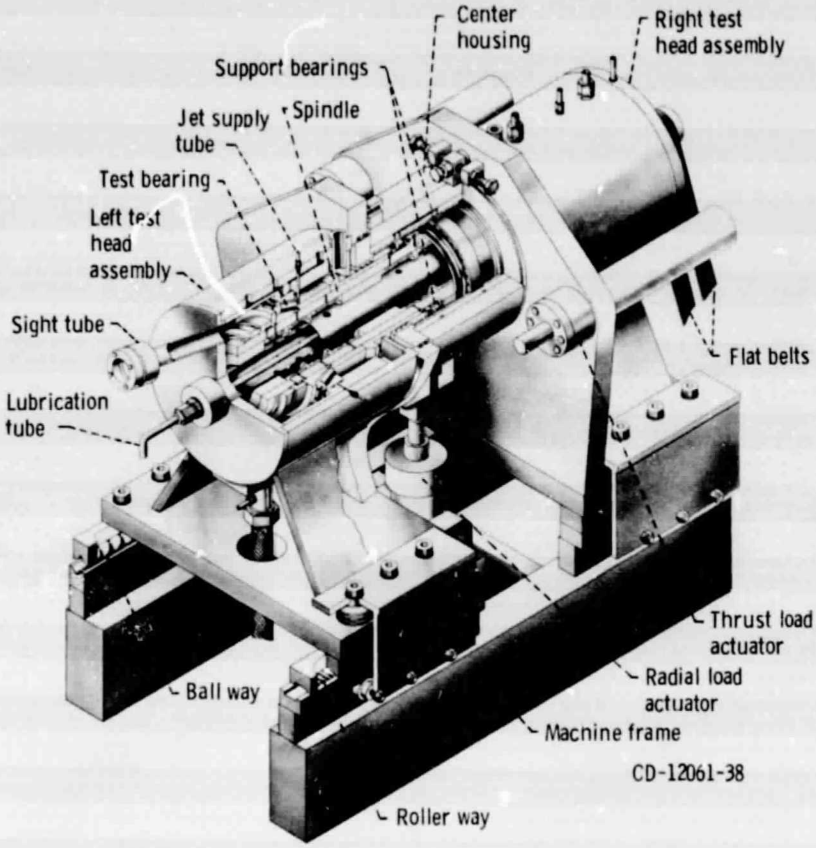
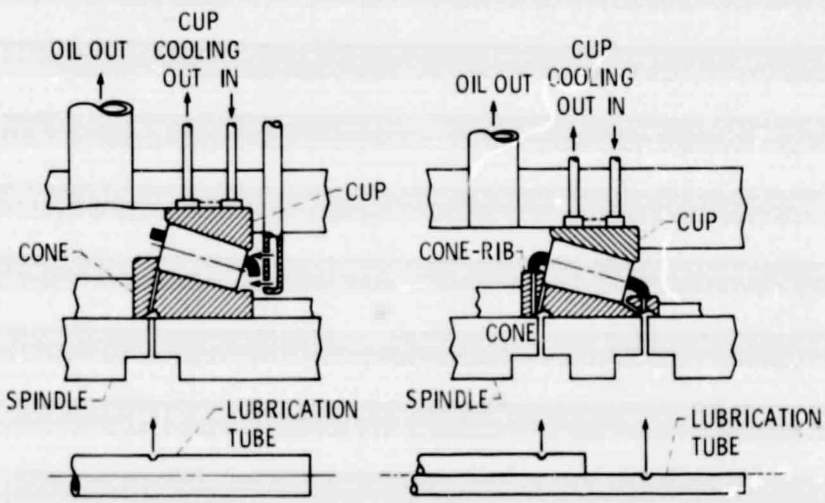


Figure 1. - Pictorial view of high-speed tapered roller bearing test rig.



(a) Standard design bearing. (b) Computer optimized design bearing.

Figure 2. - Lubrication and cooling of test bearings.

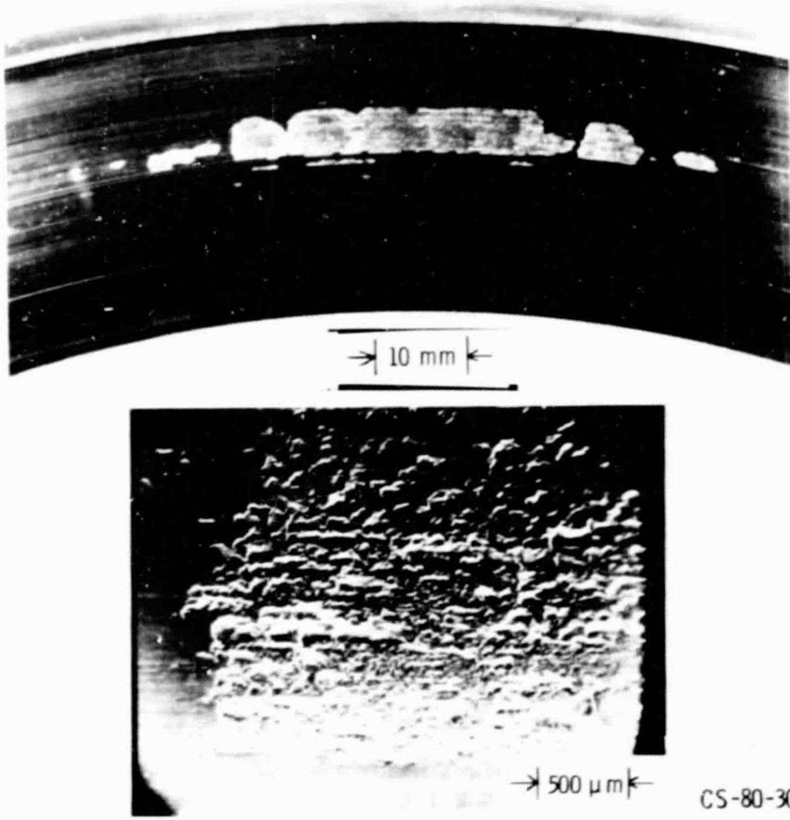


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Figure 3. - Peeling failure on cup raceway surface after 569 hours with standard design bearing run at 12 500 rpm.

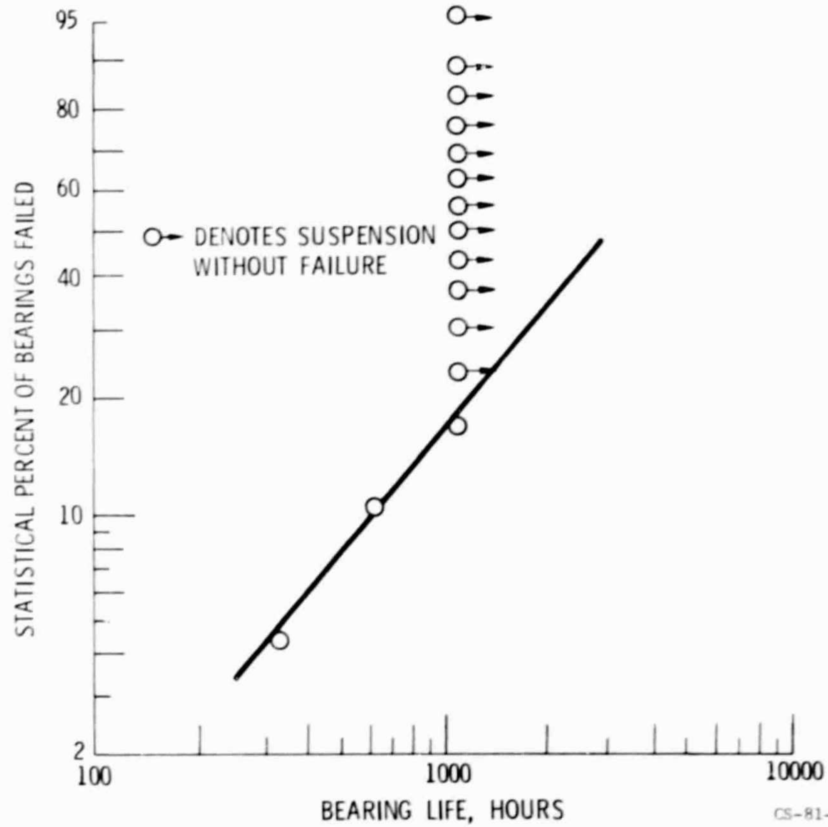

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Figure 4. - Rolling-element fatigue life of standard design CBS 1000M tapered-roller bearings at 12 500 rpm. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); bearing temperature, 394 K (250° F).

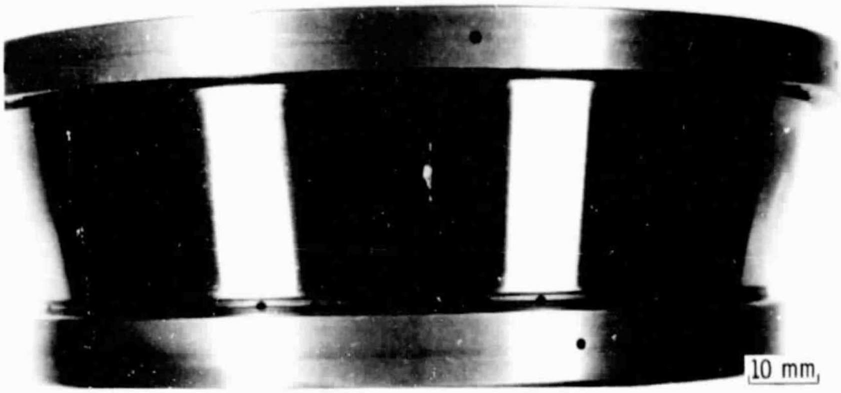

Figure 5. - Cone of AISI M-50 bearing S/N 78-9 showing raceway crack after 188 hours of operation at 18 500 rpm.

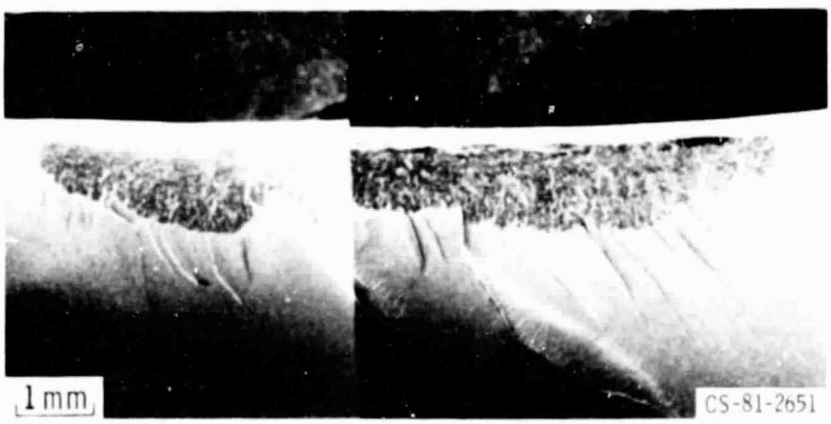

Figure 6. - Photomicrograph of cone raceway crack on AISI M-50 bearing S/N 78-9 after 188 hours of operation at 18 500 rpm.

CS-81-2650

Figure 7. - Fractured cone of AISI M-50 bearing S/N 78-10 after 188 hours of operation at 18 500 rpm.

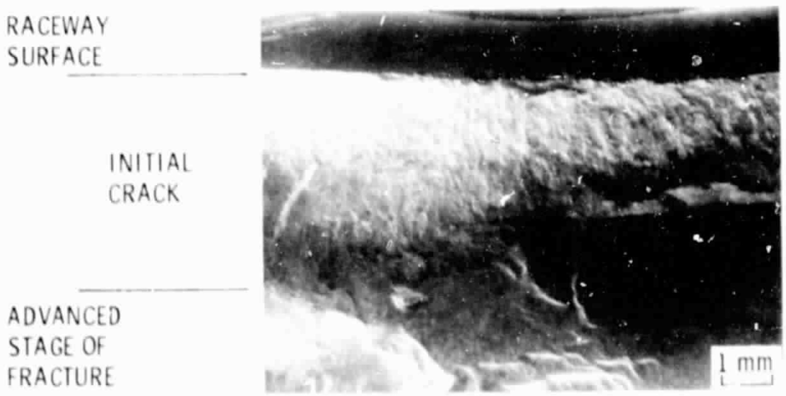

Figure 8. - Photomicrograph of cone raceway crack on AISI M-50 bearing S/N 78-10 after 188 hours of operation at 18 500 rpm.

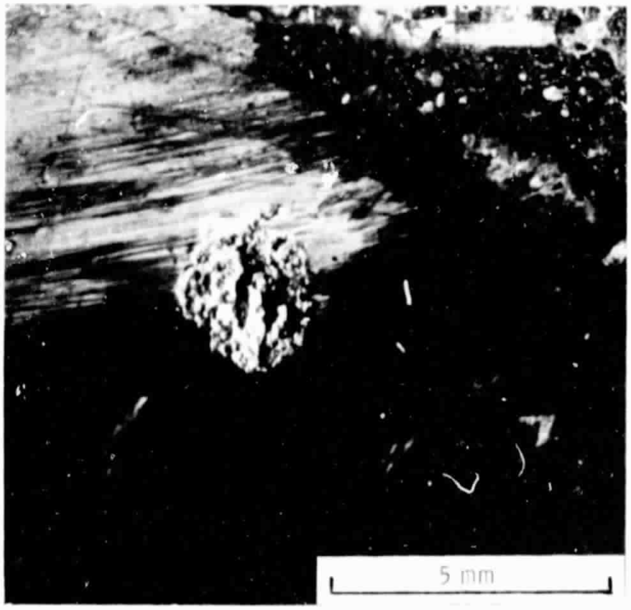

Figure 9. - Fatigue spall on cone raceway of CBS-1000M bearing S/N 76-17 after 135 hours of operation at 18 500 rpm.