Large-Bore Tapered-Roller Bearing Performance and Endurance to 2.4 Million DN

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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. Tapered-roller bearings have greater load capacity for a given envelope or for a given bearing weight than the combinations of ball and cylindrical roller bearings commonly used in this application. Speed limitations have restricted the use of tapered-roller bearings to lower speed applications than ball and cylindrical roller bearings. The speed of tapered-roller bearings is limited to approximately 0.5 million DN (a cone-rib tangential velocity of approximately 36 m/sec (7000 ft/min) unless special attention is given to lubricating and designing the cone-rib/roller-large-end contact. At higher speeds, centrifugal effects starve this critical contact of lubricant.

Several means of supplying lubricant directly to this cone-rib contact were investigated at higher speeds (ref. 1). Results of that work indicate that the most successful means was to supply lubricant to the cone-rib contact through holes from the bore of the cone. Additionally, the radius of the spherical, large end of the roller was optimized at 75 to 80 percent of the apex length. Development of a large, high-speed tapered-roller bearing for a heavy-lift helicopter transmission was reported in reference 2.

The feasibility of tapered-roller bearings for the high-speed and nearly pure thrust-load conditions of turbine engine main-shaft bearings has been reported for large and small bores in references 3 and 4. Under these thrust-load-only conditions, speeds as high as 3.5 million DN have been attained (ref. 5). Life tests with these specially designed bearings at 3 million DN yielded an estimated experimental life of approximately three times the unadjusted manufacturer's catalog life.

The use of computer programs can increase the capability of designing and analyzing tapered-roller bearings for such high-speed applications. These programs (described in refs. 6 and 7) take into account the difficulty of lubricating the contacts in high-speed tapered-roller bearings, and consider the effects of the elastohydrodynamic (EHD) films in these contacts. The effects of the EHD films in tapered-roller bearing contacts are discussed in reference 8. Experimental data at higher speeds are needed to verify the predictions of these computer programs.

Advanced helicopter transmissions which require the higher-speed capability of tapered-roller bearings also require higher temperature capability (ref. 2). Thus, materials with temperature capabilities higher than the conventional carburizing steels are required.

The objective of the research reported herein was to determine the operating characteristics and experimental life estimates 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 was tested at speeds up to 15 000 rpm. A computer-optimized, high-speed design was tested at speeds up to 20 000 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

The mechanical arrangement of the test rig is shown in figure 1 where one of the two test bearings is shown in a cutaway view. Thrust and radial loads are applied with hydraulic actuators. A

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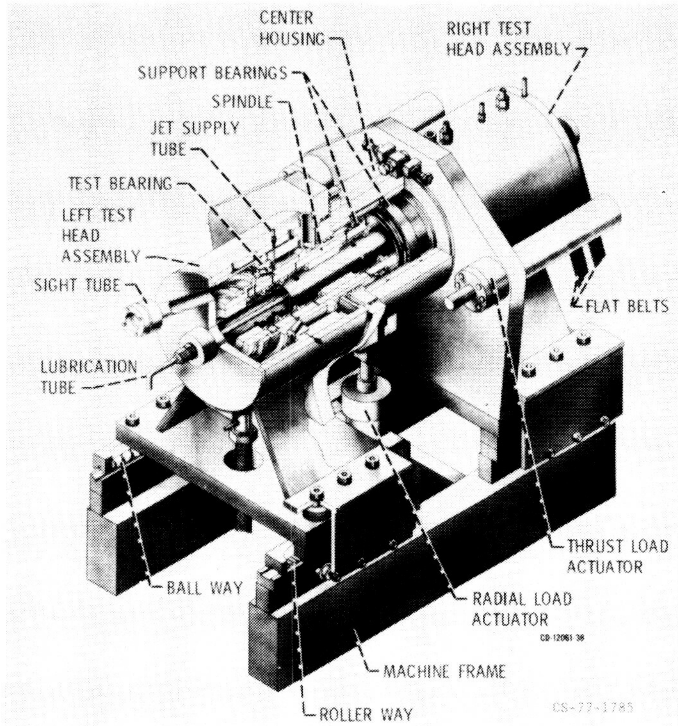

Figure 1. - Pictorial view of high-speed tapered roller bearing test rig.

flat-belt pulley system is used to drive the test spindle from a 93-kW (125-hp), 3600-rpm, 460-V, three-phase electric motor. Test spindle speeds are chosen by exchanging drive pulleys on the motor. The pair of flat belts is guided by an idler pulley arrangement which maintains controlled preload on the slack side of the belts.

The hollow spindle contains annular grooves on the inside diameter to distribute lubricant to radial holes for lubrication of the test bearings and the load bearings. A stationary lubrication tube delivers the desired lubricant flow to the annular grooves. For jet lubrication of the test bearings, two supply tubes are located 180° apart at the roller small end of each test bearing. The lubrication system, including pumps, filters, flowmeters, and load actuators, is described in detail in reference 9.

Thermocouples are installed for temperature measurements of each test bearing cup outer surface, each cylindrical load bearing outer ring, and oil inlet and outlet of both test and load bearings. Temperatures of the cone bore and cone face (fig. 2) on the drive end of the test spindle were measured with thermocouples and transmitted with an FM telemetry system. For the endurance tests, the cone-face temperature was measured with an infrared pyrometer aimed through an air-purged sight tube assembly.

The test rig vibration level is measured with piezoelectric accelerometers which automatically shut down the rig when vibration due to bearing failure exceeds a predetermined level. Chip detectors were located in the oil drain line from each test bearing for additional failure detection. Proximity probes measure shaft excursion in two planes as well as shaft speed and test bearing separator speed. A power meter was incorporated to monitor test rig power requirements. Preset safety flow switches and oil level switches were used to shut down the test machine in the event of lubrication system malfunction.

Test Bearings

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

The standard bearings were of catalog design with modified roller ends, which were ground to a spherical radius equal to 80 percent of the apex length. The cone was also modified with 40 oil holes, 1.02 mm (0.040 in.) in diameter, drilled through from a manifold on the cone bore to the undercut at

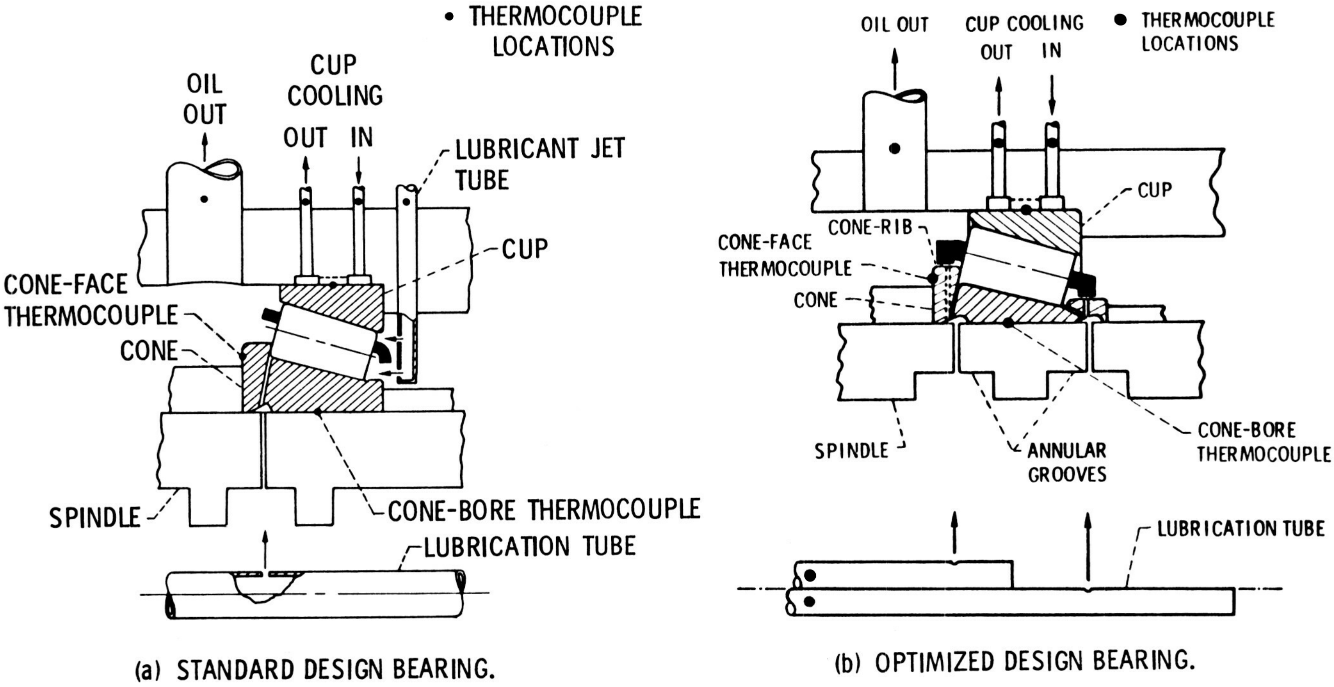

Figure 2. - Test bearing lubrication, cooling, and thermocouple locations.

TABLE I. - TEST BEARING GEOMETRY

	Standard design	Computer-optimized design
Cup half angle	17°	15°53'
Roller half angle	1°35'	1°35'
Roller large end diameter, mm (in.)	18.29 (0.720)	18.29 (0.720)
Number 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

the large end of the cone. The cage was silver-plated AISI 1010 steel, also 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 specification shown in table II. The AISI 4320 material is representative of the conventional carburizing steels with high industrial usage. CBS 1000M is a material alloyed for hardness retention at temperatures to 589 K (600° F) for continuous service. Chemical compositions of the materials used for cups, cones, and rollers are shown in table III.

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 to 20 000 rpm (ref. 10). 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 was designed to be guided by lands on the cone. The cone contained 48 oil holes, 1.02 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 other from through-hardened, double vacuum melted AISI

TABLE II. - TEST BEARING SPECIFICATION

	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	0.20 (8)	0.10 (4)
Cone rib	0.18 (7)	0.15 (6)
Roller taper	0.13 (5)	0.05 (2)
Roller spherical	0.15 (6)	0.08 (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 III. - CHEMICAL COMPOSITION OF THE TEST BEARING MATERIALS

	Alloying element, wt. %, (balance 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	.12/0.16	.40/0.60	.40/0.60	.90/1.20	2.75/3.25	4.75/5.25	0.25/0.50
AISI M-50	.80/0.85	.15/0.35	.10/0.25	4.00/4.25	.10 max.	4.00/4.50	.90/1.10

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 II. 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.

Procedure

Performance tests. – The test procedure was adjusted according to the test conditions to be evaluated. Generally, a program cycle was defined that would allow the evaluation of a number of conditions without a major interruption. Test parameters such as load, speed, and oil inlet temperature were held constant while the tester was in operation. Lubricant flow rates were adjusted during operation. The test bearings were allowed to reach an equilibrium condition before data were recorded and the next test condition was sought.

Lubrication of the standard bearing was accomplished with a combination of cone-rib flow, through holes in the cone directly to the cone-rib roller-end contact, and jet flow at the roller small end (fig. 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 (fig. 2(b)).

Life testing. – During life testing, the following data were recorded twice per day: test bearing cup and cone temperatures, support-bearing outer-ring temperature, lubricant in and out temperatures, lubricant flow rates to test bearings and support bearings, spindle rotational speed, spindle excursion, test rig vibration level, and load system pressures. Tests were run continuously until a failure was indicated or until a predetermined cutoff time of 1100 hr 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 the 1100-hr cutoff.

The test conditions for these life tests are given in table IV. 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 IV are those used in subsequent tests, where the peeling failures no longer occurred.

TABLE IV. - 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 (6000)	26 700 (6000)
Lubricant flow rate, m^3/min (gal/min):		
Jet at small end	0.0076 (2.0)	0
Through cone, small end	0	0.0055 (1.45)
Through cone, large end	0.0038 (1.0)	0.0055 (1.45)
Cup cooling flow rate, m^3/min (gal/min)	0.0019 (0.5)	0.0023 (0.6)
Lubricant-in temperature, K (°F)	350 (170)	355 (180)
Average cup temperature, K (°F)	391 (245)	419 (295)
Average cone face temperature, K (°F)	391 (245)	425 (305)

Results and Discussion

Performance Tests with Standard Design Bearing

The modified standard design bearing was tested over a range of speeds, loads, lubricant flow rates, and temperatures. Lubricant was delivered to the bearing either by jets (jet lubrication) or through holes in the cone (cone-rib lubrication).

Test spindle speeds were 6000, 10 000, 12 500, and 15 000 rpm. Thrust load was varied from 26 700 to 53 400 N (6000 to 12 000 lb); radial load was varied from 13 300 to 26 700 N (3000 to 6000 lb). Lubricant flow rate was varied from 1.9×10^{-3} to 15.1×10^{-3} m^3/min (0.5 to 4.0 gal/min). When cone-rib lubrication was used a constant 3.8×10^{-3} m^3/min (1.0 gal/min) of jet flow was also used to assure some lubrication of the roller small end. The lubricant was a 5-cS neopentylpolyol tetraester (MIL-L-23699).

The results of these tests are discussed in detail in reference 9. Briefly, it was shown that increasing speed resulted in increased bearing temperatures and power loss. Increased lubricant flow rate decreased bearing temperature, but increased heat generation. The effect of load on bearing temperatures was very small relative to speed and flow rate effects. Increasing oil-in temperature from 350 to 364 K (170° to 195° F) increased bearing temperature approximately 7 to 10 K (12° to 18° F).

Of greatest significance in these tests was the improvement in bearing performance with cone-rib lubrication compared with jet lubrication. The effect of shaft speed on cone-face temperature is shown in figure 3 for an oil-in temperature of 350 K (170° F). Increasing the shaft speed from 6000 to 15 000 rpm increases cone-face temperature by as much as 49 K (89° F). Shaft speed has a lesser effect on cone-face temperature where cone-rib lubrication is used rather than jet lubrication. It is apparent that extrapolation of the data in figure 3(a) to 15 000 rpm for jet flow rates less than 7.6×10^{-3} m^3/min (2.0 gal/min) at 350 K (170° F) oil-in temperature would give cone-face temperatures in excess of 430 K (320° F)—where previous tests had indicated that failure of the cone-

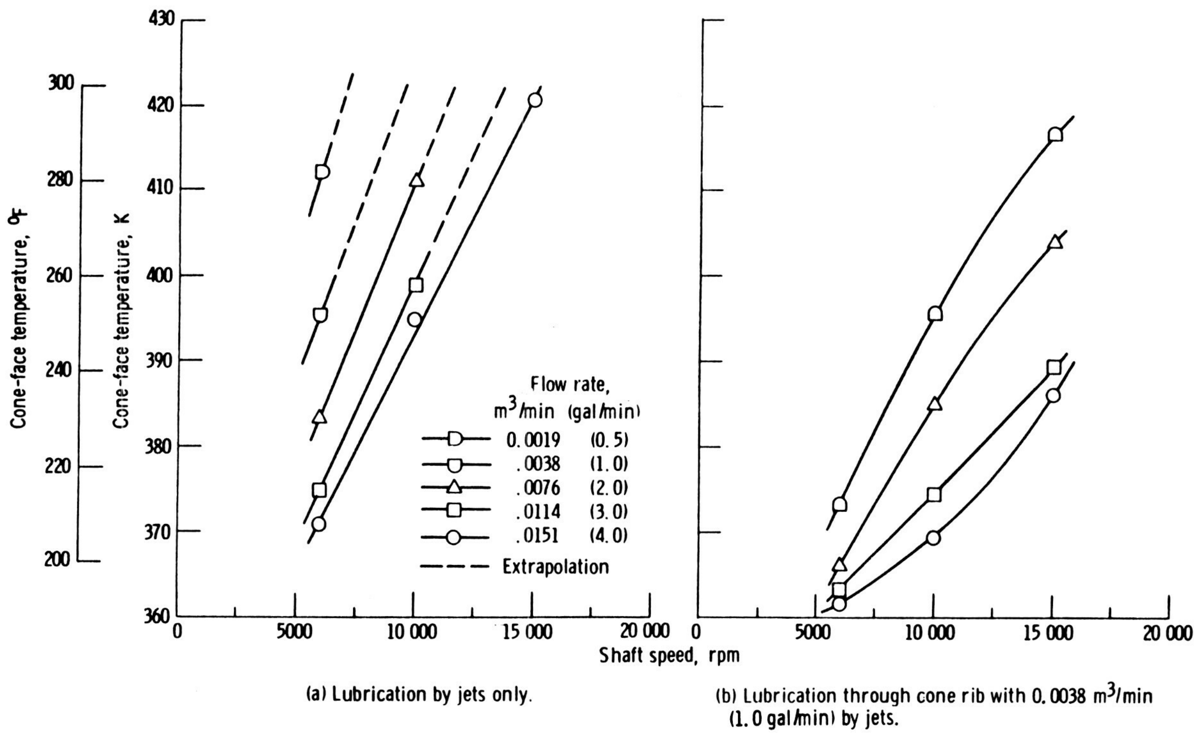

Figure 3. - Effect of shaft speed and flow rate on cone-face temperature. Oil-in temperature, 350 K (170° F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

rib would occur. For lubrication through the cone-rib (with 3.8×10^{-3} m^3/min (1.0 gal/min) jet flow), satisfactory cone-face temperatures were obtained at 15 000 rpm with total flow rates as low as 5.7×10^{-3} m^3/min (1.5 gal/min) (fig. 3(b)).

The advantage of cone-rib lubrication is further illustrated in figure 4. The difference in the temperature of the cone-face with jet lubrication and that with cone-rib lubrication increases with shaft speed. At 15 000 rpm the difference is 34 K (62° F). Even at the lower speed of 6000 rpm, the temperature improvement is an average 13 K (23° F).

It was observed that when cone-rib lubrication is used, the highest bearing temperature measured at each condition is at the cup outer surface. When jet lubrication alone is used, the highest measured temperatures were on the cone face. This effect is illustrated in figure 5 where the temperatures are plotted against shaft speed for an oil-in temperature of 364 K (195° F) and a total oil flow of 11.4×10^{-3} m^3/min (3.0 gal/min). Cone-bore and oil-out temperatures for jet lubrication and for cone-rib lubrication are not significantly different.

It is believed that, when cone-rib lubrication is used, less oil is thrown centrifugally outward to cool the cup before it leaves the bearing. Also, the oil that is directed through the cone-rib and does contact the cup has been heated somewhat in cooling the cone rib. Thus, a somewhat greater cup temperature has accompanied a cooler cone rib, but because of the critical nature of the cone-rib contact in high-speed tapered roller bearings, this small sacrifice appears justified.

The higher cup temperatures may be decreased with the use of cup cooling oil flowing in the cup housing in contact with the outer surface of the cup. Figure 6 includes some additional temperature data obtained at a shaft speed of 12 500 rpm and 5.7×10^{-3} m^3/min (1.5 gal/min) total oil flow (cone-rib flow of 1.9×10^{-3} m^3/min (0.5 gal/min) plus jet flow of 3.8×10^{-3} m^3/min (1.0 gal/min). With the addition of 2.6×10^{-3} m^3/min (0.7 gal/min) cup cooling flow (solid symbols in fig. 6), the cup outer surface temperature is decreased 14 K (25° F) without significant change in cone-face and cone-bore temperatures. Oil-out temperature was 6 K (11° F) lower because of the quantity of heat removed by the 364 K (195° F) cup cooling oil, which was measured at 380 K (225° F) upon exit from the cooling passages.

The heat generated in the bearing is dissipated in the form of heat by conduction to the lubricant and by conduction, convection, and radiation to the surrounding environment. Lubricant outlet

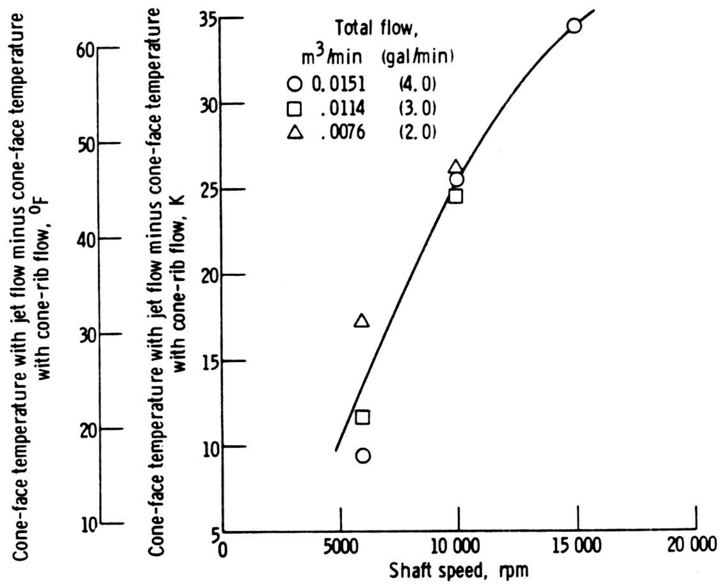

Figure 4. - Effect of shaft speed on cone-face temperature with jet lubrication minus that with cone-rib lubrication. Oil-in temperature, 350 K (170° F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

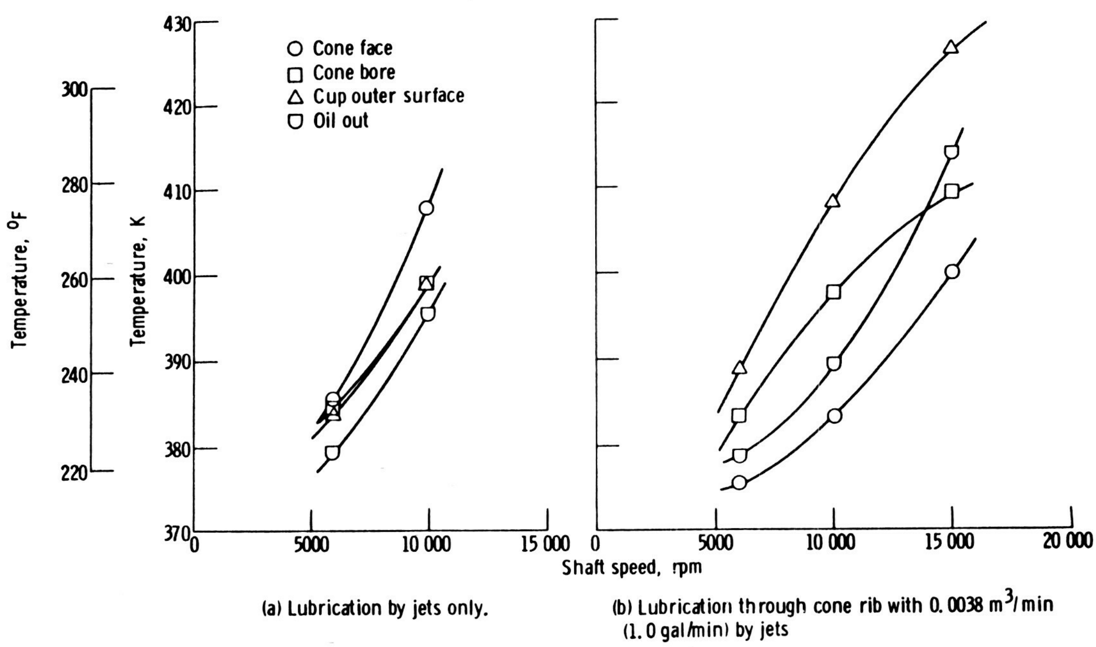

Figure 5. - Effect of jet lubrication and cone-rib lubrication on bearing and oil-out temperatures. Oil-in temperature, 364 K (195° F); total oil flow rate, 0.0114 m³/min (3.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

temperature from the bearing was measured for all conditions of flow. Heat transferred to the lubricant was calculated using the standard heat-transfer equation:

$$Q_T = MC_P(t_{\text{out}} - t_{\text{in}}) \tag{1}$$

where

Q_T total heat transfer to lubricant, J/min (Btu/min)

M lubricant mass flow, kg/min (lb/min)

C_P specific heat, J/kg K (Btu/lb °F)

t_{out} oil outlet temperature, K (F°)

t_{in} oil inlet temperature, K (°F)

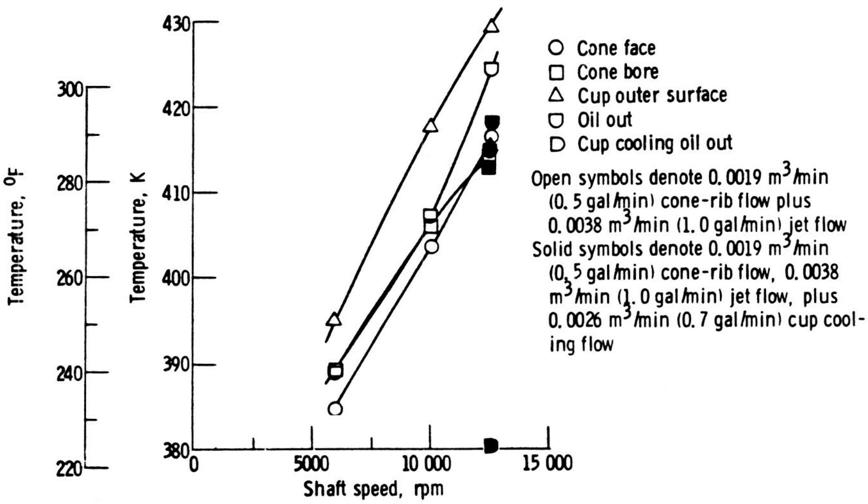

Figure 6. - Effect of cup cooling on bearing and oil-out temperatures. Oil-in temperature, 364 K (195° F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

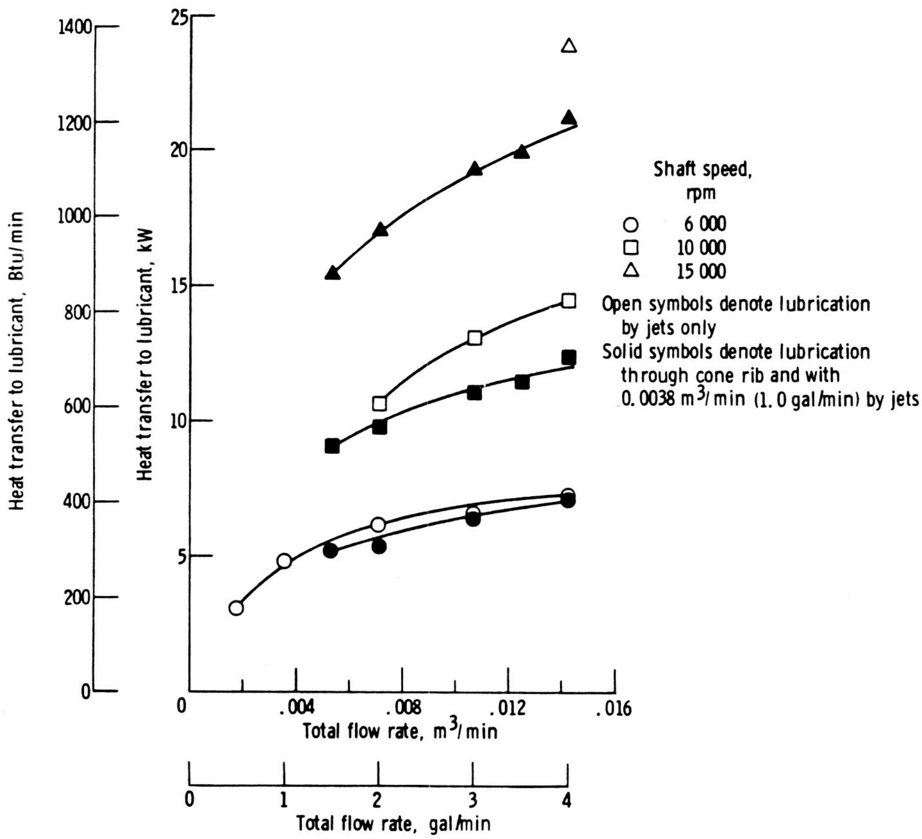

Figure 7. - Heat transferred to lubricant as function of total flow rate. Oil-in temperature, 350 K (170° F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

The result of these heat-transfer calculations are shown in figure 7 as a function of shaft speed and total flow rate. (For convenience, heat values were converted from J/min to kW.) The heat transferred to the lubricant increases with both increased shaft speed and increased lubricant flow rate. These increases are expected because of increased lubricant drag or churning. These heat quantities are a portion of the heat generated in the test bearings and do not include heat transferred from the bearing by conduction, convection, and radiation. At higher bearing temperatures, the heat transferred by these latter forms should become a greater portion of the total.

Lower heat generation is observed with cone-rib lubrication than with jet lubrication at the same flow rates. The effect is greater at higher total flow rates and higher speeds.

Performance Tests with Computer Optimized Design Bearing

The computer optimized design bearing was tested over the same load and lubricant flow rate conditions as the standard bearing. The optimized design bearing was run at speeds to 20 000 rpm, whereas the standard bearing was limited to 15 000 rpm.

Similar trends were observed with the optimized design bearing and the standard design bearing; that is, bearing temperature and heat generation increased with increased shaft speed. Also, with increased lubricant flow rates, bearing temperatures decreased and heat generation increased. More detailed discussion of these results can be found in reference 10.

The improvement in performance of the computer-optimized design bearing was assessed by comparing temperature and heat generation with the two designs at identical conditions. Although lubricant to the standard bearing was directed through holes in the large end of the cone to the cone-rib surface, lubricant at the small end was through a pair of jets directed at the small end of the rollers. Comparisons herein were made with equal flow rates for both bearings at both the large end and the small end. At the small end, a constant rate of 3.8×10^{-3} m^3/min (1.0 gal/min) was used, and oil was fed through jets for the standard design and through the cone small end for the high-speed design.

Materials with higher temperature capabilities were used for the computer-optimized design bearings. Cups, cones, and rollers were made of CBS-1000M, and the cage was of AISI 4340. The standard design bearing had AISI 4320 cups, cones, and rollers and AISI 1010 cages.

In figure 8 cone-face temperatures of the two bearing designs are compared. The data symbols are used to identify the curves at each flow rate. At 15 000 rpm the optimized design bearing operates 8 to 11 K (15° to 20° F) cooler than the standard design. The improvement at 6000 rpm is much less.

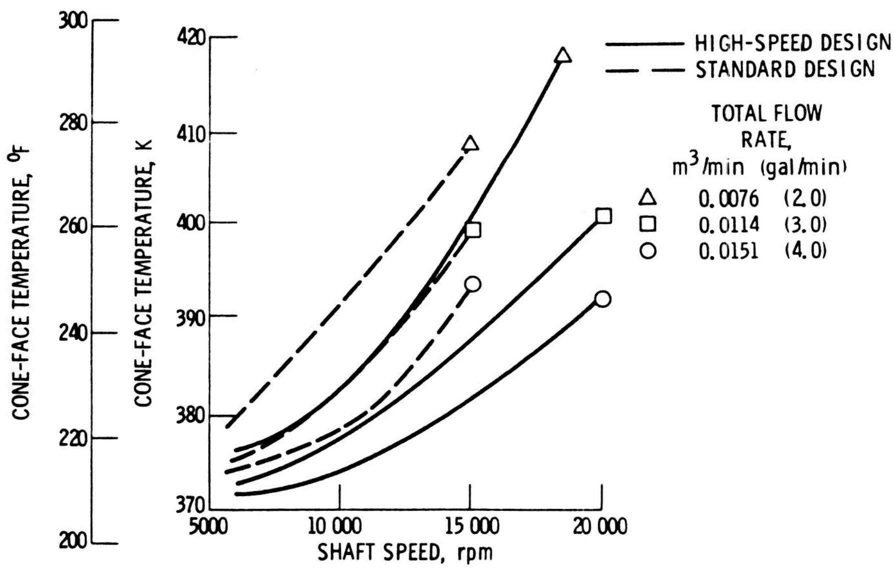

Figure 8. - Cone face temperature with optimized design compared with that with standard design. Oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

The cup outer-surface temperature is also lower for the optimized design bearing as shown in figure 9. Here, the improvement is slightly less than that at the cone face.

Figure 10 compares data for the standard and optimized design bearings showing heat transferred to the lubricant at 6000 and 15 000 rpm. The optimized design bearing has lower heat generation, as represented by heat transferred to the lubricant, than the standard design bearing at both speeds. As shown in figure 11, the improvement with the optimized design is approximately 16 percent throughout the range of 6000 to 15 000 rpm.

Life Tests with 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 1.9×10^{-3} m³/min (0.5 gal/min), a jet flow rate of 3.8×10^{-3} m³/min (1.0 gal/min), and a cup cooling flow rate of 2.6×10^{-3} m³/min (0.7 gal/min). The lubricant-in temperature was 366 K (200° F). These conditions were selected to maintain cup and cone temperatures less than 422 K (300° F), based on results of the performance tests with identical bearings previously discussed. 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 hr, revealed that a shallow surface distress was occurring on the raceway and roller taper surfaces. This type of distress is called peeling (ref. 11) and appears as a very shallow area of uniform depth. Typical peeled areas in these bearings, approximately 0.008 mm (0.0003 in.) deep, are shown in figure 12. The peeling tended to initiate at minor surface defects such as the deeper surface scratches or indentations.

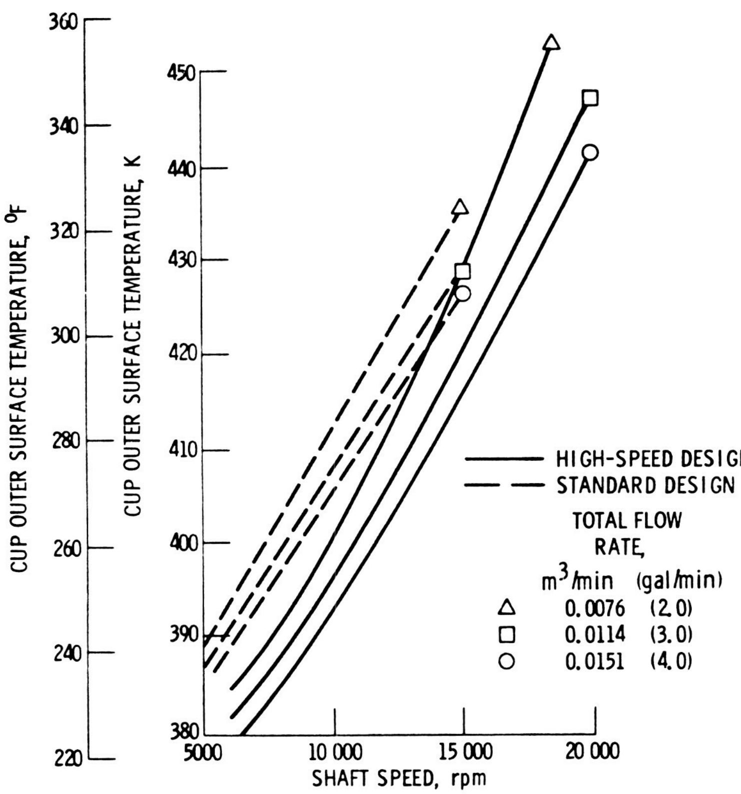

Figure 9. - Cup outer surface temperature with optimized design compared with that with standard design. Oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m³/min (1.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

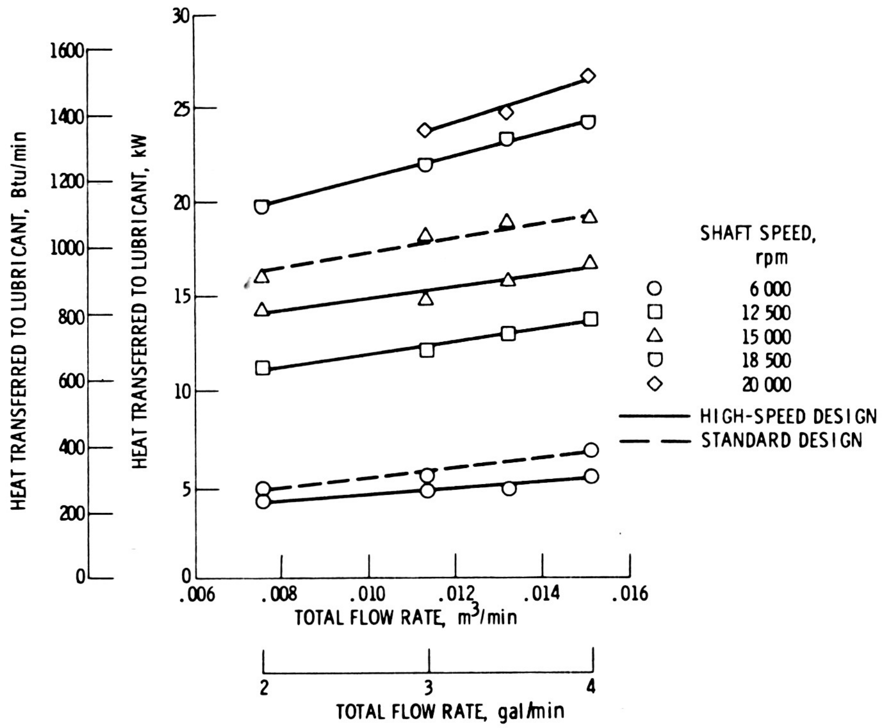

Figure 10. - Heat transferred to lubricant as function of total flow rate with optimized design. Oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

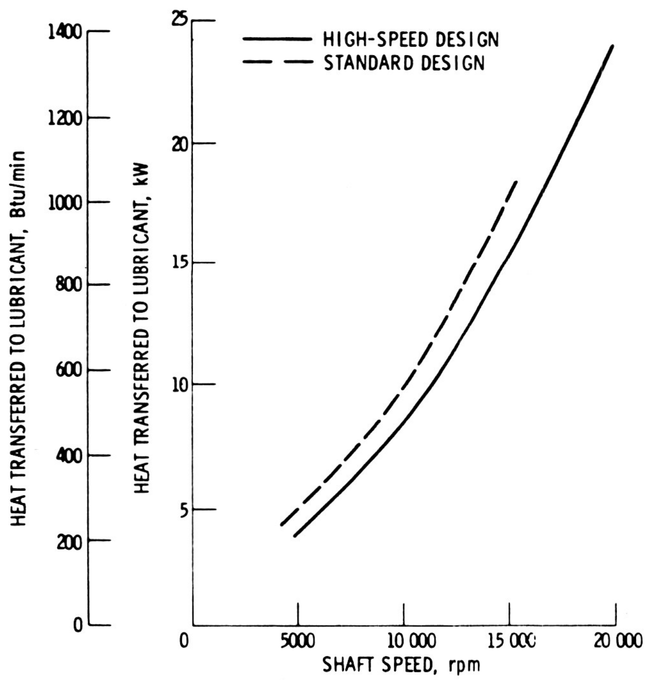

Figure 11. - Heat transferred to lubricant as function of shaft speed. Oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gal/min); large end flow rate, 0.0076 m^3/min (2.0 gal/min); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

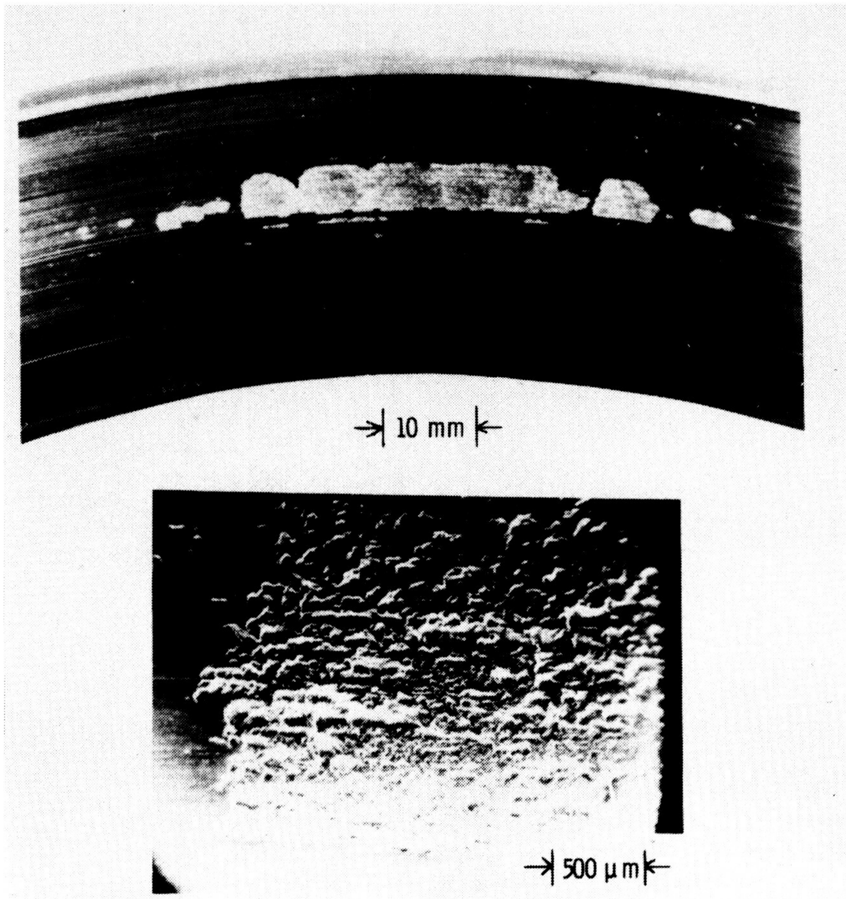

Figure 12. - Peeling failure on cup raceway surface after 569 hr with standard design bearing run at 12 500 rpm.

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. Further stress concentration occurred at the center of the roller and aggravated the peeling. This exaggerated crown is attributed to material growth caused by an uneven transformation of retained austenite to martensite (refs. 2 and 12).

The major cause of peeling type surface distress is believed to be due to an inadequate lubricant film parameter, Λ, which is the ratio of the elastohydrodynamic (EHD) 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 similar.

Improved surface finish. – Further reduction of bearing temperatures to improve the lubricant film parameter was considered to be impractical since lower lubricant-in temperatures would not be representative of helicopter transmissions and a further increase in flow rate would not result in significantly lower bearing temperatures (ref. 9). The alternative 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 where the raceways and roller tapers were honed to improve the surfaces from the values given in table II to 0.10 μm (4 μin.) or better. After honing, the measured roughness of the raceways and roller tapers was typically 0.09 and 0.06 μm (3.5 and 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 hr 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 V.

Life tests at 12 500 rpm. – The remaining test bearings of both materials, after rehoning, were run at the test conditions shown in table IV. Twelve bearings of the AISI 4320 material ran to the 1100-hr cutoff without failure of any type. At these conditions the rated catalog life of this bearing design is 102 hr, so that the experimental 10-percent life is greater than 10 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 rehoned bearings of the CBS 1000M material were run to spalling fatigue failure or to the 1100-hr cutoff. Twelve of the bearings ran to 1100-hr without failure. Three bearings experienced spalls on the cup or cone raceways. One bearing was suspended at 820 hr without spalling failure, because 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 reference 13 in figure 13. A least-squares line drawn through the three failure points shows an estimated 10-percent life of approximately 600 hr, or about six times the rated catalog life.

A comparison of the results with the two materials shows that the CBS 1000M bearing life is less than the AISI 4320 bearing life. However, a quantitative estimate of the difference is not possible

TABLE V. - EFFECT OF LUBRICANT FILM PARAMETER ON PEELING OF TAPERED-ROLLER BEARING RACEWAY

Bearing temperatures, 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)	.15 (6)	.13 (5)	.20 (7.8)	.38 (15)	1.9	Peeling
391 (245)	.09 (3.5)	.06 (2.5)	.11 (4.3)	.38 (15)	3.5	No Peeling

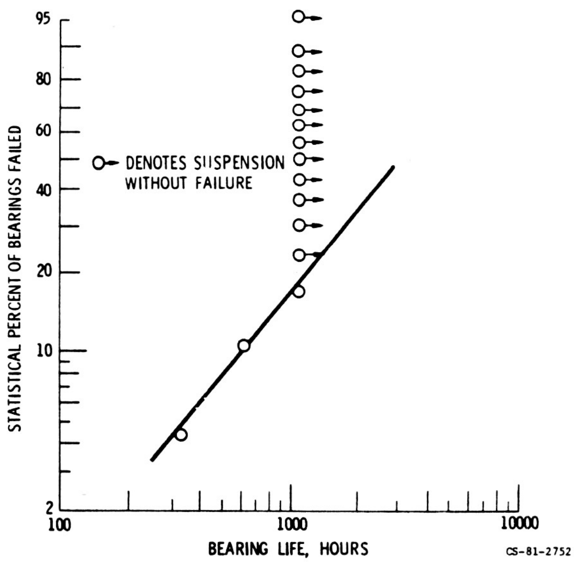

Figure 13. - Rolling-element fatigue life of standard design CBS1000M tapered-roller bearings at 12 500 rpm. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); bearing temperature, 394 K (250° F).

from these results since no fatigue failures occurred with the AISI 4320 bearings. Metallurgical analysis of the CBS 1000M bearings revealed that the cup and cone materials had coarser than desired grain structure. These bearings were made from the first large heat of the material and were exposed to forging temperatures later found to be excessive. Recent knowledge in forging and heat treating this material should provide a more desirable structure for better rolling-element fatigue life.

Life Tests with 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. Under these load and speed conditions, the rated catalog life of this bearing design is 46 hr. Lubricant flow rates and temperatures are shown in table IV.

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

After 188 hr, the other test was stopped because of 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 serial number 78-9 was subsequently cut partially through and fractured at that location to reveal the crack. The extent of the crack is shown in figure 14. The dimensions are approximately 9.5 mm (0.37 in.) long and 1 mm (0.04 in.) deep.

The cone of bearing serial number 78-10, with the complete fracture is shown in figure 15. Initial crack dimensions were observed to be approximately 22 mm (0.87 in.) long and 6 mm (0.24 in.) deep.

Fatigue spalling was noted on both cone raceways adjacent to the cracks. It is believed that the fatigue cracks, related initially to the spalling process, propagated at an accelerated rate because of 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 serial number 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 references 14 and 15 it was shown that the critical crack size can readily be reached in through-hardened AISI M-50 and that fracture of inner races (or cones in the case of tapered roller bearings) is probable.

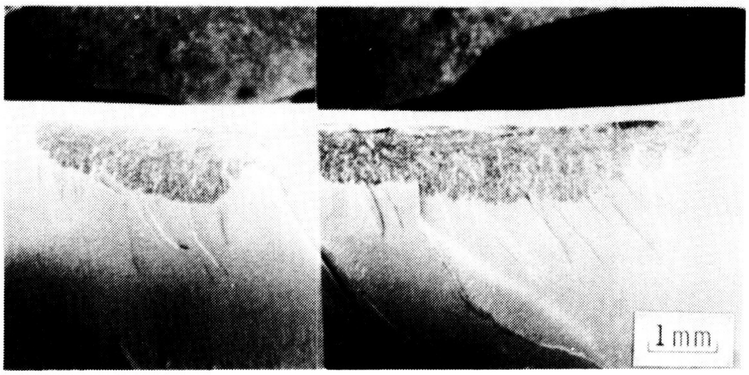

Figure 14. - Photomicrograph of cone raceway crack on AISI M-50 bearing serial number 78-9 after 188 hr of operation at 18 500 rpm.

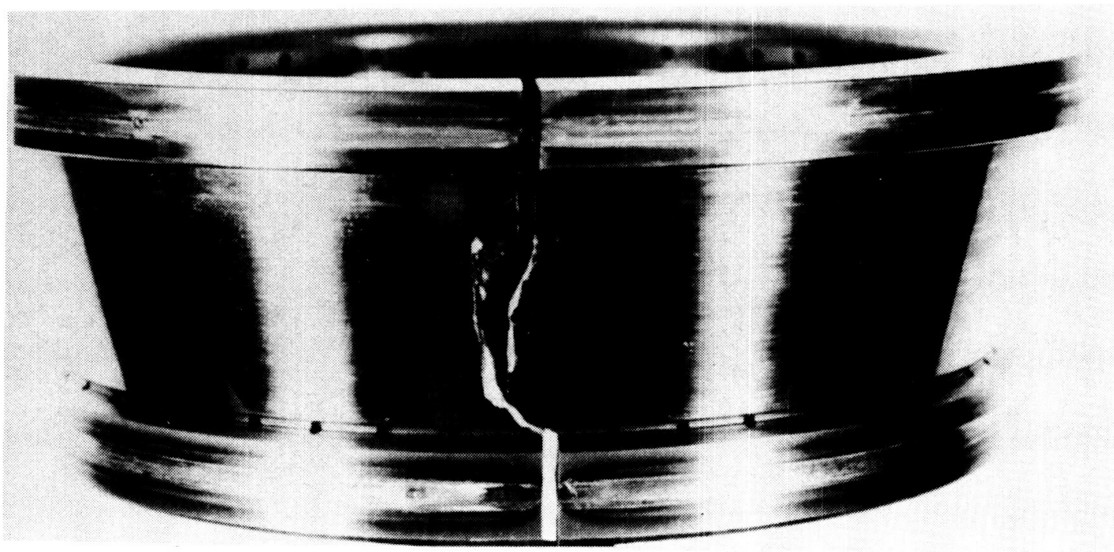

Figure 15. - Fractured cone of AISI M-50 bearing serial number 78-10 after 188 hr of operation at 18 500 rpm.

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 risk of catastrophic fracture of the cone similar to that for ball bearing inner rings at high speeds (ref. 16). It is apparent that materials for high-speed tapered-roller bearings must have higher fracture toughness such as that associated with case-hardened materials.

CBS 1000M bearing tests. – 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-hr cutoff time without failure. One bearing suffered a severe failure after 135 hr as a result of failure detection and shutdown-system malfunction. The extent of damage was too great to determine the cause of the failure. However, close observation revealed two spalls in the cone raceway. The largest spall is shown in figure 16. No cracks or fracture of the cone was observed, although considerable damage to the raceway, cone rib, and cage occurred. This result offers some 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 occurs.

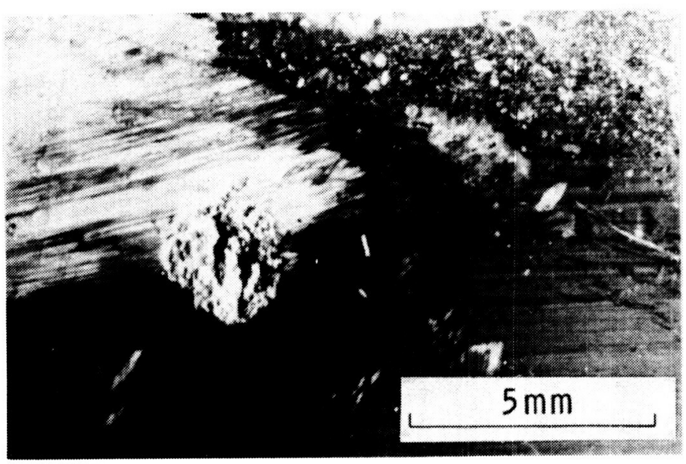

Figure 16. - Fatigue spall on cone raceway of CBS-1000M bearing serial number 76-17 after 135 hr of operation at 18 500 rpm.

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

Summary of Results

The performance and endurance life of 120.65-mm (4.75-in.) bore tapered-roller bearings of two designs were investigated under heavy combined radial and thrust loads. Performance tests were run to 15 000 rpm with the modified standard design bearing and to 20 000 rpm with the computer-optimized design bearing. Temperature distribution and bearing heat generation were determined as functions of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied to the test bearings either by jets or through holes in the cone to the undercuts at the edges of the raceway.

Endurance tests were run with the modified standard design bearings of two materials at 12 500 rpm. With the computer-optimized design bearings, endurance tests were run with two materials at 18 500 rpm. Bearing temperatures were maintained in the range from 394 to 422 K (250° to 300° F). The following results were obtained:

1. Direct cone-rib lubrication significantly improved the performance of the modified standard design tapered-roller bearings at high speeds. With cone-rib lubrication, total flow rates as low as 5.7×10^{-3} m^3/min (1.5 gal/min) provided stable bearing operation at 15 000 rpm, whereas with jet lubrication alone, a flow rate of 15.1×10^{-3} m^3/min (4.0 gal/min) was required. Bearing heat generation was less with cone-rib lubrication than with jet lubrication at the same lubricant flow rates.
2. The computer-optimized, high-speed-design, tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Stable temperatures were reached, and no surface distress was observed.
3. Bearing temperatures and heat generation with the high-speed-design bearing were significantly less than those of a modified standard bearing.
4. Bearing temperatures and heat generation increased as expected with increased shaft speed. The highest temperatures measured were at the cup outer surface.
5. With increased lubricant flow rate, bearing temperatures decreased, and bearing heat generation increased.
6. Standard design bearings of CBS 1000M material gave an estimated 10-percent life about six times the rated catalog life. Twelve identical bearings of AISI 4320 material ran to a cutoff time of 10 times the rated catalog life without failure.
7. Under conditions of marginal lubricant-film-thickness-to composite-surface-roughness ratio, peeling surface distress was a dominant mode of failure. However, by reducing bearing temperatures to increase lubricant film thickness and rehoning the raceways and rollers to improve surface finish, the peeling mode of failure was effectively eliminated.
8. Cracking and fracture of the cones of AISI M-50 bearings occurred as a result of the high tensile hoop stresses at 18 500 rpm with the optimized high-speed design bearings.
9. CBS 1000M bearings of the optimized high-speed design did not crack or fracture. Four of these bearings ran to 24 times the rated catalog life without failure.

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