Predicted and Experimental Performance of Large-Bore High-Speed Ball and Roller Bearings

Harold H. Coe*

Trends in gas-turbine design have indicated that future aircraft engines may require bearings that can operate reliably at DN values of 3 million or higher (refs. 1 to 3) (the speed parameter DN is the bearing bore in millimeters multiplied by the shaft speed in rpm). Consequently, there has been a large amount of work done in the area of high-speed bearings over the last several years. Successful operation of ball bearings at 3 million DN was reported in references 4 and 5. Roller bearing operation to 3 million DN was reported in references 3 and 6.

The question of how to design bearings for high-speed applications is increasingly being answered by computer studies (refs. 3 and 7). Several comprehensive computer programs currently in use are capable of predicting rolling-bearing operating and performance characteristics (e.g., refs. 8 to 14). These programs generally accept input data of bearing internal geometry (such as sizes, clearance, and contact angles), bearing material and lubricant properties, and bearing operating conditions (load, speed, and ambient temperature). The programs then solve several sets of equations that characterize rolling-element bearings. The output produced typically consists of rolling-element loads and Hertz stresses, operating contact angles, component speed, heat generation, local temperatures, bearing fatigue life, and power loss. Two of these programs, Shaberth (ref. 9) and Cybean (refs. 10 and 11) developed under government sponsorship, have been used extensively at the Lewis Research Center.

After a computer program is developed, it is important that values calculated using such program be compared with actual bearing performance data to assess the programs predictive capability. Therefore, the objective of the work reported herein was to compare the values of inner- and outer-race temperatures, cage speed, and heat transferred to the lubricant or bearing power loss, calculated using the computer programs Shaberth and Cybean, with the corresponding experimental data for the large-bore ball and roller bearings described in references 6 and 15. Most of this work has been previously reported in references 16 to 18.

Test Data

Ball Bearing

The experimental work used for comparison, originally reported in reference 15, was performed on the high-speed bearing tester described in reference 19. Lubrication was provided to the test bearings through a jet feed system with two lubricant jets positioned 180° apart. The jets had a double orifice (fig. 1). The lubricant used was a tetraester, type II oil qualified to the MIL–L–23699 specification. The major properties of the oil are listed in table I.

The test bearing specifications are listed in table II. The bearings were 120-mm-bore, split inner race with fifteen 20.6-mm (0.8125-in.) diameter balls and a contact angle of 20°. The races and balls were made of double-vacuum-melted (VIM–VAR) AISI M–50 material. The bearings had one-piece machined cages which were inner-race riding. These cages were made of silver-plated AMS 6415 steel.

Power loss per bearing was determined by measuring line-to-line voltage and line current to the test-rig drive motor. Motor-drive power was then calculated as a function of line current, reflecting bearing power usage at the various operating speeds.

Data were recorded at bearing thrust loads of 6672, 13 350, and 22 240 N (1500, 3000, and 5000 lb) and at three shaft speeds, 12 000, 16 700, and 20 800 rpm. The lubricant flow rates were 0.0038 and 0.0083 m^3/min (1.0 and 2.2 gal/min). The oil inlet temperature was held constant at 394 K (250° F).

*NASA Lewis Research Center.

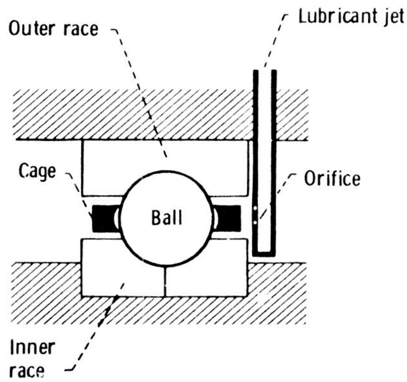


Figure 1. - Bearing lubrication schematic. Number of jets, two per bearing; dual orifice; inner-land riding cage.

TABLE I. - PROPERTIES OF TETRAESTER LUBRICANT[a]

Additives	Antiwear, oxidation inhibitor, antifoam
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Specific heat at 477 K (400° F) J/(kg)(K), (Btu/(lb)(°F))	2340 (0.54)
Thermal conductivity at 477 K (400° F), J/(m)(sec)(K), (Btu/(hr)(ft)(°F))	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

[a]From ref. 15.

TABLE II. - BALL BEARING[a] SPECIFICATIONS

Bearing outside diameter, mm	190
Bearing inside diameter, mm	120
Bearing width, mm	35
Bearing contact angle, deg	20
Outer-race curvature	0.52
Inner-race curvature	0.54
Number of balls.	15
Ball diameter, mm (in.)	20.6 (0.8125)
Retainer design	One-piece machined
Retainer material	AMS 6415[b]
Race and ball material	AISI M-50[c]
Ball surface finish, μm (μin.)	0.025 (1)
Raceway surface finish, μm (μin.)	0.05 (2)

[a]Tolerance grade ABEC-5.
[b]Silver plated per AMS-2410.
[c]Vacuum-induction melted, vacuum-arc remelted.

Roller Bearing

The experimental roller bearing data used for comparison were initially reported in reference 6. In this reference a large-bore roller bearing was tested at speeds up to 3 million DN. Lubrication was provided to the test bearing through axial grooves under the inner race with small radial holes through to the rolling elements. The inner ring was also cooled by oil flowing through axial grooves that had no radial holes. About one-half of the total oil introduced to the inner ring was used for cooling only.

The lubricant used was also a tetraester, type II oil qualified to the MIL-L-23699 specification (table I). The bearing tester is described in detail in reference 6.

The test roller bearing had a 118-mm bore, a flanged inner ring, and 28 rollers, each 12.65 mm (0.4979 in.) in diameter by 14.56 mm (0.573 in.) long. More complete specifications are shown in table III.

Oil inlet temperature was held constant at 366 K (200° F). Accurate measurement of bearing oil inlet and outlet temperatures allowed determination of the amount of heat transferred to the lubricant at any operating condition. Data were recorded at bearing loads of 2220, 4450, 6670, and 8900 N (500, 1000, 1500, and 2000 lb), and at shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The total oil flow rates to the inner ring varied from 0.0019 to 0.0102 m^3/min (0.5 to 2.7 gal/min).

Computer Programs

Two computer programs were used. The program called Shaberth (ref. 8) was used to calculate the ball bearing data, and Cybean (refs. 10 to 12) was used for the cylindrical roller bearing calculations. Shaberth can analyze up to five bearings on a single shaft and includes shaft deflection. Cybean analyzes a single cylindrical roller bearing. Both programs are capable of calculating the thermal and kinematic performance of high-speed bearings, and Cybean includes a roller skew prediction for misalined conditions. The calculations include determination of inner- and outer-ring temperatures, oil outlet temperatures, cage speed, and bearing power loss.

Use of either Shaberth or Cybean to predict bearing performance requires as input an estimate of the volume percent of the bearing cavity that is occupied by the lubricant. The bearing cavity is defined as the space between the inner- and outer-races that is not occupied by the cage or the rolling elements. The authors of references 8 and 11 recommended that the values used be less than 5 percent. When these programs are used for a thermal analysis, additional input is required, since all the thermal nodes must be defined. The maximum number of nodes permitted is 100. With Shaberth, a relatively simple thermal grid system was chosen, using only 17 nodes for the ball bearing. With Cybean, 41 nodes were used, including 19 in the lubricant system (fig. 2). It should be noted here that, because of the simple nodal system, the lubricant flow rate was not included directly in the thermal calculations with Shaberth. However, the flow rate was used directly with the Cybean program.

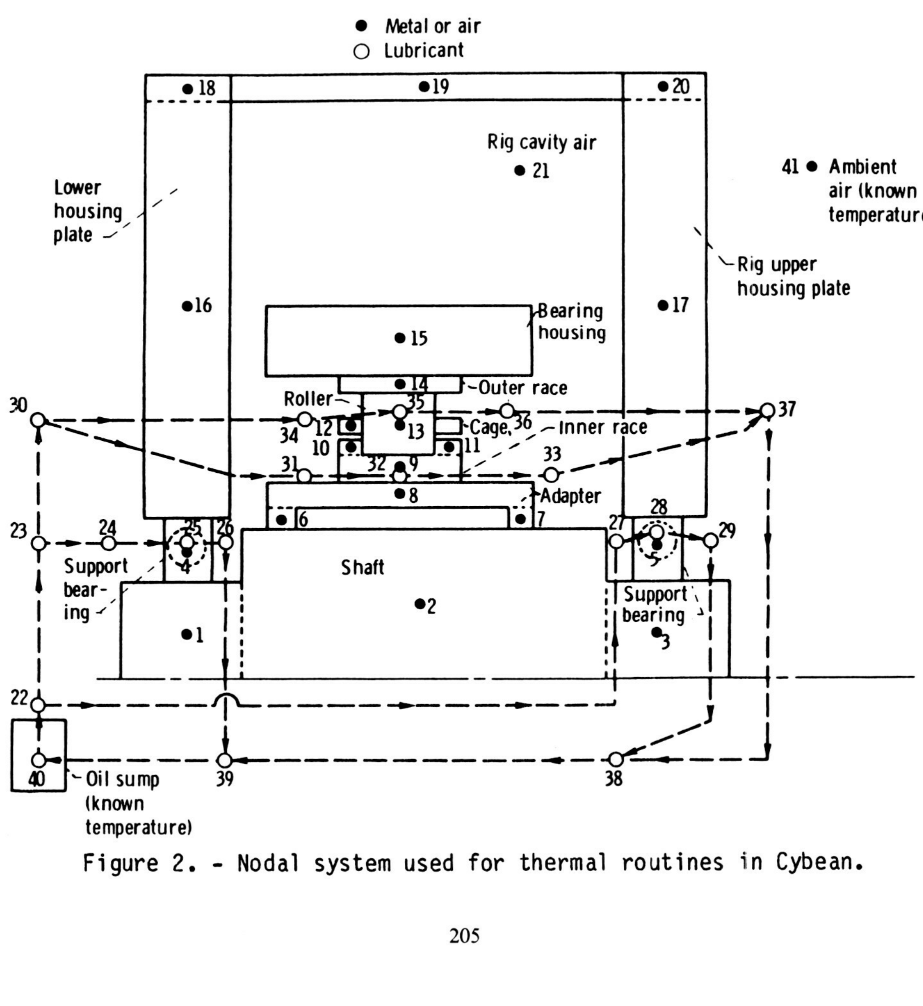


Figure 2. - Nodal system used for thermal routines in Cybean.

TABLE III. - ROLLER-BEARING SPECIFICATIONS

Inner race	
Bore diameter, mm (in.)	118 (4.6457)
Raceway diameter, mm (in.)	131.66 (5.1834)
Flange diameter, mm (in.)	137.47 (5.4122)
Total width, mm (in.)	26.92 (1.060)
Groove width, mm (in.)	14.59 (0.5746)
Flange angle, deg	0.6
Outer race	
Outer diameter, mm (in.)	164.49 (6.4760)
Raceway diameter, mm (in.)	157.08 (6.1842)
Total width, mm (in.)	23.9 (0.942)
Rollers	
Diameter, mm (in.)	12.65 (0.4979)
Length, mm (in.):	
Overall	14.56 (0.5733)
Effective	13.04 (0.5133)
Flat	8.40 (0.3307)
Crown radius, mm (in.)	622.3 (24.5)
End radius, mm (in.)	381.0 (15)
Number	28
Cage	
Land diameter, mm (in.)	137.95 (5.4312)
Axial pocket clearance, mm (in.)	0.020 (0.0008)
Tangential pocket clearance, mm (in.)	0.221 (0.0087)
Single rail width, mm (in.)	4.6 (0.18)
Bearing	
Cold diametral clearance, mm (in.)	0.12 (0.0047)

Results and Discussion

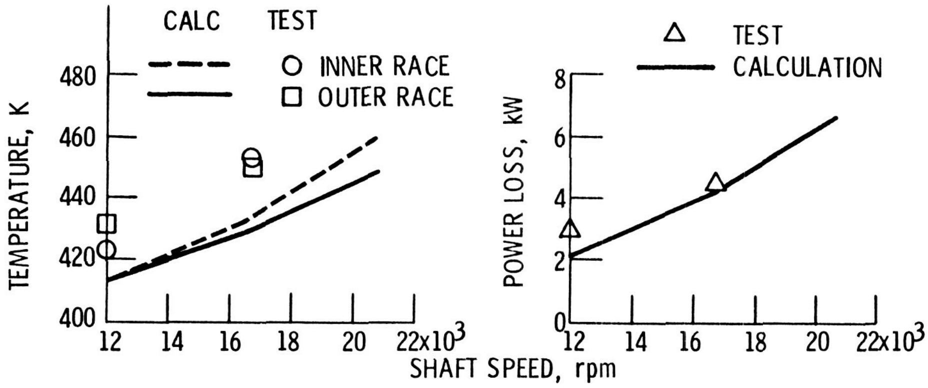
To effect a direct comparison of predicted and experimental bearing performance, the computer programs were generally run at the stated operating conditions of the bearings tested (refs. 6 and 15). All combinations of load, speed, and flow rate were not computed, because they are not all necessary for the comparisons and would require unnecessary computer time. Note that in all the figures in this report, the calculated values are always just connected with straight line segments.

Shaft Bearing Program (Shaberth)

Calculations were made for the 6672-N (1500-lb) thrust load case at 12 000, 16 700, and 20 000 rpm. The lubricant volume was set at 1.0 percent. The calculated results are shown in figure 3 compared with the experimental data for an oil flow rate of 3.8×10^{-3} m^3/min (1 gal/min). The calculated values of inner- and outer-race temperatures are lower than the experimental data, and the calculated bearing power loss (fig. 3(b)) is close to the measured value.

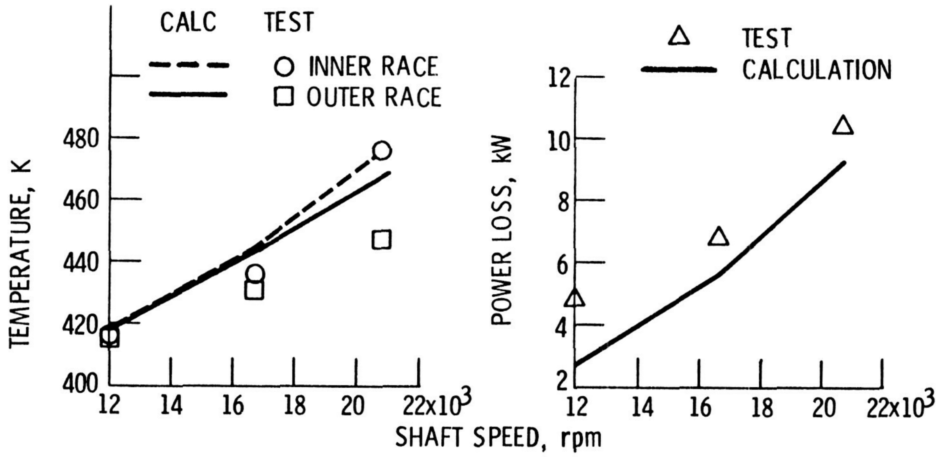
To compare with the experimental data taken at the oil flow rate of 8.3×10^{-3} m^3/min (2.2 gal/min), the Shaberth program was run for the same conditions given previously except that the lubricant volume in the cavity was set at 2 percent. This value was chosen assuming that the percentage of lubricant should increase with increased flow rate. The results are shown in figure 4. The calculated race temperatures are reasonably close to the experimental data. The calculated bearing power losses resulted in fairly good agreement with the experimental data.

A comparison of figures 3 and 4 shows that the experimental temperature data decreased with the increased flow rate; whereas the calculated temperature increased with the increased percent lubricant. The predicted values would have undoubtedly been even better if flow rate were included



(a) Race temperature. (b) Bearing power loss.

Figure 3. - Comparison of calculated and experimental (ref.15) bearing operating characteristics as functions of shaft speed using Shaberth computer program. Lubricant flow rate, 0 0038 m^3/min (1 gal/min); volume of lubricant, 1.0 percent; thrust load, 6672 N (1500 lb).



(a) Race temperature. (b) Bearing power loss.

Figure 4. - Comparison of calculated and experimental (ref.15) bearing temperatures and power loss as function of shaft speed using Shaberth computer program. Lubricant flow rate, 0.0083 m^3/min (2.2 gal/min); volume of lubricant, 2.0 percent; thrust load, 6672 N (1500 lb).

the thermal calculations. The experimental bearing power loss increased with flow rate, and the calculated values increased with percent lubricant. To observe how the race temperatures and power loss might vary, the program was run at one condition for several other values of percent lubricant. The results with 6672-N (1500-lb) thrust load and 16 700-rpm shaft speed are shown in figure 5. Both the inner- and outer-race temperatures, as well as the bearing power loss, increased linearly with the volume percent over the range calculated. The change in temperature is about 10 percent over the volume range, while the change in power loss is a very significant 150 percent.

The relatively good agreement between calculated and measured values shown in figures 3 and 4 indicates that for these operating conditions the values of percent lubricant in the bearing cavity assumed for the comparison calculations were reasonably correct.

The Shaberth program (as noted in ref. 8) uses a modification to the film thickness calculation as proposed by Loewenthal, Parker, and Zaretsky (ref. 20). Therefore, the lubricant film thicknesses, as calculated by the program for the same 6672-N (1500-lbf) load and 8.3×10^{-3}-m^3/min (2.2-gal/min) oil flow case as before, were plotted as shown in figure 6. The values show little change with shaft speed, and the outer-race contact film is slightly thicker than that at the inner race. Apparently the effects of speed on film thickness are offset by the effects of temperature and load.

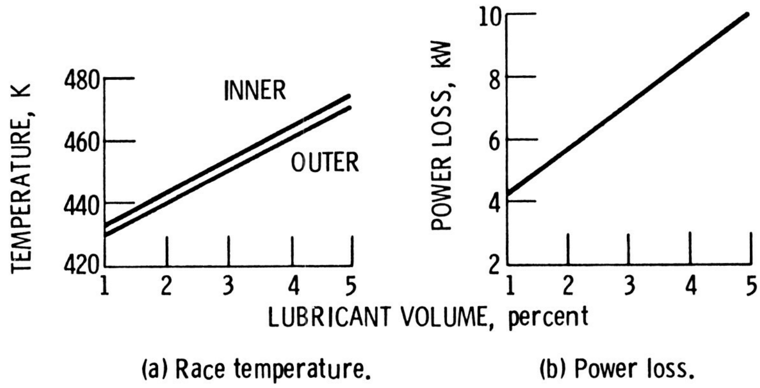


Figure 5. - Calculated bearing operating characteristics as functions of volume percent lubricant in bearing cavity using Shaberth program. Thrust load, 6672 N (1500 lb); shaft speed, 16 700 rpm.

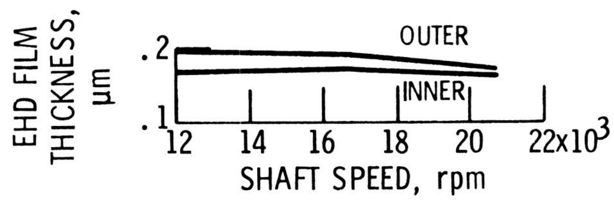


Figure 6. - Values of elastohydrodynamic (EHD) film thickness at inner- and outer- race-ball contacts as functions of speed as calculated by Shaberth computer program. Thrust load, 6672 N (1500 lb); lubricant volume, 2 percent.

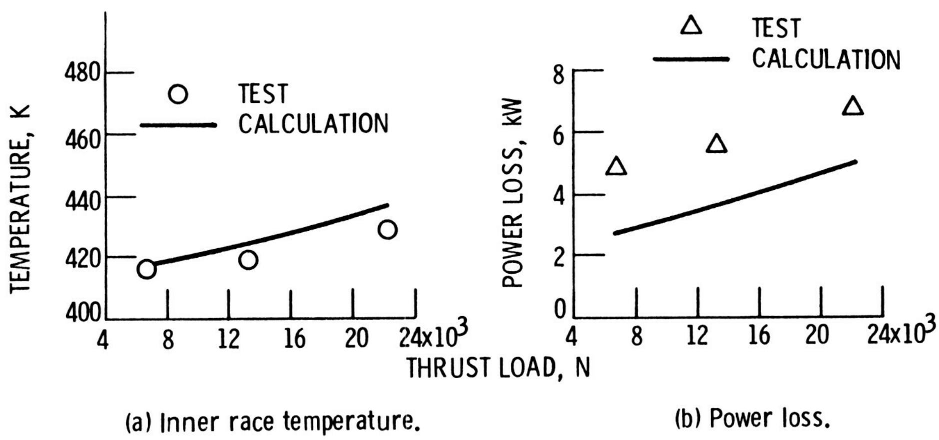
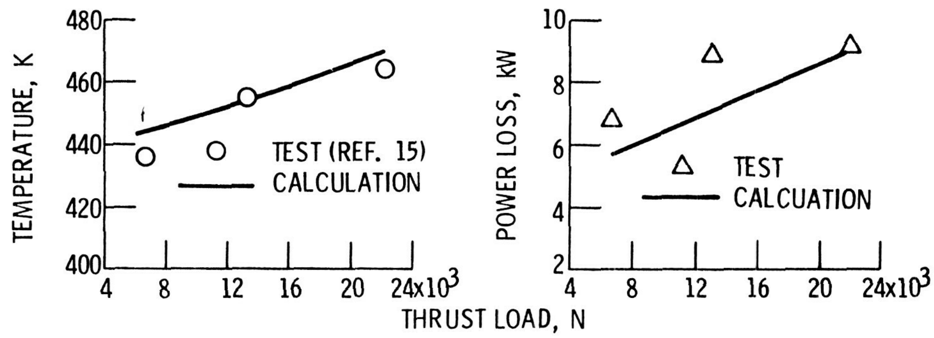


Figure 7. - Comparison of calculated and experimental (ref.15) values of race temperatures and bearing power loss as functions of thrust load using Shaberth. Shaft speed, 12 000 rpm; lubricant flow rate, 8.3×10^{-3} m^3/min (2.2 gal/min); lubricant volume, 2.0 percent.

To determine the effect of load on bearing operation, Shaberth was run at the 13 350- and 22 240-N (3000- and 5000-lbf) load conditions. The results are shown in figures 7 and 8, where the inner-race temperatures and bearing power loss are plotted as functions of bearing thrust load for shaft speeds of 12 000 and 16 700 rpm. The calculated values increase with thrust load and give a reasonably close prediction of the experimental data.

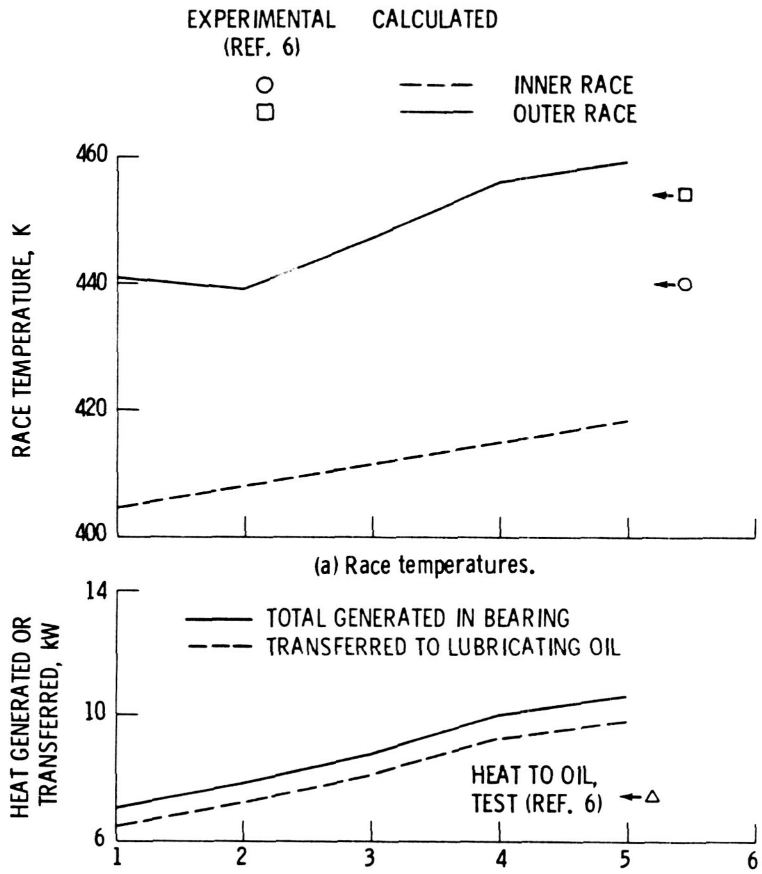
Cylindrical Roller Bearing Program (Cybean)

Using the computer program Cybean, the effect of load was calculated using one speed (20 000 rpm), the effect of speed was determined using one load (8900 N (2000 lb)), and the effect of flow rate was observed using one load and one speed (8900 N (2000 lb) and 25 500 rpm).



(a) Inner race temperature. (b) Power loss.

Figure 8. - Comparison of calculated and experimental values of bearing characteristics as functions of thrust load using Shaberth. Shaft speed, 16 700 rpm; lubricant flow rate, 0.0083 m^3/min (2.2 gal/min); lubricant volume, 2.0 percent.



(b) Heat generated in bearing, or transferred to lubricant.

Figure 9. - Calculated bearing operating characteristics as functions of volume percent lubricant. (Test data shown for comparison.) Thrust load, 4450 N (1000 lb); shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 m^3/min (1.5 gal/min).

Effect of lubricant volume. – To determine how the race temperatures and bearing heat generation vary with the value assumed for percent lubricant in the bearing cavity, the program was run for several values of this volume percent at the 4450-N (1000-lb), 20 000-rpm condition. The total oil flow rate chosen was 0.0057 m^3/min (1.5 gal/min). The results are shown in figure 9. Also shown are the corresponding experimental data points.

The race temperatures (fig. 9(a)) increased with increasing lubricant volume. This would be expected since the fluid drag on the rollers and the cage would increase with the amount of liquid available. Over the full range from 1 to 5 percent oil volume, the temperature changes seem to be nearly linear and not too large, about 5 percent at these conditions.

The total heat generated in the bearing (fig. 9(b)) increased with increasing lubricant volume. These changes were also linear but the total change in heat generation over the volume range was a more significant 50 percent. Since reference 6 includes data on heat transferred to the lubricant (as an indication of the power loss within a bearing), this type of data was calculated from the computer-predicted oil-outlet temperature and is also shown in figure 9(b). The amount of heat transferred to the oil closely follows the amount of heat generated in the bearing. Over this range of volume percent, the amount of heat transferred to the lubricant is about 90 percent of the heat generated in the bearing. Based on the experimental data for this test condition, the range of volume percent from 1 to 5 percent is adequate for the outer-race temperature and the heat transferred to the oil. The calculated inner-race temperature remained below the experimental value for the whole volume range.

Effect of bearing load and speed. – The computer program Cybean was run to determine the effect of bearing load on race temperature and heat generation. Calculations were made with a lubricant flow rate of 0.0057 m^3/min (1.5 gal/min) for two lubricant volumes (2 and 3 percent), and a shaft speed of 20 000 rpm. The results, compared with experimental data, are shown in figure 10 for radial loads from 2220 to 8900 N (500 to 2000 lb). The predicted race temperatures (fig. 10(a)) increase very slightly over the load range, and the experimental values are practically constant. While the outer-race temperatures compare favorably, the predicted inner-race temperatures remain about

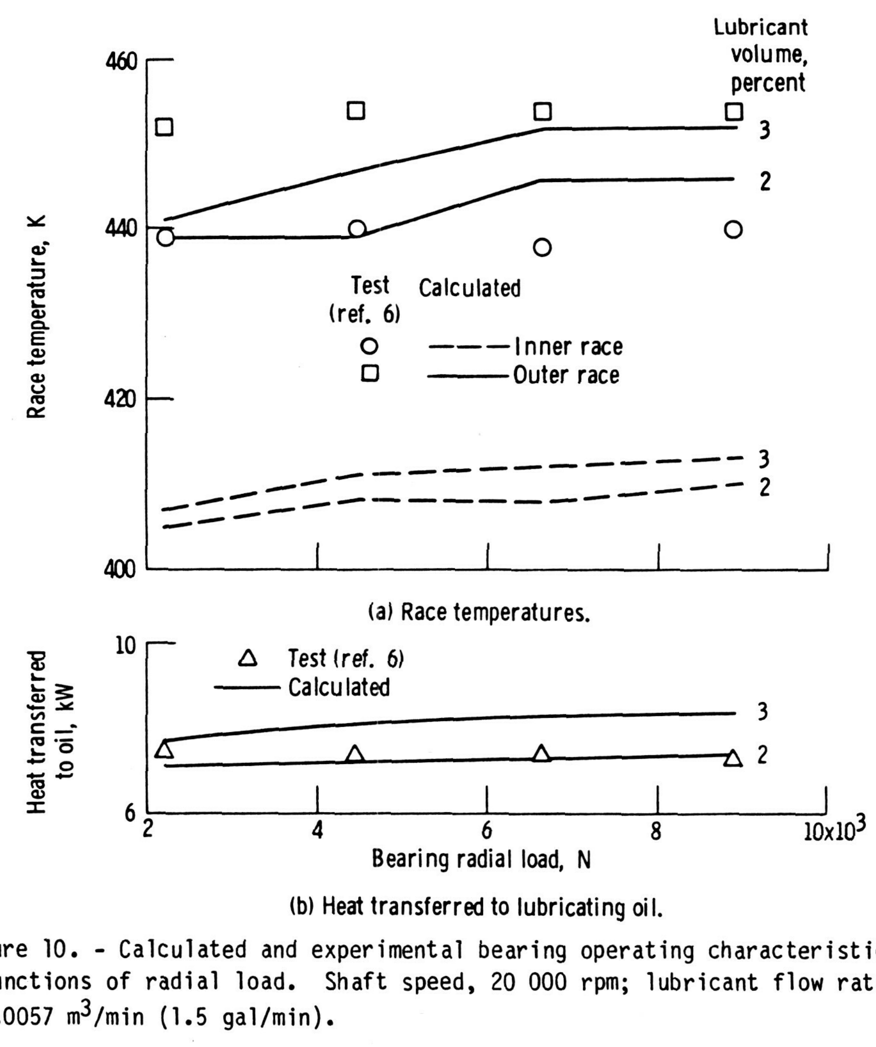
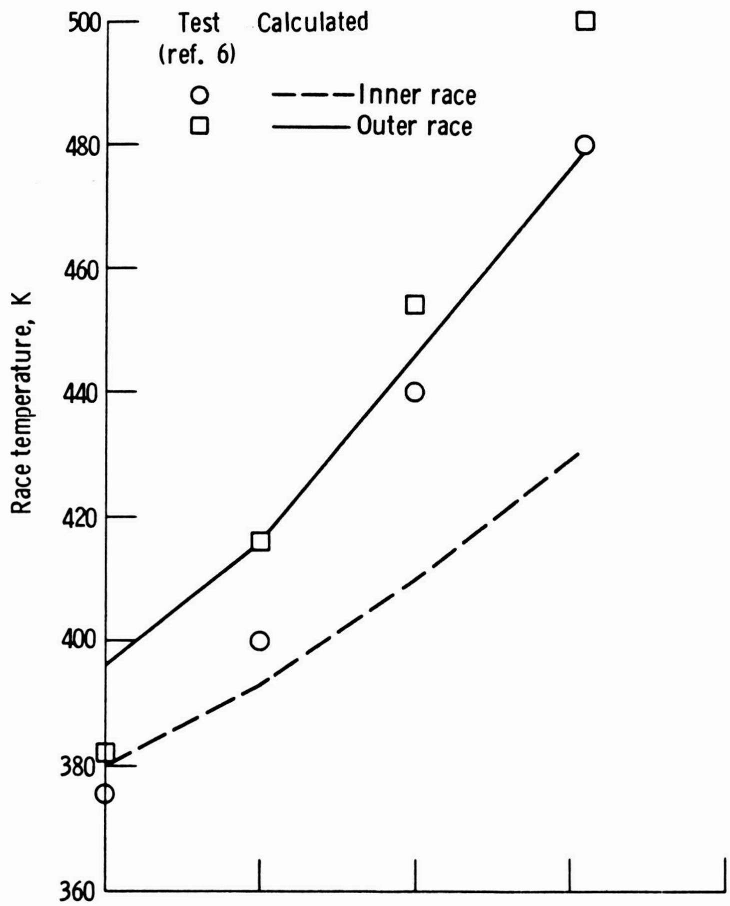


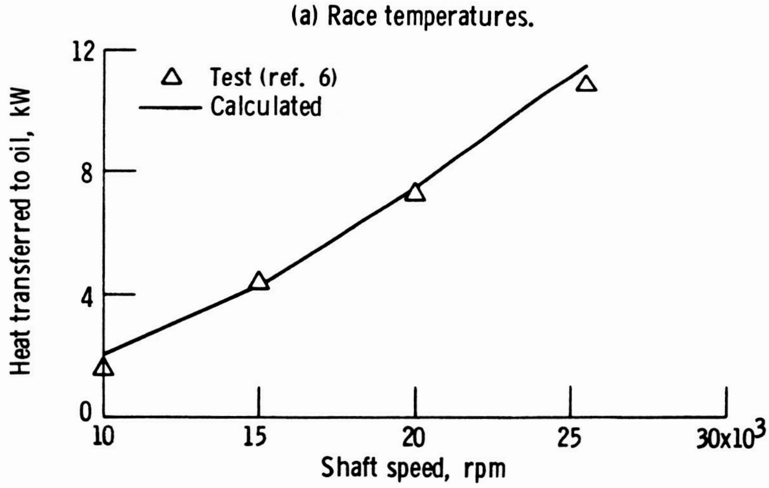
Figure 10. - Calculated and experimental bearing operating characteristics as functions of radial load. Shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 m^3/min (1.5 gal/min).

10 percent lower than the test values. The amount of heat transferred to the oil, predicted using the lubricant volume of 2 percent, compares very well with the test data (fig. 10(b)).

The effect of shaft speed was observed by using the program with an 8900-N (2000-lb) load for several values of shaft speed. The flow rate was set at 0.0057 m^3/min (1.5 gal/min), and the lubricant volume at 2 percent. The results are shown in figure 11 for shaft speeds from 10 000 to 25 500 rpm (1.2 to 3.0 million DN). The predicted values of outer-race temperature (fig. 11(a)) compared with



(a) Race temperatures.



(b) Heat transferred to lubricating oil.

Figure 11. - Calculated and experimental bearing operating characteristics as functions of shaft speed. Load, 8900 N (2000 lb); lubricant flow rate, 0.0057 m^3/min (1.5 gal/min); lubricant volume, 2 percent.

the experimental data are slightly high at the lower speeds and slightly low at the higher speeds. The predicted inner-race temperature, fairly close to the experimental data at the lower speeds, becomes very low at the higher speeds. The heat transferred to the oil, however, as predicted by the program compared very well with the experimental values over the whole speed range (fig. 11(b)). It appears that the calculations for the total bearing heat generation are correct but that insufficient heat transfer is predicted to the inner race at the higher speeds. At this point it is not clear whether the discrepancy with the inner-race temperature is a problem of using a proper thermal model or of using proper input data.

Effect of lubricant flow rate. – Since the oil flow rate can have a significant effect on bearing temperature and power loss, Cybean was run over a range of oil flow rates from 0.0038 to 0.0102 m^3/min (1.0 to 2.7 gal/min) for several values of lubricant volume percent. Calculations were made at 8900-N (2000-lb) load for 25 500 rpm. The results are shown in figure 12. The calculated trends are in the right direction; that is, the race temperatures are reduced by increasing the oil flow rate for the values of lubricant volume shown. The calculated outer-race temperatures (fig. 12(a)) are reasonable for volumes of 2 or 3 percent, while the calculated inner-race temperatures remain low over the entire

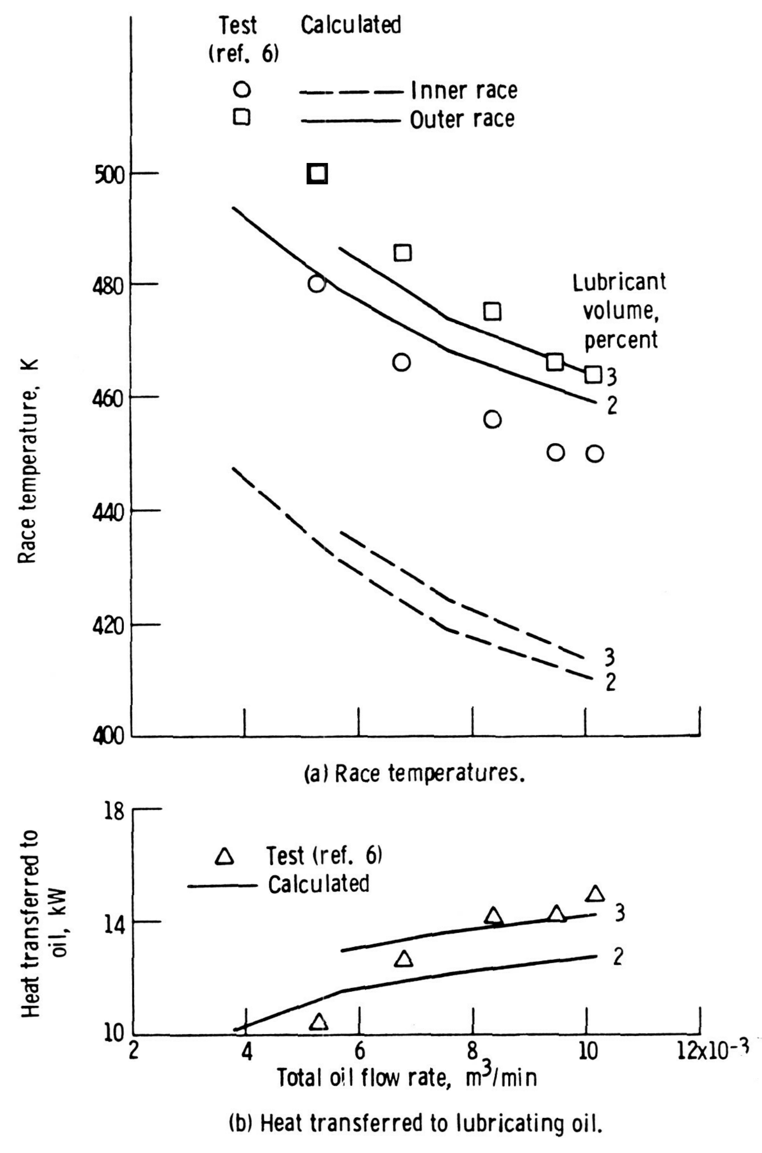


Figure 12. - Calculated and experimental bearing operating characteristics as functions of lubricant flow rate. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb).

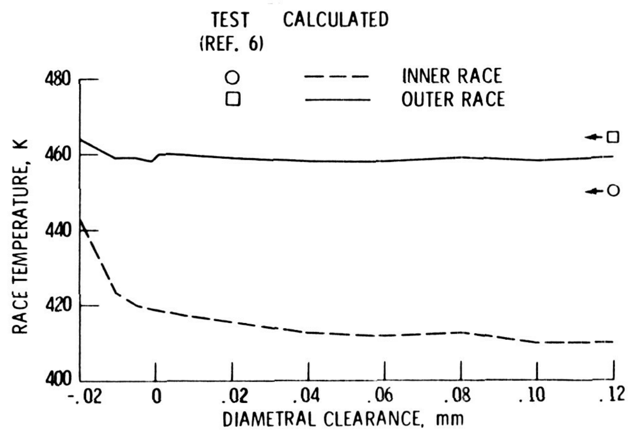


Figure 13. - Calculated race temperature as function of diametral clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent. Test values are shown for comparison, plotted at maximum possible clearance.

flow rate range. The computed heat transferred to the oil (fig. 12(b)) compares fairly well with the test data over the flow range for these two set values of lubricant volume.

From these results it is not clear how the lubricant volume should vary with flow rate. The outer-race temperature comparison would indicate that the volume percent should decrease with flow rate. A comparison of the heat transferred to the lubricant suggests that the lubricant volume should increase with flow rate. More work needs to be done in this area.

Effects of miscellaneous input data. – Several calculations were made in an effort to detect any errors in data input or thermal modeling that would explain why inner-race temperature predictions were low. The program was run at the 8900 N (2000 lb), 25 500-rpm condition with a flow rate of 0.0102 m^3/min (2.7 gal/min). The convergence criterion (used with an iterative procedure to determine when a solution has been reached; explained in detail in ref. 11) was changed from 0.1 to 0.05 and 0.01, and the calculated temperatures remained the same. A bearing misalinement of 5 minutes was assumed, and the race temperatures changed only 1 K. An additional 300-W heat generation was arbitrarily added to nodes 1 and 3 (shaft ends, see fig. 2) (in case proper accounting was not made for the support-bearing heat generation). While this managed to raise the inner-race temperatures 3 K, the temperature of nodes 1 and 3 became unacceptably high, 610 K (638° F). The nodal structure was changed slightly by adding four nodes, two on the shaft and two on the inner-ring adapter. The resulting calculated temperatures did not change. The value of the heat-transfer coefficient relating the rotating shaft and inner-ring adapter to the air in the rig cavity (e.g., from node 2 to node 21, fig. 2) was changed from 981 to 170 W/m^2 °C. The inner-race temperature changed 1 K. It was concluded that, since none of the above items had any significant effect on the bearing race temperatures, the inner-race temperatures were mostly affected by the bearing's internally generated heat.

Effect of diametral clearance. – One item that could have a large effect on the bearing heat generation is the diametral clearance, that is, the total free movement of the bearing components in a radial direction. Initial calculations using the original thermal nodes (fig. 2) showed only a small change in inner-race temperature from a clearance of 0.12 mm (maximum unmounted value) down to 0.001 mm. However, reference 6 suggests that a negative clearance exists at 25 500 rpm, so additional calculations were made using the revised nodal structure for several values of negative clearance. The results are shown in figure 13. The increase in inner-race temperature as the clearance is lowered below zero is dramatic and approaches the experimental value closely when the clearance is a minus 0.02 mm. At this point, and for the first time, all 28 rollers are loaded at the inner ring. At minus 0.01-mm clearance, 13 of the rollers were loaded at the inner race. This is possible because the actual operating clearance is slightly larger due to the contact deformations. These calculations indicate that it is very likely the bearing (ref. 6) was indeed operating with a negative clearance at 25 500 rpm.

For comparison purposes, the data of figure 12 were recalculated using a diametral clearance of minus 0.02 mm. These results are shown in figure 14. Both race temperatures (fig. 14(a)) and heat

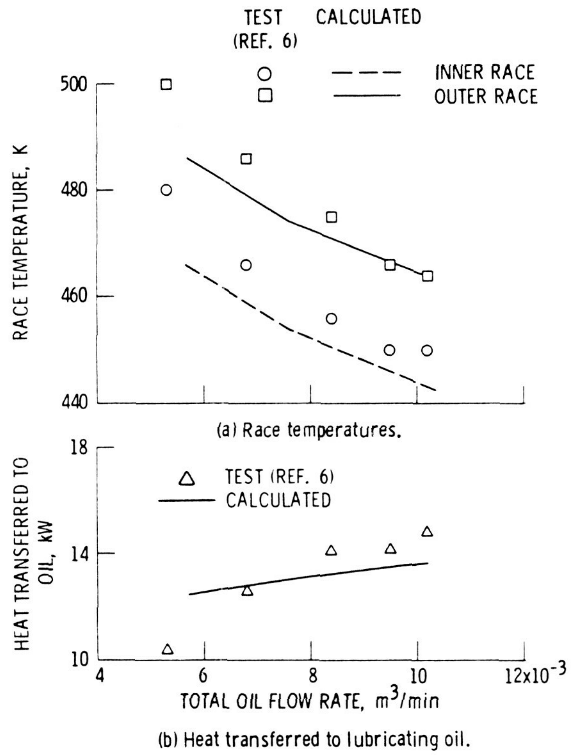


Figure 14. - Comparison of calculated and experimental bearing data using a diametral clearance of -0.02 mm in the computer program. Shaft speed, 25 500 rpm; lubricant volume, 2 percent.

transferred to the lubricant (fig. 14(b)) compare very well with the experimental data.

Although the computer program Cybean has the capability of evaluating bearings with out-of-round outer raceways, the program at this point did not predict the effective bearing operating clearance (i.e., the diametral clearance that would exist at operating speed and temperature). Since it became very evident that this was an important parameter for high-speed bearings, subroutines were added to Cybean to calculate changes in diametral clearance due to initial fits and due to temperature and high-speed effects. Further calculations were then made utilizing this capability.

The program Cybean was run for several values of bearing unmounted diametral clearance to determine the values of diametral clearance of the mounted bearing at operating speeds and temperatures calculated by the program. This clearance, which does not yet contain the effects of bearing load, will be called the effective hot, mounted diametral clearance. The bearing conditions used were 8900-N (2000-lb) load, 25 500-rpm shaft speed, 0.0102-m^3/min (2.7-gal/min) lubricant flow rate, and a lubricant volume of 2 percent. The initial shaft, inner-ring interference was set at the measured value of 0.0712 mm on the diameter. The results are shown in figure 15. With the actual measured value of 0.12 mm for cold, unmounted diametral clearance, the program predicted about 0.03 mm remaining as the effective hot, mounted clearance at operating conditions. To obtain a negative effective clearance, closer to the value of minus 0.02 mm noted previously, an input of only 0.09 mm initial unmounted clearance had to be used. Again, at this point, all 28 rollers are in contact with the inner race. It is interesting to note the large change in fatigue life of this bearing as the clearance gets smaller and the number of rollers in contact with the inner race gets larger. At first, the fatigue life increases and probably becomes a maximum at just that point where all 28 rollers are first in contact. The fatigue life then decreases rapidly as the tighter clearance causes increased stress.

To check program operation with the new clearance change subroutines, Cybean was run again for several flow rates, with an input of 0.09 mm cold, unmounted diametral clearance. The results at 25 500-rpm shaft speed were virtually the same as shown in figure 14 (i.e., the predicted temperatures were very close to the experimental values); however, the results with 20 000-rpm shaft speed again showed the calculated inner-race temperature almost 30 K low (i.e., similar to the results in fig. 12).

Cage slip and misalinement. – Two final checks were made by Cybean, using the original,

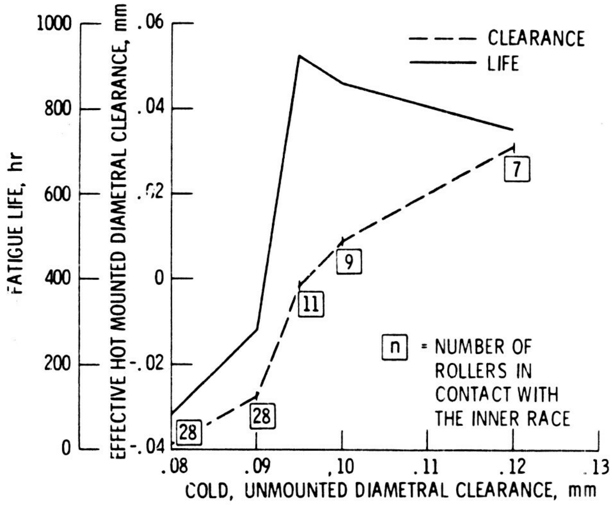


Figure 15. - Calculated values of effective hot, mounted clearance and fatigue life as functions of cold, unmounted clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); oil flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

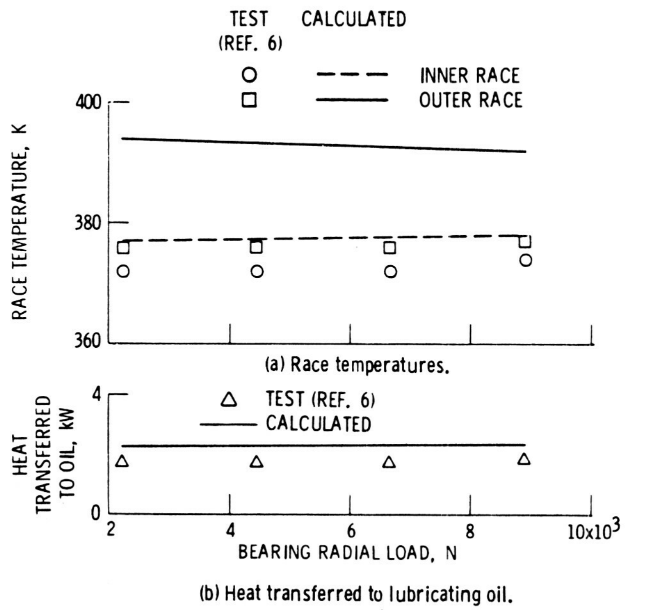


Figure 16. - Comparison of test data with values calculated using a cold diametral clearance of 0.12 mm in the computer program. Shaft speed 10 000 rpm; lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

measured, cold, unmounted diametral clearance of 0.12 mm. The first check was with several loads at the low-speed (10 000 rpm) condition. This low-speed condition was chosen because of the large values of cage slip indicated in reference 6. All previous calculated values of cage slip were less than 1 percent at 8900 N (2000 lb) and less than 3 percent at 2220 N (500 lb). The experimental values at the higher speeds (20 000 and 25 500 rpm) were all less than 2 percent. The flow rate was set at 0.0102 m^3/min (2.7 gal/min), and the lubricant volume at 2 percent. The results are shown in figure 16. Here, the inner-race temperature predictions are close to the experimental values, and the outer-race temperature predictions are about 20 K higher than the corresponding data. The heat transferred to the oil compares well (fig. 16(b)). The calculated values were only slightly higher than the test values.

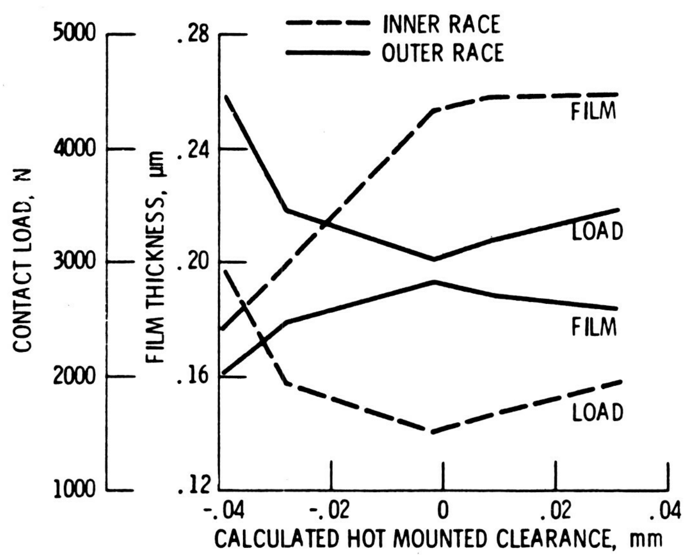


Figure 17. - Film thickness and contact load at most heavily loaded roller as function of calculated hot, mounted clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

However, while the tests indicated cage slips of over 46 percent, for the entire load range from 2220 to 8900 N (500 to 2000 lb), the corresponding predicted values were all less than 3 percent. Without this slip, the experimental temperatures would have been somewhat higher (see ref. 6, where the bearing temperatures varied inversely with cage slip). The calculated effective hot, mounted diametral clearance was about 0.08 mm.

The second check was to run the program at the high-speed condition (25 500 rpm) with 8900 N (2000 lb) load, set the misalinement angle to 5 minutes, and see if the resulting skew would be sufficient to change the predicted race temperatures significantly. This calculation showed the inner-race temperature to be only 8 K higher with skewing and the outer-race temperature to be 3 K higher. The heat transferred to the lubricant was the same. The calculated effective hot, mounted clearance was about 0.03 mm. Since the 5-minute misalinement angle is large for this test rig, it may be concluded that the presence of misalinement alone would not have been sufficient to cause the experimental inner-race temperatures to be so much higher than calculated values. Since the effect of misalinement on bearing temperatures was small, and the amount of computer time increased by a factor of 10, no further calculations were made with misalinement.

Roller loads, film thickness, and skew angle. – While the main focus of the work reported herein was to calculate bearing characteristics that were measured experimentally, the program Cybean does provide calculated values of other items of interest. Some of these are the EHD film thickness at the roller-raceway contact, the individual roller-race contact loads and stresses, the roller-cage forces, and, with misalinement, the roller-flange forces and the roller skew and tilt angles. Several of these items are shown in figures 17 to 21. The calculations were performed for a shaft speed of 25 500 rpm, a radial load of 8900 N (2000 lb), a flow rate of 0.0102 m^3/min (2.7 gal/min), and a lubricant volume of 2 percent.

The values of the EHD film thickness for the most heavily loaded roller contact, calculated for those conditions of figure 15, are shown in figure 17. The film thickness is plotted as a function of the calculated effective hot, mounted clearance. Also shown is the corresponding roller contact load. In general, the film thickness diminishes, and the contact loads increase rapidly once the bearing clearance becomes negative, and all the rollers are in contact with the inner ring (see fig. 15). Other calculations with a fixed diametral clearance as input for those same bearing operating conditions (see fig. 13) indicate that the film thickness changes very little as the clearance increases from 0 to 0.12 mm. The corresponding outer-ring contact load continues to increase slightly and reaches 4000 N at 0.12-mm clearance. Likewise, the inner-ring load increases to 2600 N at 0.12-mm clearance.

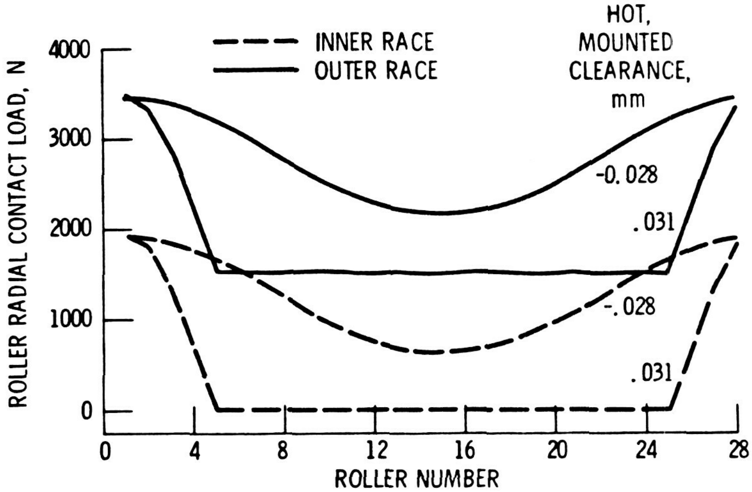


Figure 18 - Roller-race contact load variation with roller position for two values of clearance. Applied load at number 1 roller position. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

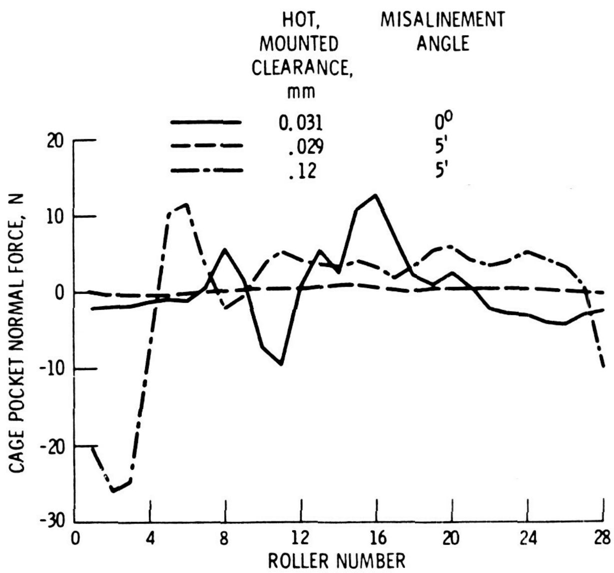


Figure 19. - Cage pocket forces as function of roller number. Positive force is the cage pushing the roller. Bearing load at roller 1 position. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

The inner- and outer-ring contact loads for each roller, calculated for these same conditions, are shown in figure 18 for two values of hot, mounted clearance. For the 0.031-mm clearance bearing, there are seven rollers in contact with the inner ring, and the remaining rollers have only the centrifugal loading at the outer ring. When all 28 rollers are in contact at the clearance of −0.028 mm, the maximum load is about the same. However, all the other rollers are carrying heavier loads at both the inner and outer rings than with the positive clearance. When the clearance was at −0.039 mm, the contact loads for each of the 28 rollers increased by 1000 N at both the inner and outer rings. The calculation for these same operatingconditions with a 5-minute mislinement, where the hot mounted clearance was 0.029 mm, showed little change from the contact loads of the 0.031-mm clearance curve.

The program estimates of the roller-cage forces are shown in figure 19 for three combinations of

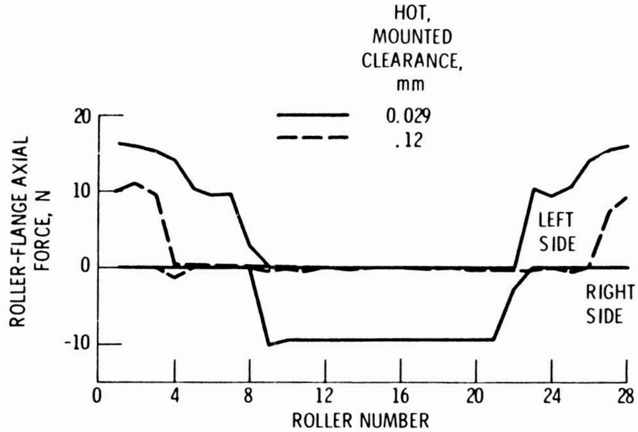


Figure 20. - Roller-flange forces as function of roller number. Bearing load at roller 1 position. Misalinement angle, 5 minutes; shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

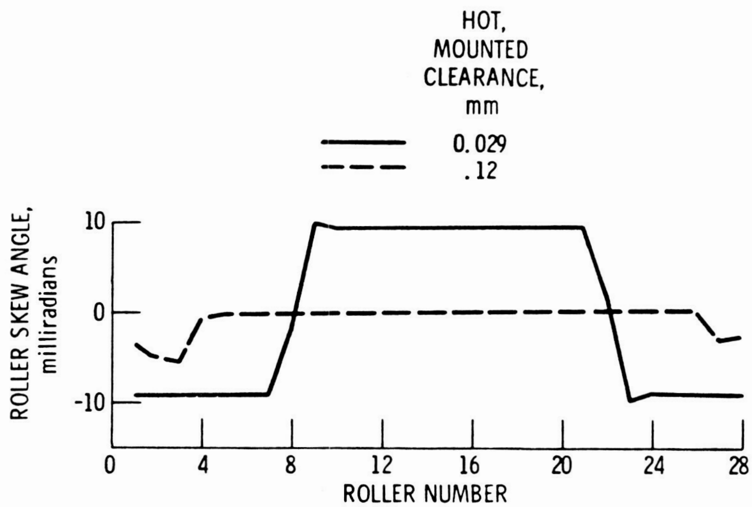


Figure 21. - Roller relative skew angle as function of roller position. Bearing load at roller 1 position. Misalinement angle, 5 minutes; shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant flow rate, 0.0102 m^3/min (2.7 gal/min); lubricant volume, 2 percent.

hot, mounted clearance and misalinement angle. A positive force means the cage is pushing the roller. With no misalinement the maximum cage forces were about 10 N. For the same clearance, with a 5-minute misalinement, the cage forces seemed to be all very small. With a large clearance of 0.12 mm, however, and a 5-minute misalinement, the maximum cage force reached 25 N. These forces are of the same order of magnitude as those measured experimentally for the same size bearing. (See refs. 21 and 22. The sign convention in refs. 21 and 22 is of the opposite sense to that used here.)

The flange axial forces calculated for the same two cases with the 5-minute misalinement are shown in figure 20. These forces are small, about the same order of magnitude as the cage-roller forces. The larger clearance bearing shows less roller-flange contact.

The predicted roller skew angle for these same two cases is shown in figure 21. In both cases there is a negative skew angle at the load zone. For the 0.029-mm clearance case, the remainder of the rollers exhibits a positive skew angle of about the same size. With the large 0.12-mm clearance, however, the remaining rollers show essentially zero skew. The skew angle shown is the angle of the roller relative to the inner ring and is called the relative skew angle. The angle of the roller relative to the outer ring, called the absolute skew angle, would be the relative angle plus (or minus) the amount of the misalinement present at that particular roller position. These skew angles are also of the same order of magnitude as those measured experimentally for the same size bearing and reported in reference 23.

Concluding Remarks

The Shaberth thermal predictions were fairly close even though the program capabilities were not fully utilized. For example, since the calculated bearing-race temperatures were fairly close to the experimental data using the small thermal grid, we speculate that the temperatures could be predicted more accurately if a larger or more complex thermal grid system were used. Also, in the present calculations a constant coefficient of convective heat transfer, calculated as suggested in reference 8, was used. This coefficient could be calculated in the program as a function of the lubricant viscosity for a closer approximation. Furthermore, the lubricant flow rate was not used directly in the thermal calculations. There were no temperature nodes in the fluid, other than oil inlet to the bearing and oil outlet from the bearing. This would have some influence on the temperature calculations, as noted previously. Introduction of lubricant flow rate could permit the race temperatures to decrease with flow rate and still have increasing power loss.

In general, the Cybean computer program as used predicted values of outer-race temperature and heat transferred to the oil that compared reasonably well with the corresponding experimental data. However, the calculated values of inner-race temperature were usually somewhat low, especially at the higher shaft speeds. The program did not predict the high cage slip experienced (ref. 6) at the lower shaft speeds. This is probably the reason the experimental temperatures were lower than the calculated values for those low-speed conditions. Nevertheless, it should be noted that on the basis of absolute temperatures, all calculated values were within 10 percent of the corresponding experimental data and most were within 5 percent. Considering the nature of heat-transfer calculations, this is reasonably close correlation. In all cases outer-race temperature was predicted to be greater than inner-race temperature, which agreed with all experimental data. The predicted trends of changing bearing temperature and power loss with shaft speed, load, and lubricant flow rate compared well with experimental data.

The calculations performed also show the importance of the effective hot, mounted diametral clearance for useful bearing operation. Care should be taken that the bearing effective diametral clearance remains positive at all operating conditions to assure a reasonable rolling-element fatigue life for the bearing.

The largest unknown quantity of the input data required for both Shaberth and Cybean is the volume percent of lubricant in the bearing cavity. The values chosen for these calculations were in the range recommended by the authors of references 8 and 11. From the comparisons presented in this paper, it can be concluded that the values of percent lubricant used are reasonably correct for these programs. However, since it is still not clear how these values should vary with oil flow rate and/or shaft speed, work needs to be done in this area.

Summary of Results

Computer programs (Shaberth for ball bearings, Cybean for roller bearings) were used to predict bearing inner- and outer-race temperatures, cage speed, and friction power loss or heat transferred to the lubricant over a range of operating conditions. The results were compared with experimental data obtained previously. The 120-mm-bore ball bearings were operated at thrust loads of 6672, 13 350, and 22 240 N (1500, 3000, and 5000 lb) and at shaft speeds of 12 000, 16 700, and 20 800 rpm with jet-lubrication flow rates of 0.0038 and 0.0083 m^3/min (1.0 and 2.2 gal/min). The oil inlet temperature was maintained constant at 394 K (250° F).

The 118-mm-bore, cylindrical roller bearings were operated at radial loads of 2220, 4450, 6670, and 8900 N (500, 1000, 1500, and 2000 lb) and at shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The bearings were lubricated and cooled by flowing oil through and under the inner race at total rates of 0.0038 to 0.0102 m^3/min (1.0 to 2.7 gal/min). The inlet temperature was maintained constant at 366 K (200° F). The following results were obtained:

1. The bearing-shaft program (Shaberth) can predict race temperatures and bearing power loss reasonably well.

2. The cylindrical roller bearing analysis computer program (Cybean) can predict outer-race temperature and the amount of heat transferred to the lubricant reasonably well; however, at the higher shaft speeds, the calculated inner-race temperatures were much lower than the corresponding experimental data, unless the effective hot, diametral clearances were set negative, about 0.02 mm.

3. Cybean did not predict the high cage slip experimentally obtained with the roller bearing at low shaft speeds; however, the computer-estimated values of roller-cage forces were of the same order of magnitude as that obtained experimentally for the same size bearing.

4. The bearing power losses predicted by the computer programs were a strong function of the value assumed for the volume percent of the bearing cavity occupied by the lubricant.

References

1. Brown, P. F.; Dobek, L. J.; and Tobiasz, E. J.: High Speed Cylindrical Roller Bearing Development. PWA-FR-12598, Pratt & Whitney Aircraft Group, 1980. (AFWAL-TR-80-2072, AD-A095357.)
2. Povinelli, V. P., Jr.: Current Seal Designs and Future Requirements for Turbine Engine Seals and Bearings. J. Aircr., vol. 12, no. 4, Apr. 1975, pp. 266-273.
3. Brown, P. F.; et al.: Mainshaft High Speed Cylindrical Roller Bearings for Gas Turbine Engines. PWA-FR-8615, Pratt & Whitney Aircraft Group, 1977. (AD-A052351.)
4. Holmes, P. W.: Evaluation of Drilled-Ball Bearings at DN Values to Three Million. I—Variable Oil Flow Tests. NASA CR-2004, 1972.
5. Signer, H.; Bamberger, E. N.; and Zaretsky, E. V.: Parametric Study of the Lubrication of Thrust Loaded 120-mm Bore Ball Bearings to 3 Million DN. J. Lubr. Technol., vol. 96, no. 3, July 1974, pp. 515-524.
6. Schuller, F. T.: Operating Characteristics of a Large Bore Roller Bearing to Speeds of 3×10^6 DN. NASA TP-1413, 1979.
7. Kennedy, F. E.; and Cheng, H. S., eds.: Computer-Aided Design of Bearings and Seals. American Society of Mechanical Engineers, 1976.
8. Crecelius, W. J.; Heller, S.; and Chiu, Y. P.: Improved Flexible Shaft-Bearing Thermal Analysis with NASA Friction Models and Cage Effects. SKF-AL76P003, SKF Industries, Inc., 1976.
9. Crecelius, W. J.; and Pirvics, J.: Computer Program Operation Manual on SHABERTH: A Computer Program for the Analysis of the Steady State and Transient Thermal Performance of Shaft-Bearing Systems. SKF-AL76P030, SKF Industries, Inc., 1976. (AFAPL-TR-76-90, AD-A042981.)
10. Kleckner, R. J.; and Pirvics, J.: High Speed Cylindrical Roller Bearing Analysis. SKF Computer Program CYBEAN. Volume I: Analysis. (AL78P022-VOL-1, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159460, 1978.
11. Kleckner, R. J.; and Pirvics, J.: High Speed Cylinder Roller Bearing Analysis. SKF Computer Program CYBEAN: Volume II: User's Manual. (AL78P023-VOL-2, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159461, 1978.
12. Kleckner, R. J.; Pirvics, J.; and Castelli, V.: High Speed Cylindrical Rolling Element Bearing Analysis "CYBEAN"—Analytic Formulation. ASME Paper 70-LUB-35, Oct. 1979.
13. Gupta, P. K.: Dynamics of Rolling Element Bearings, Parts I, II, III, and IV. J. Lubr. Technol., vol. 101, no. 3, July 1979, pp. 293-326.
14. Conry, T. F.; and Goglia, P. R.: Transient Dynamic Analysis of High-Speed Lightly Loaded Cylindrical Roller Bearings. Part II, Computer Program and Results. NASA CR-3335, 1981.
15. Zaretsky, E. V.; Signer, H.; and Bamberger, E. N.: Operating Limitations of High-Speed Jet-Lubricated Ball Bearings. J. Lubr. Technol., vol. 98, no. 1, Jan. 1976, pp. 32-39.
16. Coe, H. H.; and Zaretsky, E. V.: Predicted and Experimental Performance of Jet-Lubricated 120-Millimeter-Bore Ball Bearings Operating to 2.5 Million DN. NASA TP-1196, 1978.
17. Coe, H. H.; and Schuller, F. T.: Comparison of predicted and Experimental Performance of Large-Bore Roller Bearing Operating to 3.0 Million DN. NASA TP-1599, 1980.
18. Coe, H. H.; and Schuller, F. T.: Calculated and Experimental Data for a 118-mm-Bore Roller Bearing to 3 Million DN. ASME Paper 80-C2/Lub-14, Aug. 1980.
19. Bamberger, E. N.; Zaretsky, E. V.; and Anderson, W. J.: Effect of Three Advanced Lubricants on High Temperature Bearing Life. J. Lubr. Technol., vol. 92, no. 1, Jan. 1970, pp. 23-33.
20. Loewenthal, Stuart H.; Parker, Richard J.; and Zaretsky, Erwin V.: Correlation of Elastohydrodynamic Film Thickness Measurements for Fluorocarbon, Type II Ester, and Polyphenyl Ether Lubricants. NASA TN D-7825, 1974.
21. Nypan, L. J.: Roller to Separator Contact Forces and Cage to Shaft Speed Ratios in Roller Bearings. NASA CR-3048, 1978.
22. Nypan, L. J.: Separator Contact Forces and Speed Ratios in Roller Bearings. ASLE Preprint 79-LC-4D-1, Oct. 1979.
23. Nypan, L. J.: Roller Skewing Measurements in Cylindrical Roller Bearings. NASA CR-3381, 1981.