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CALCULATED AND EXPERIMENTAL DATA FOR A 118-mm BORE ROLLER BEARING TO 3 MILLION DN

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ABSTRACT

Operating characteristics for a 118-mm bore cylindrical roller bearing were calculated using the computer program CYBEAN. The predicted results of innerand outer-race temperatures and heat transferred to the lubricant generally compared well with experimental data for shaft speeds to 3 million DN (25 500 rpm), radial loads to 8900 N (2000 lb) and total lubricant flow rates to 0.0102 m^3/min (2.7 gal/min).

INTRODUCTION

For the last several years, 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 [1-3] (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 in recent years. Successful operation of ball bearings at 3 million DN was reported [4,5]. Roller bearings were operated to 3 million DN [3,6].

The question of how to design bearings for high-speed applications is increasingly being answered by computer studies [3,7]. There are currently several comprehensive computer programs in use that are capable of predicting rolling bearing operating and performance characteristics (e.g., [8-13]). 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 speeds, heat generation, local temperatures, bearing fatigue life, and power loss. In [14], data were published which compared computer predictions with actual ball bearing performance. However, little data has been published which compare computer predictions with actual cylindrical roller bearing performance.

As reported in [8-10], a computer program called CYBEAN has recently been developed for analysis of high-speed cylindrical roller bearings. It is therefore the objective of the work reported herein to compare the values of inner- and outer-race temperatures, cage speed, and heat transferred to the lubricant calculated using the computer program CYBEAN, with the corresponding experimental data for the 118-millimeter bore bearing described in [6].

BEARING TEST DATA

The experimental data used for comparison purposes in this report were initially reported in [6]. In this reference, a large bore roller bearing was tested at speeds up to 3 million DN, loads up to 8900 N (2000 lb) and with total oil flow rates up to 0.0102 cubic meters per minute (2.7 gal/min). 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 did not have any radial holes. About one-half of the total oil introduced to the inner ring was used for cooling only.

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 1. The bearing tester is described in detail in [6].

The test bearing was a roller bearing with a 118-millimeter bore, a flanged inner ring, and 28 rollers, each 12.65 millim ters (0.4979 in.) diameter by 14.56 millimeters (0.573 in.) long. More complete specifications are shown in Table 2.

Oil inlet temperature was held constant at 366 K (200^{o} 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 cubic meter per minute (0.5 to 2.7 gal/min).

THE COMPUTER PROGRAM

The comprehensive computer program CYBEAN, for analysis of a single cylindrical roller bearing, is completely described in [8-10]. This program is capable of calculating the thermal and kinematic performance of high-speed bearings and includes a roller skew prediction for misaligned conditions. The calculations include determination of inner- and outer-ring temperatures, oil outlet temperatures, cage speed, and bearing power loss.

Use of 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 [9] recommended that the values used be less than 5 percent.

When CYBEAN is also 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. For this investigation, 41 nodes were used, including 19 in the lubricant system, as shown in Fig. 1(a). Later four metal nodes were added (causing the two known-temperature nodes to be renumbered) as shown in Fig. 1(b).

RESULTS AND DISCUSSION

To effect the direct comparison of predicted and experimental bearing performance, the computer program was generally run at the stated operating conditions of the bearing tested [6]. All combinations of load, speed and flow rate were not computed, because they are not all necessary for comparison purposes and would require unnecessary computer time. Therefore, 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 two speeds (8900 N (2000 lb)) and (20 000 and 25 500 rpm).

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-cubic meter per minute (1.5 gal/min). The results are shown in Fig. 2. Note that in all the figures in this report, the calculated values are always just connected with straight line segments. Also shown are the corresponding experimental data points.

The race temperatures (Fig. 2(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 of 5 percent oil volume, the temperature changes seem to be linear and not too large, about 5 percent at these conditions.

The total heat generated in the bearing (Fig. 2(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 Ref. 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 Fig. 2(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 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 cubic meter per minute (1.5 gal/min) for two lubricant volumes (2 and 3%), and a shaft speed of 20 000 rpm. The results, compared with experimental data, are shown in Fig. 3 for radial loads from 2220 to 8900 N (500 to 2000 lb). The predicted race temperatures (Fig.

3(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 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. 3(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 cubic meters per minute (1.5 gal/min) and the lubricant volume at 2 percent. The results are shown in Fig. 4 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. 4(a)) compared to 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. 4(b)). It is therefore apparent that the calculations for the total bearing heat generation seem to be 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, the computer program was run over a range of oil flow rates from 0.0038 to 0.0102 cubic meter per minute (1.0 to 2.7 gal/min) for several values of lubricant volume percent. Calculations were made at 8900 N (2000 lb) load for both 20 000 and 25 500 rpm. The results are shown in Figs. 5 and 6. 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 for 20 000 rpm (Fig. 5(a)) are reasonable for volumes of 2 or 4 percent, while the calculated inner-race temperatures remain low over the entire flow rate range. The computed heat transferred to the oil (Fig. 5(b)) at 20 000 rpm compares fairly well with the test data over the flow range for these two set values of lubricant volume. The comparisons at 25 500 rpm (Fig. 6) are very similar to those of 20 000 rpm (Fig. 5). From these figures, 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 modelling. The program was run at the 8900 N (2000 lb), 25 500 rpm condition with a flow rate of 0.0102 cubic meters per minute (2.7 gal/min). The convergence criterion (used with an iterative procedure to determine when a solution has been reached-explained in detail in [9]) was changed from 0.1 to 0.05 and 0.01 and the calculated temperatures remained the same. A bearing misalignment of 5 minutes was assumed, and the race temperatures changed only 1 Kelvin. An additional 300 watts heat generation was arbitrarily added to nodes 1 and 3 (shaft ends, see Fig. 1(a)) (In case the support bearing heat generation was not properly accounted for). While this managed to raise the inner-race temperatures 3 Kelvin, 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 (as shown in Fig. 1(b)). 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 41 to node 21, Fig. 1(b)) was changed from 981 to 170 watts per square meter-degree C. The inner-race temperature change 1 Kelvin. 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. 1(a)) showed only a small change in inner race temperature from a clearance of 0.12 millimeter (maximum unmounted value) down to 0.001 millimeter. However, [6] suggests that a negative clearance exists at 25 500 rpm, so additional calculations were made, using the nodal structure of Fig. 1(b), for several values of negative clearance. The results are shown in Fig. 7. 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 millimeter. At this point, and for the first time, all 28 rollers are loaded at the inner ring. At minus 0.01

millimeter clearance, 13 of the rollers were loaded at the inner race. These calculations indicate that it is very likely the bearing [6] was indeed operating with a negative clearance at 25 500 rpm.

For comparison purposes, the data of Fig. 6 were recalculated using a diametral clearance of minus 0.02 millimeter. These results are shown in Fig. 8. Both race temperatures (Fig. 8(a)) and heat transferred to the lubricant (Fig. 8(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 cubic meter per minute (2.7 gal/min) lubricant flow rate, and a lubricant volume of 2 percent. The initial shaft-inner ring interference was set at 0.0712 millimeter on the diameter. The results are shown in Fig. 9. With the actual measured value of 0.12 millimeter for cold, unmounted diametral clearance, the program predicted about 0.03 millimeter remaining as the effective hot mounted clearance at operating conditions. To obtain a negative effective clearance, closer to the value of minus 0.02 millimeter noted previously, an input of only 0.09 millimeter 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 predictions at other conditions, CYBEAN was operated with an input of 0.09 millimeter cold, unmounted diametral clearance for several values of lubricant flow rate. The first calculations were for the 8900 N (2000 lb) 25 500 rpm condition, while the second calculations were for the 8900 N (2000 lb), 20 000 rpm condition. The lubricant volume was 2 percent. The results are shown in Figs. 10 and 11. In Fig. 10, the program predicted race temperature for 25 500 rpm shaft speed that had the correct trend with lubricant flow rate and were exceptionally close to the experimental values. The heat transferred to the oil (Fig. 10(b)) also compared well with the experimental data, although the value at the low flow rate is almost two kW low. In Fig. 11, however, while the predicted trends were correct for 20 000 rpm, the calculated inner-race temperatures were almost 30 Kelvin low. The outer-race temperatures compared fairly well. The heat transferred to the oil (Fig. 11(b)) was also reasonably close, although at the highest flow rates the calculated values were about 2 kW low.

Two final checks were made with CYBEAN, using the original cold unmounted diametral clearance of 0.12 millimeter. 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 [6]. All previous calculated values of cage slip were less than 1 percent at 8900 N (2000 lb) and up to 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 cubic meter per minute (2.7 gal/min) and the lubricant volume at 2 percent. The results are shown in Fig. 12. Here, the inner-race temperature predictions are close to the experimental values, and the outer-race temperature predictions are about 20 Kelvins higher than the corresponding data. The calculated heat transferred to the oil compares well (Fig. 12(b)), and were slightly higher than the test values. 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, judging from [6], where the bearing temperatures varied inversely with cage slip. The calculated effective hot mounted diametral clearance was about 0.08 millimeter.

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 Kelvin higher with skewing and the outer-race temperature was 3 Kelvin higher. The heat transferred to the lubricant was the same. The calculated effective hot mounted clearance was about 0.03 millimeter. Since the 5 minute misalignment 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 Figs. 13 to 17. 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, and a lube volume of 2 percent.

The values of the EHD film thickness for the most heavily loaded roller contact, calculated for those conditions of Fig. 9 are shown in Fig. 13. 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. 9). Other calculations with a fixed diametral clearance as input for those same bearing operating conditions (see Fig. 7) indicate that the film thickness changes very little as the clearance increases from zero 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.

The inner- and outer-ring contact loads for each roller, calculated for these same conditions are shown in Fig. 14 for two values of hot mounted clearance. For the 0.031 mm clearance bearing, there are 7 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 a -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 operating conditions with a 5 minute misalinement, 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 Fig. 15 for three combinations of 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 [15,16] (The sign convention in [15,16] 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 Fig. 16. 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 Fig. 17. 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 exhibit 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.

CONCLUDING REMARKS

In general, the CYBEAN computer program as utilized 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 [6] at the lower shaft speeds. This is probably the reason the experimental temperatures were lower than the calculated values for those 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.

The calculations performed 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 obtaining a reasonable rolling-element fatigue life for the bearing.

As for the ball bearing computer program SHABERTH [12], the largest unknown quantity of the input data required for CYBEAN is the volume percent of lubricant in the bearing cavity. The values chosen for these calculations were in the range recommended [9]. From the comparisons presented in this paper, it can be conluded that the values of percent lubricant used are reasonably correct for this program. However, just how these values should vary with oil flow rate and/or shaft speed is still not clear.

SUMMARY OF RESULTS

The computer program CYBEAN was used to predict inner- and outer-race temperatures, cage speed, and heat transferred to the lubricant for a 118-millimeter bore cylindrical roller bearing. The results, calculated over a range of operating conditions, were compared with experimental data obtained previously. The 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 cubic meter per minute (1.0 to 2.7 gal/min). The oil inlet temperature was maintained constant at 366 K (200° F). The following results were obtained:

1. The cylindrical roller bearing analysis computer program (CYBEAN) can predict outer-race temperature and the amount of heat transferred to the lubricant reasonably well.

2. 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 millimeter.

3. A bearing can operate with an effective hot mounted diametral clearance of zero or less, but the calculated fatigue life decreases rapidly once all of the rollers are in contact with the inner race.

4. The computer program did not predict the high cage slip experimentally obtained with the roller bearing at low shaft speeds.

5. The computer estimated values of roller-cage forces were of the same order of magnitude as that obtained experimentally for the same size bearing.

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TABLE 1. - PROPERTIES OF TETRAESTER LUBRICANT [5]

Additives	Antiwear, oxidation inhibitor, and 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

TABLE 2. - ROLLER-BEARING SPECIFICATIONS

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)

TABLE I. - PROPERTIES OF TETRAESTER LUBRICANT[a]

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Thermal conductivity at 477 K (400° F), J/m sec K; Btu/hr ft °F	0.13 (0.075)
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[a]From reference 5.

TABLE II. - 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)
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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)

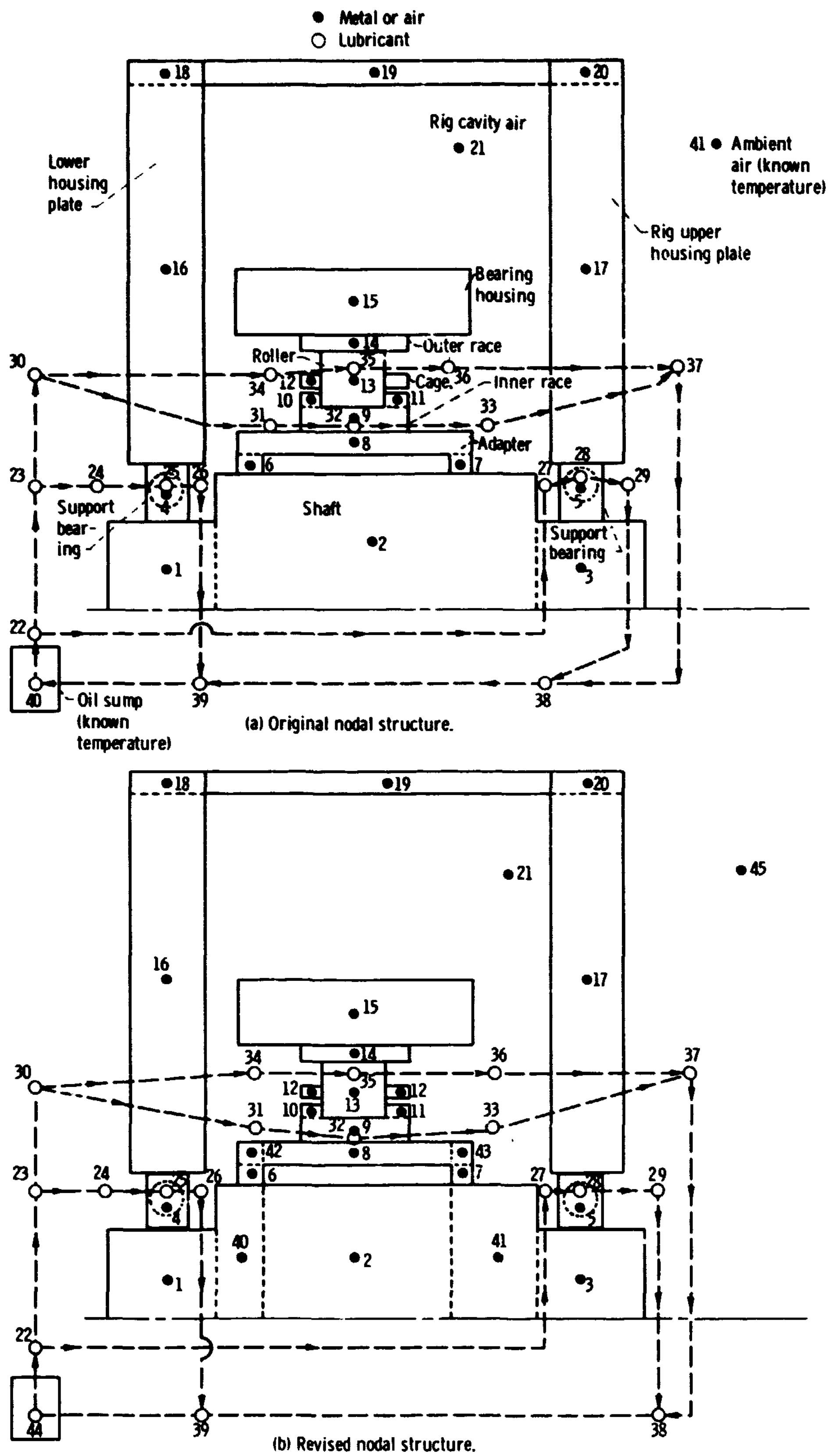
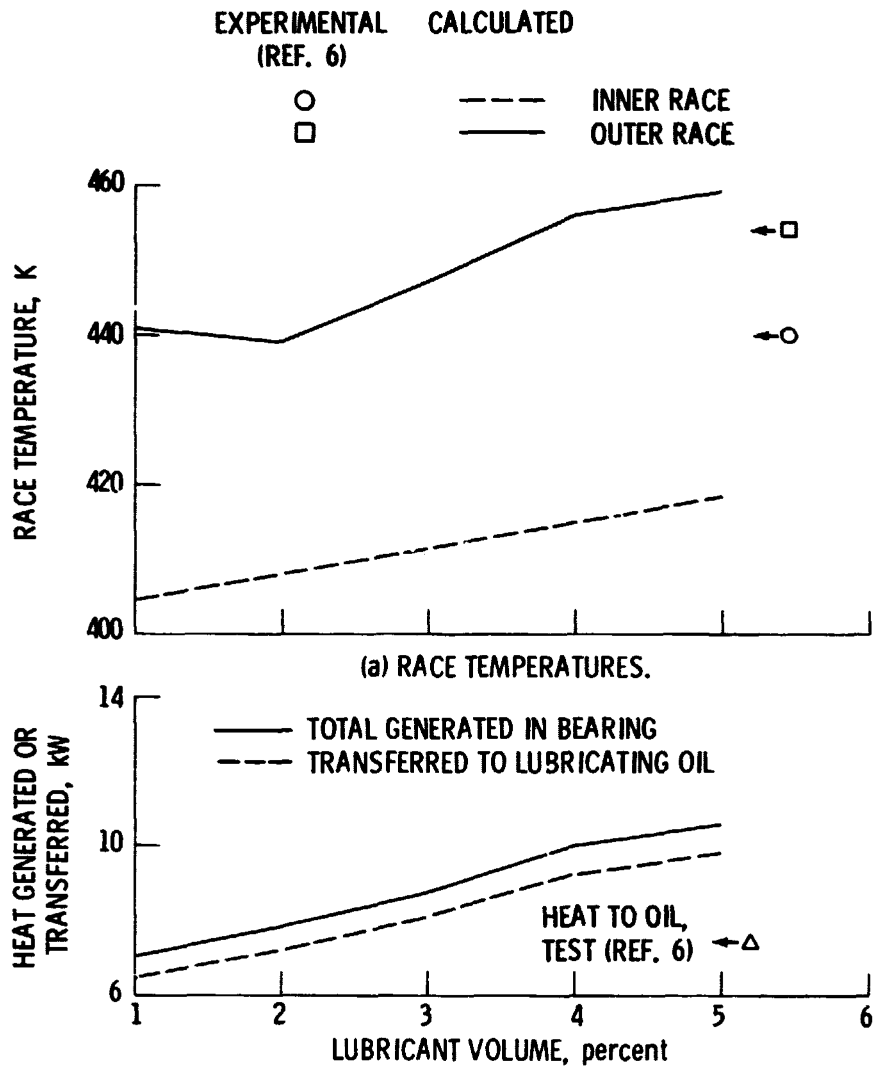
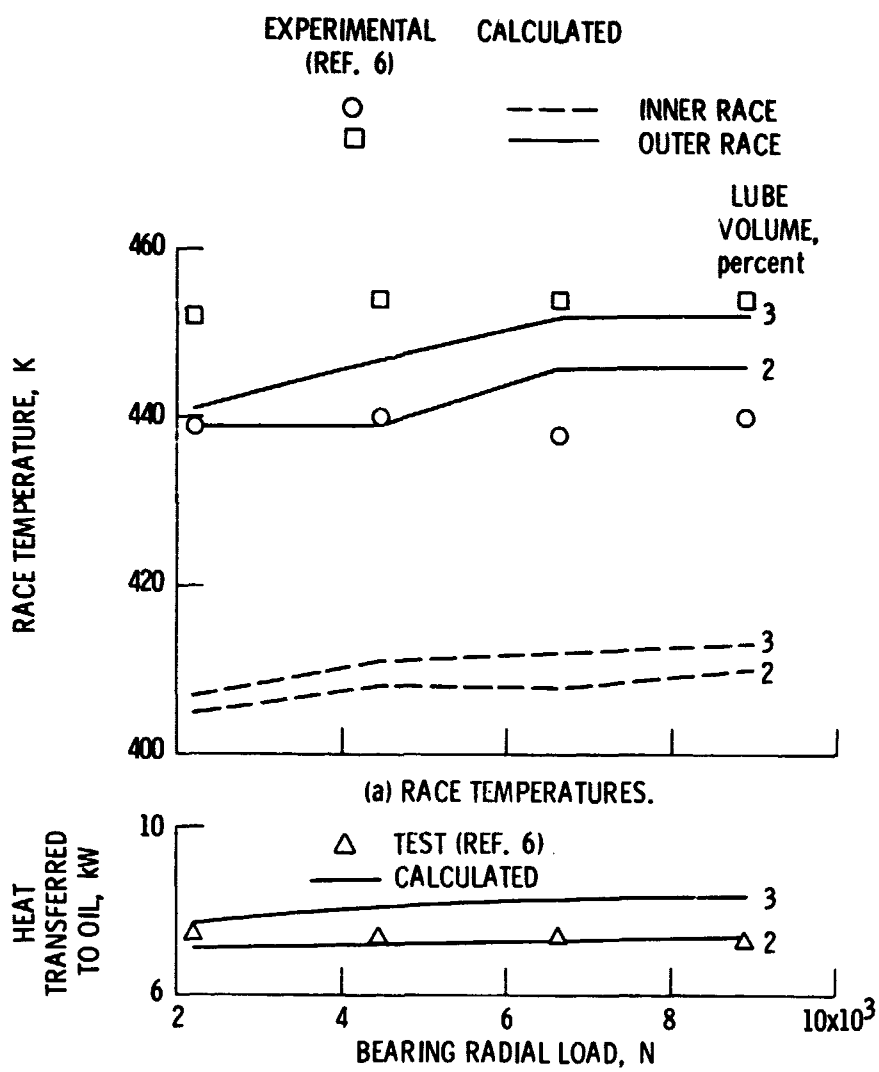


Figure 1. - Nodal system used for thermal routines in CYBEAN.



(b) HEAT GENERATED IN BEARING, OR TRANSFERRED TO LUBRICANT.

Figure 2. - Calculated values of bearing operating characteristics as functions of lubricant volume fraction. Test data shown for comparison. Load, 4450 N (1000 lb); shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 cubic meter per minute (1.5 gal/min).



(b) HEAT TRANSFERRED TO LUBRICATING OIL.

Figure 3. - Comparison of calculated and experimental values of bearing operating characteristics as functions of radial load. Shaft speed, 20 000 rpm; lubricant flow rate, 0.0057 cubic meter per minute (1.5 gal/min).

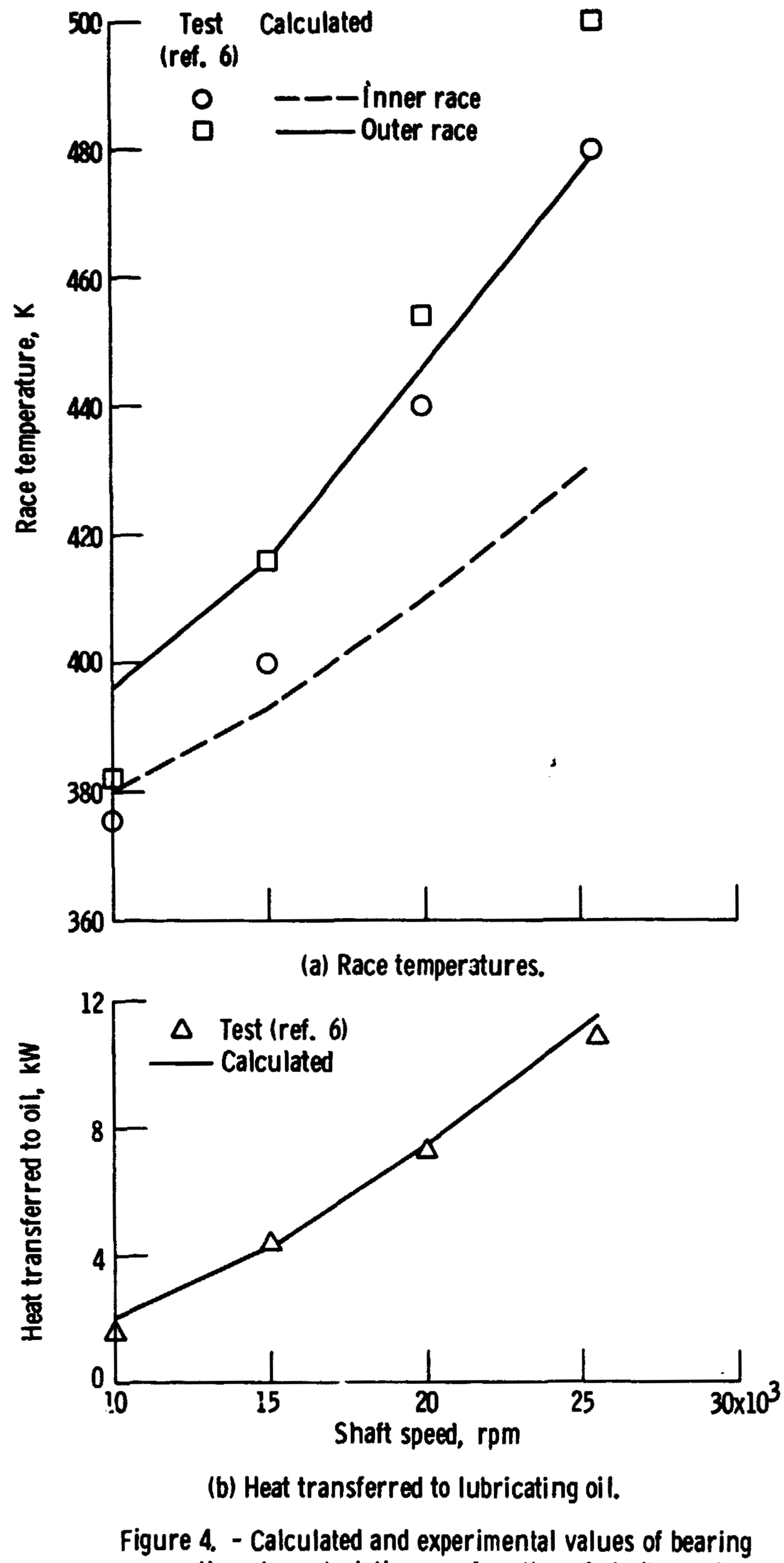
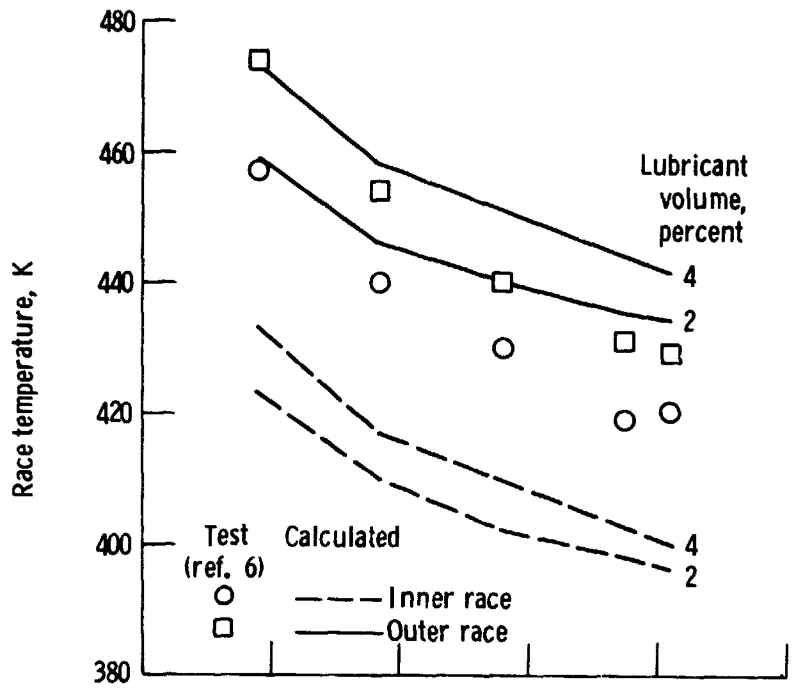
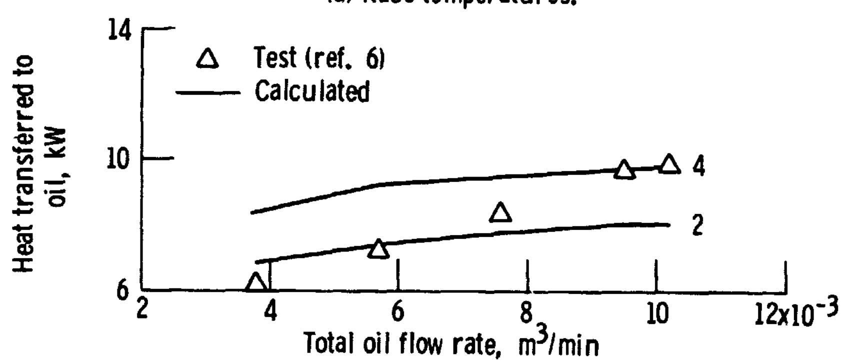


Figure 4. - Calculated and experimental values of bearing operating characteristics as a function of shaft speed. Load, 8900 N (2000 lb); lubricant flow rate, 0.0057 cubic meter per minute (1.5 gal/min); lubricant volume, 2 percent.



(a) Race temperatures.



(b) Heat transferred to lubricating oil.

Figure 5. - Calculated and experimental values of bearing operating characteristics as functions of total oil flow rate. Shaft speed, 20 000 rpm; radial load, 8900 N (2000 lb).

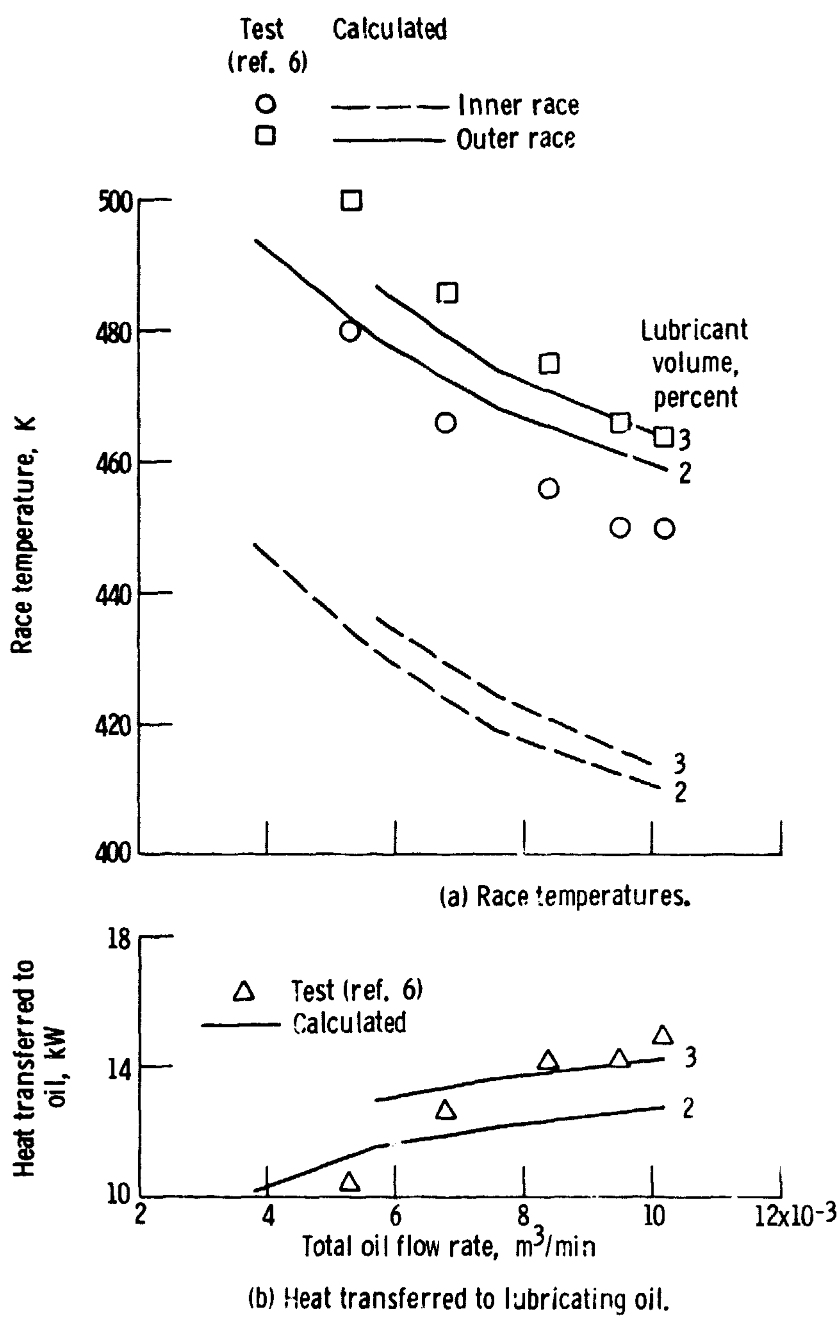


Figure 6. - Calculated and experimental values of bearing operating characteristics as functions of total oil flow rate. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb).

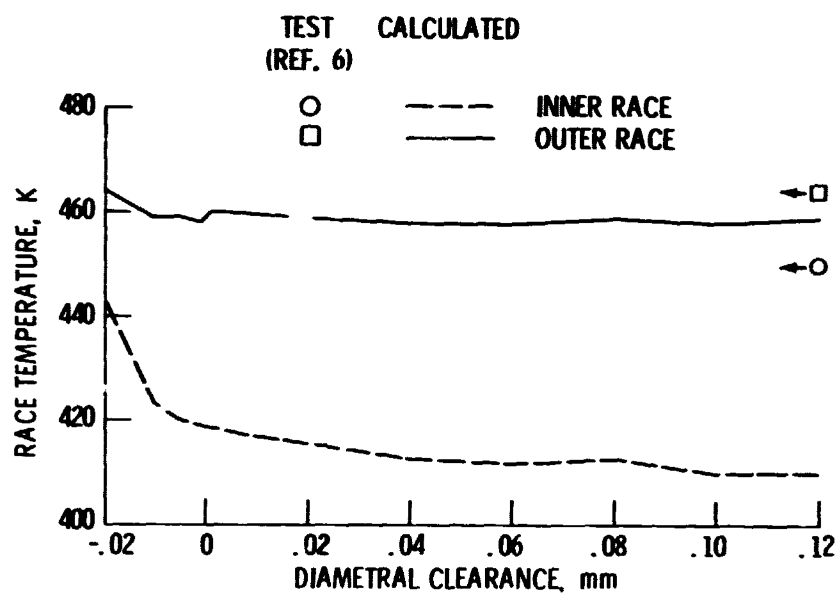
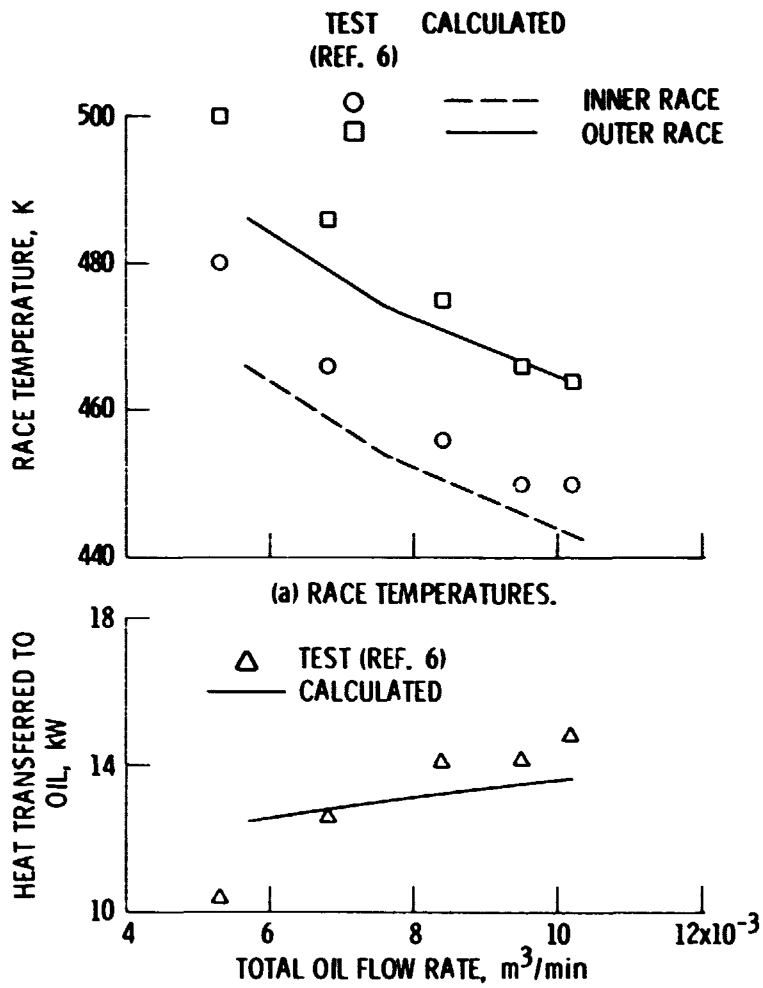


Figure 7. - Calculated race temperatures as a function of diametral clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); total oil flow rate, 0 0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent. Test values shown for comparison, plotted at maximum possible clearance.



(b) HEAT TRANSFERRED TO LUBRICATING OIL.

Figure 8. - Comparison of calculated and experimental bearing data using a diametral clearance of -0.02 mm in the computer program. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant volume, 2 percent.

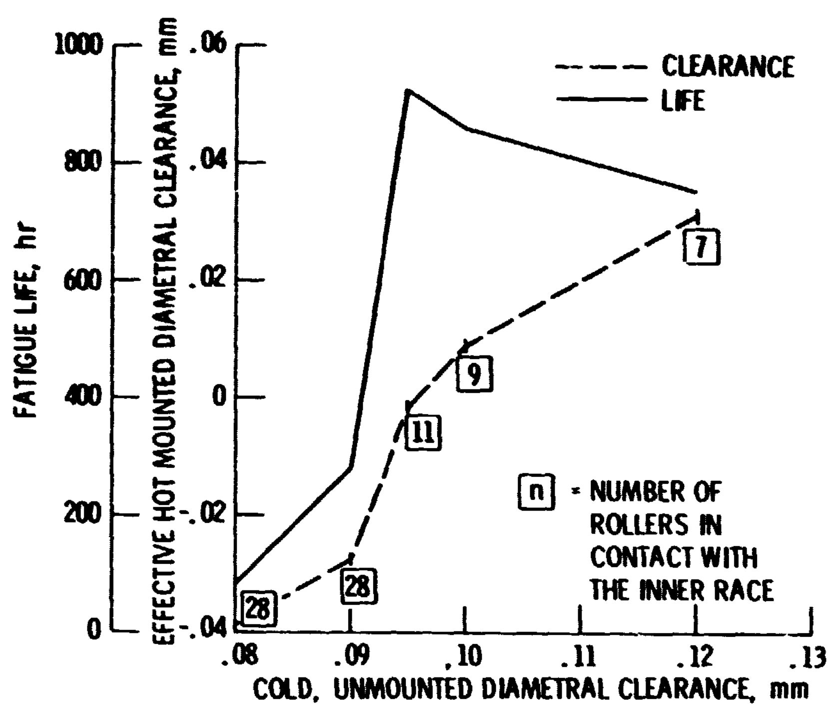
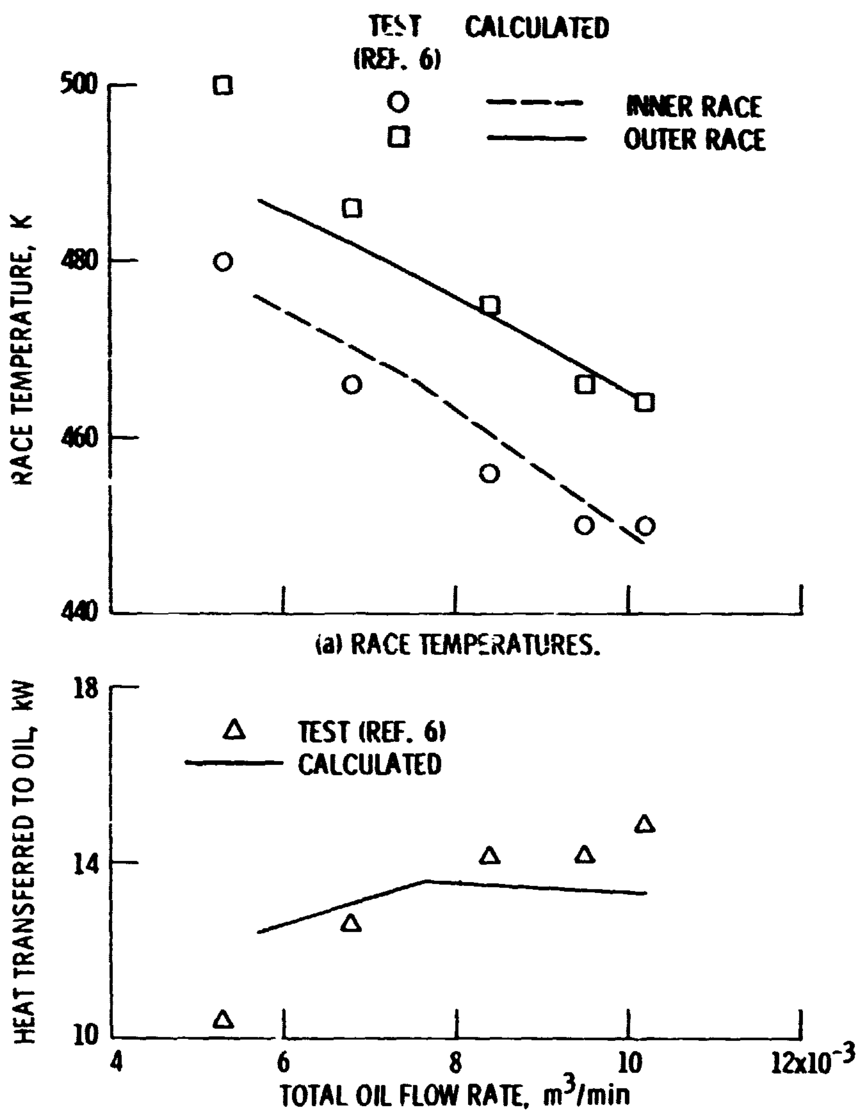
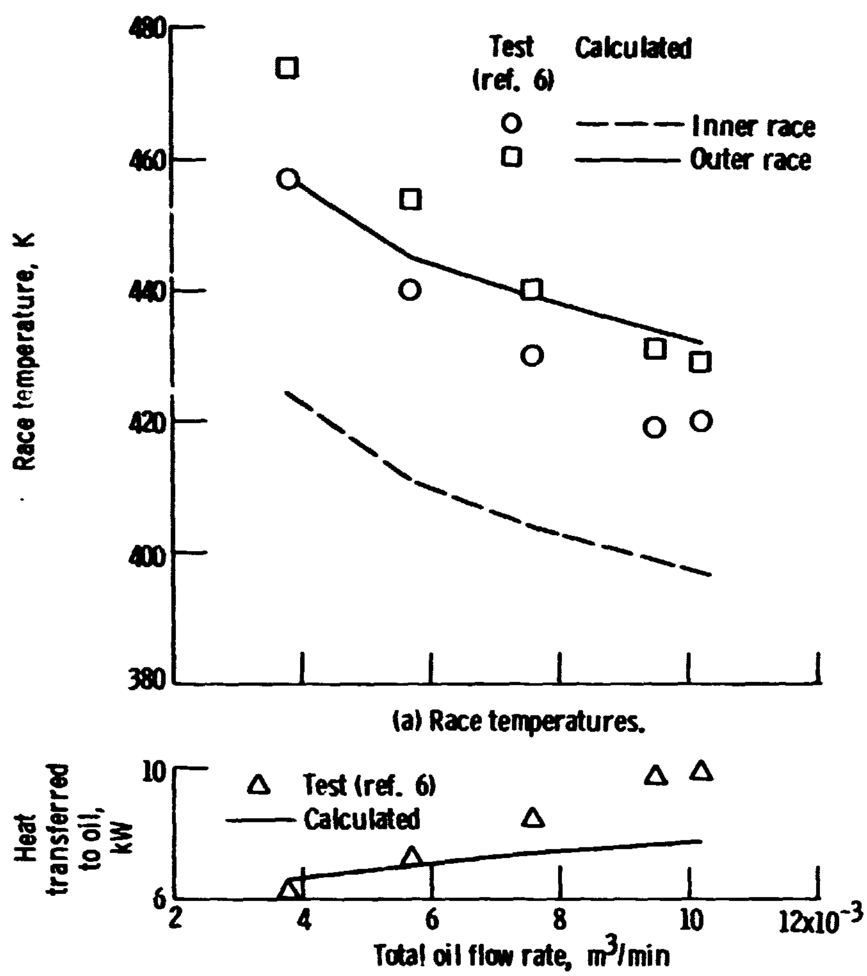


Figure 9. - Calculated values of effective hot mounted clearance and fatigue life as functions of the cold, unmounted clearance. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); oil flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent.



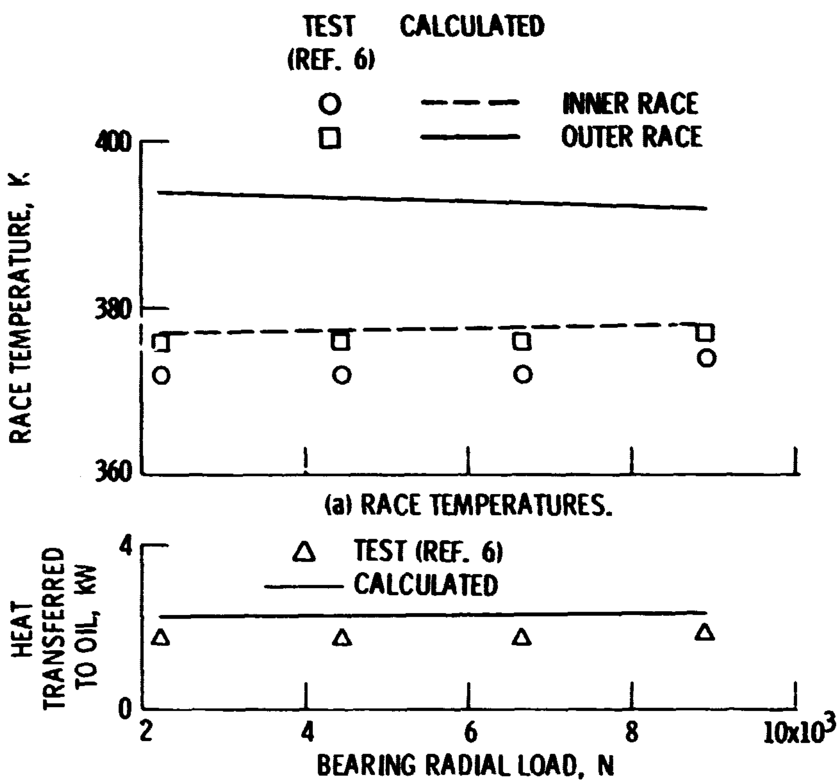
(b) HEAT TRANSFERRED TO LUBRICATING OIL.

Figure 10. - Comparison of calculated and test bearing data using an input cold diametral clearance of 0.09 mm in the computer program. Shaft speed, 25 500 rpm; radial load, 8900 N (2000 lb); lubricant volume, 2 percent.



(b) Heat transferred to lubricating oil.

Figure 11. - Comparison of calculated and experimental bearing data using a cold diametral clearance of 0.09 mm in the computer program. Shaft speed, 20 000 rpm; radial load, 8900 N (2000 lb); lubricant volume, 2 percent.



(b) HEAT TRANSFERRED TO LUBRICATING OIL.

Figure 12. - 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; total oil flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent.

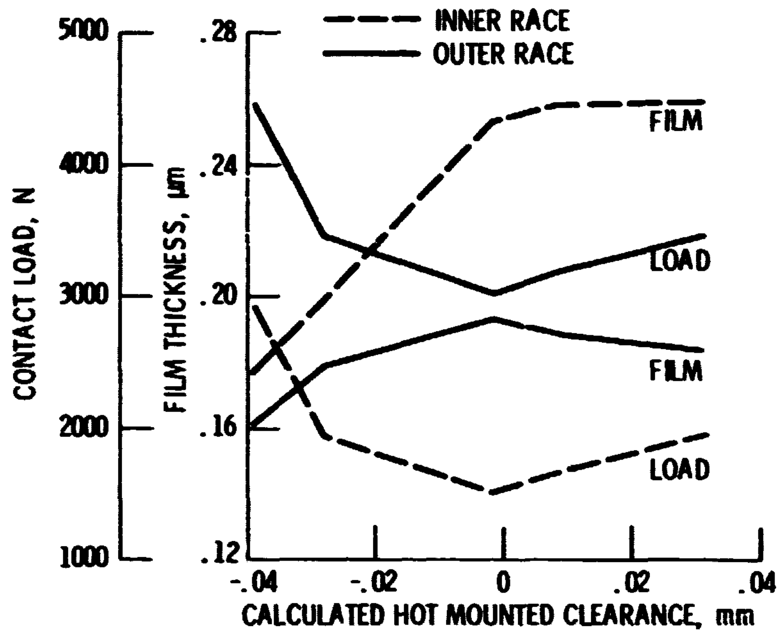


Figure 13. - Film thickness and contact load at the most heavily-loaded roller as a function of calculated hot, mounted clearance. Shaft speed, 25 500 rpm; lubricant volume, 2 percent; oil flow rate, 0.0102 cubic meter per minute (2.7 gal/min); load, 8900 N.

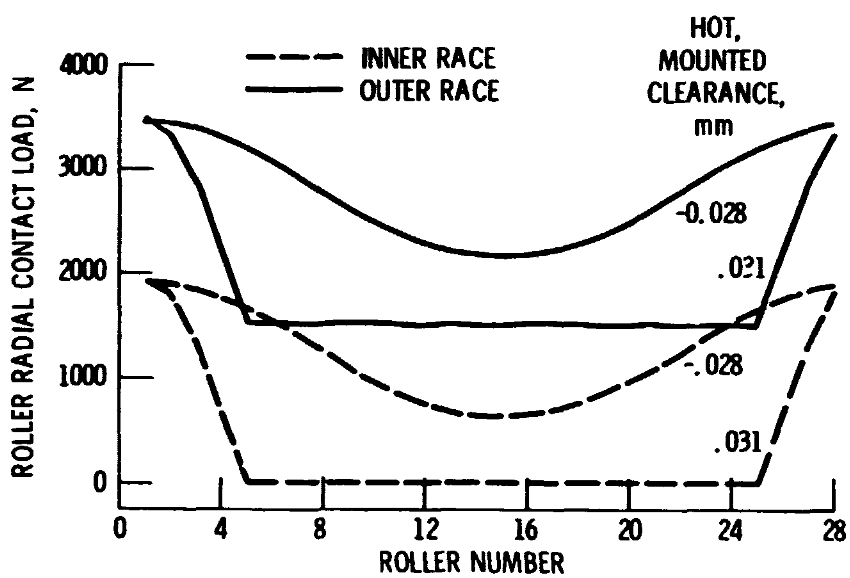


Figure 14. - Roller-race contact load variation with roller position for two values of clearance. Applied load at number 1 roller position. Shaft speed, 25 500 rpm; radial load, 8900 N; flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent.

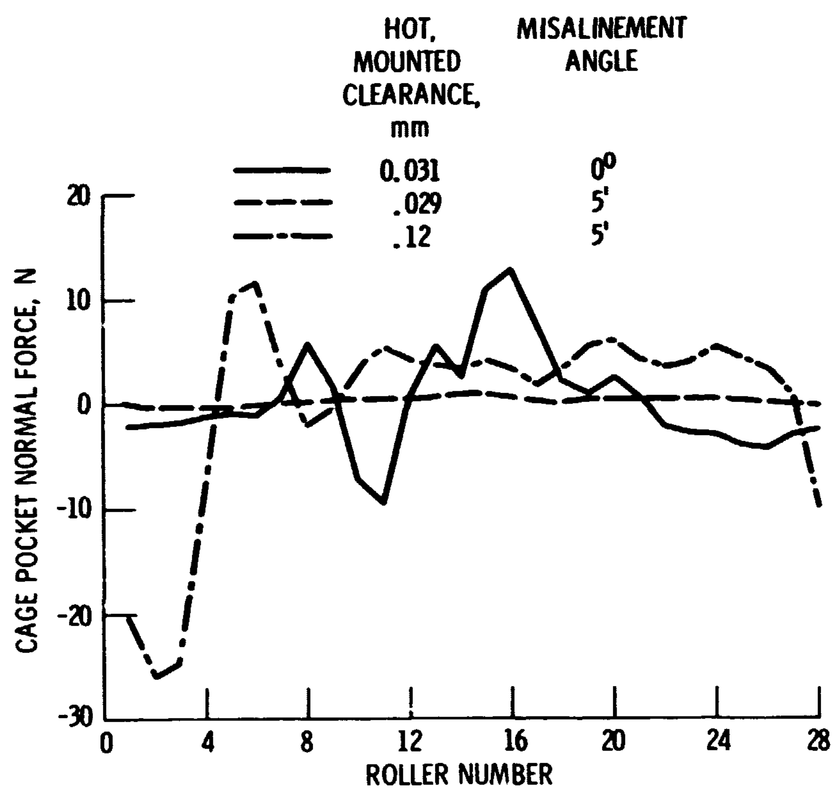


Figure 15. - Cage pocket forces as a function of roller number. Positive force is the cage pushing the roller. Bearing load at roller number 1. Shaft speed, 25 500 rpm; load, 8900 N; flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent.

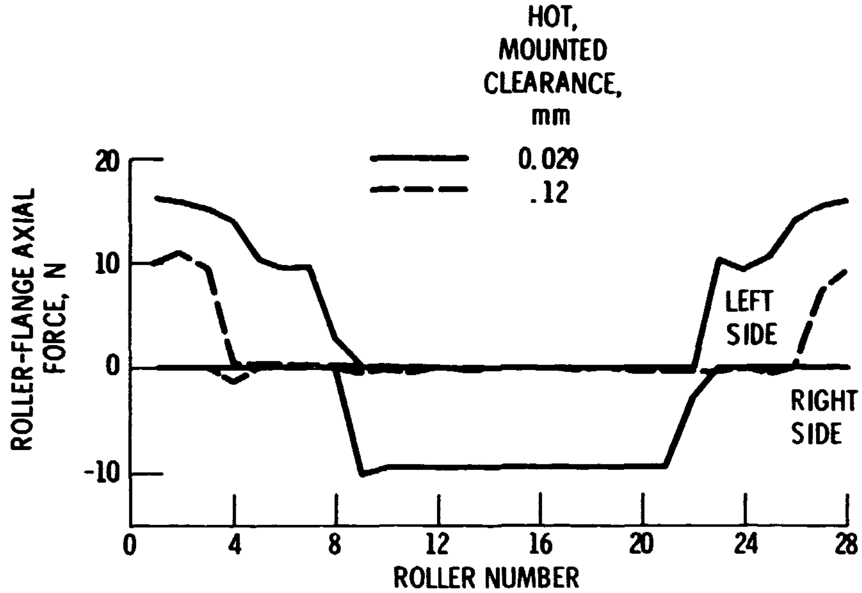


Figure 16. - Roller-flange forces as a function of roller number. Misalinement angle, 5 minutes. Shaft speed, 25 500 rpm; load, 8900 N; lubricant flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent; bearing load at roller number 1 position.

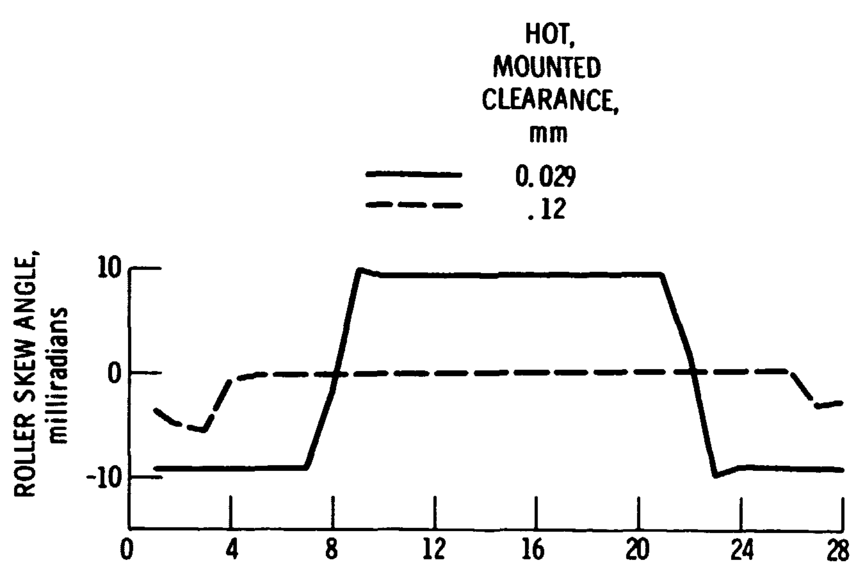


Figure 17. - Roller relative skew angle as a function of roller position. Bearing load of 8900 N at roller number 1 position. Misalinement angle, 5 minutes. Shaft speed, 25 500 rpm; lubricant flow rate, 0.0102 cubic meter per minute (2.7 gal/min); lubricant volume, 2 percent.