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Summary

Two of the computer programs developed for the analysis of rolling-element bearings are Planetsys and Spherbean. Planetsys can simulate the thermomechanical performance of a multistage planetary power transmission, including the operation of a spherical roller bearing. Spherbean can predict the performance values for a spherical roller bearing in a planetary application, including the effects of misalignment with outer-ring rotation. Since it is important to compare values calculated by these programs with actual measured data, both programs were used to simulate the thermal performance of an OH-58 helicopter main rotor transmission. After a steady-state analysis was obtained, the transmission temperatures were calculated as a function of time, assuming dry friction. In this manner, the transient thermal analysis was obtained for a transmission operating after the lubricating oil had been drained from it. The calculated temperatures were then compared with data obtained during a survivability test of the OH-58 transmission wherein the transmission was ultimately allowed to operate after all the oil was drained from it until complete failure, which occurred after about 30 minutes. The transmission was operating at a 75-percent power rating of 150 kW (202 hp) and contained three planets mounted on double-row spherical roller bearings. There were twelve 13-mm-diameter rollers per row. The bearing bore was 31.8 mm (1.25 in.), and the nominal contact angle was 15°. Both computer programs produced reasonable results. For the steady-state analysis the predicted temperatures of the mast shaft and the upper and lower cases were within 3 K (5° F) or 1 percent of the corresponding measured values. For the transient analysis the temperatures predicted by Spherbean were within 3 percent at an elapsed time of 15 minutes and within 9 percent at 25 minutes. Planetsys predicted temperatures slightly higher than Spherbean when using the same coefficient of friction (0.075). With zero misalignment, Spherbean predicted that the sun gear would be the hottest component at an elapsed time of 20 minutes. Spherbean also predicted that the bearing cage would experience a large rapid increase in temperature if the bearing became misaligned by 1°.

Introduction

In the past few years there has been a great deal of work done in developing computer programs for analyzing rolling-element bearings (ref. 1). In reference 1, Pirvics outlined the evolution of contemporary software, which included programs for ball bearings (refs. 2 and 3), cylindrical roller bearings (refs. 4 to 6), and spherical roller bearings (refs. 7 and 8). After a computer program is developed, it is important that values calculated by using that program be compared with actual bearing performance data to assess the program's capabilities.

Predicted and experimental performances of large, high-speed ball and roller bearings were compared in reference 9 by using the programs Shaberth (ref. 10) and Cybean (refs. 11 and 12). A similar comparison for spherical roller bearings, using Spherbean, was made in reference 13. A recent comparison for ball bearings using an updated version of Shaberth (ref. 14) was used in reference 15 to derive an equation for the so-called cavity factor. This factor is the percentage of the bearing cavity volume that is occupied by the lubricant and is a required input variable for most of the abovementioned programs. All of these comparisons were made for oil-lubricated bearings with inner-ring rotation by using the steady-state thermal analysis.

There are, however, important applications, such as helicopter transmissions (ref. 16), where severe performance demands are placed on bearings that require analysis for outer-ring rotation, for nonlubricated operation (dry friction), and for time-transient thermal performance. And, for these applications, few comparisons of computer predictions with actual performance data have been published.

Planetsys (ref. 17) and Spherbean (ref. 18) are two computer programs that have been developed for the analysis of rolling-element bearings used in these applications. Planetsys can simulate the thermo-mechanical performance of a multistage planetary power transmission, including spherical roller bearings. Spherbean, updated as part of a transmission system technology program (ref. 19), can include bearing operation with outer-ring rotation and with misalignment. Actual performance data for a helicopter planetary transmission can be obtained from reference 20, in which an OH-58 main rotor transmission was experimentally investigated. At the end of the performance tests, all the oil was drained from the bottom of the gearbox, and the transmission was allowed to operate to complete failure, which occurred in a little less than 30 min (ref. 20).

The updated computer programs Spherbean (ref. 18) and Planetsys (ref. 17) were used to simulate the operation of the OH-58 helicopter main rotor transmission. This report compares the steady-state and time-transient thermal results obtained with corresponding experimental data from reference 20.

OH-58 Helicopter Main Rotor Transmission

The OH-58 helicopter main transmission (fig. 1) is rated for 201-kW (270-hp) continuous duty and 236 kW (317 hp) at takeoff for 5 minutes. The 100-percent input speed is 6200 rpm, and the output speed is 354 rpm (ref. 20).

The input shaft drives a 19-tooth spiral-bevel pinion that meshes with a 71-tooth gear. The input pinion shaft is mounted on triplex ball bearings and one roller bearing. The 71-tooth bevel gear is carried on a shaft mounted in duplex ball bearings and one roller bearing. The bevel gear shaft drives a floating sun gear that has 27 teeth. The power is taken out through the planet carrier, which contains three 35-tooth planet gears mounted on spherical roller bearings. The ring gear (99 teeth) is splined to the top case and therefore is stationary. The overall gear ratio is 17.44:1 reduction.

The planet bearing inner races and rollers are AISI M-50 steel. The outer races and planet gears, which are integral, are AISI 9310. The cage material is 2024-T4 aluminum. The gear shaft duplex bearing is CVM 52CB. All other bearings are AISI 52100 with bronze cages. The sun gear and ring gear are Nitralloy N (AMS6475). The input spiral-bevel gear set is AISI 9310. Lubrication is supplied through jets located in the top case. An external air-oil cooler is used (fig. 1(b)).

Experimental Data

The experimental data used for comparison purposes in this report were initially reported in reference 20. The data used were from the final test, which was a survivability test. The transmission was operated at a power level of 150 kW (202 hp), which is a 75-percent power rating. The lubricant was an oil qualified to MIL-L-23699 specification. The input shaft speed was 6200 rpm. When the temperatures had stabilized, the oil temperature into the transmission was about 361 K (190° F). At this point the drain plug was removed and the oil drained from the gearbox. The oil pressure reached zero in about 1 1/2 minutes. The oil was completely drained after 2 minutes. The transmission operation was continued until complete failure occurred, almost 30 minutes after the oil drain was opened. The locations of the 23 iron-constantan thermocouples used to gather the thermal data are shown in figure 1. A complete description of the test procedures, data acquisition, and test results is given in reference 20.

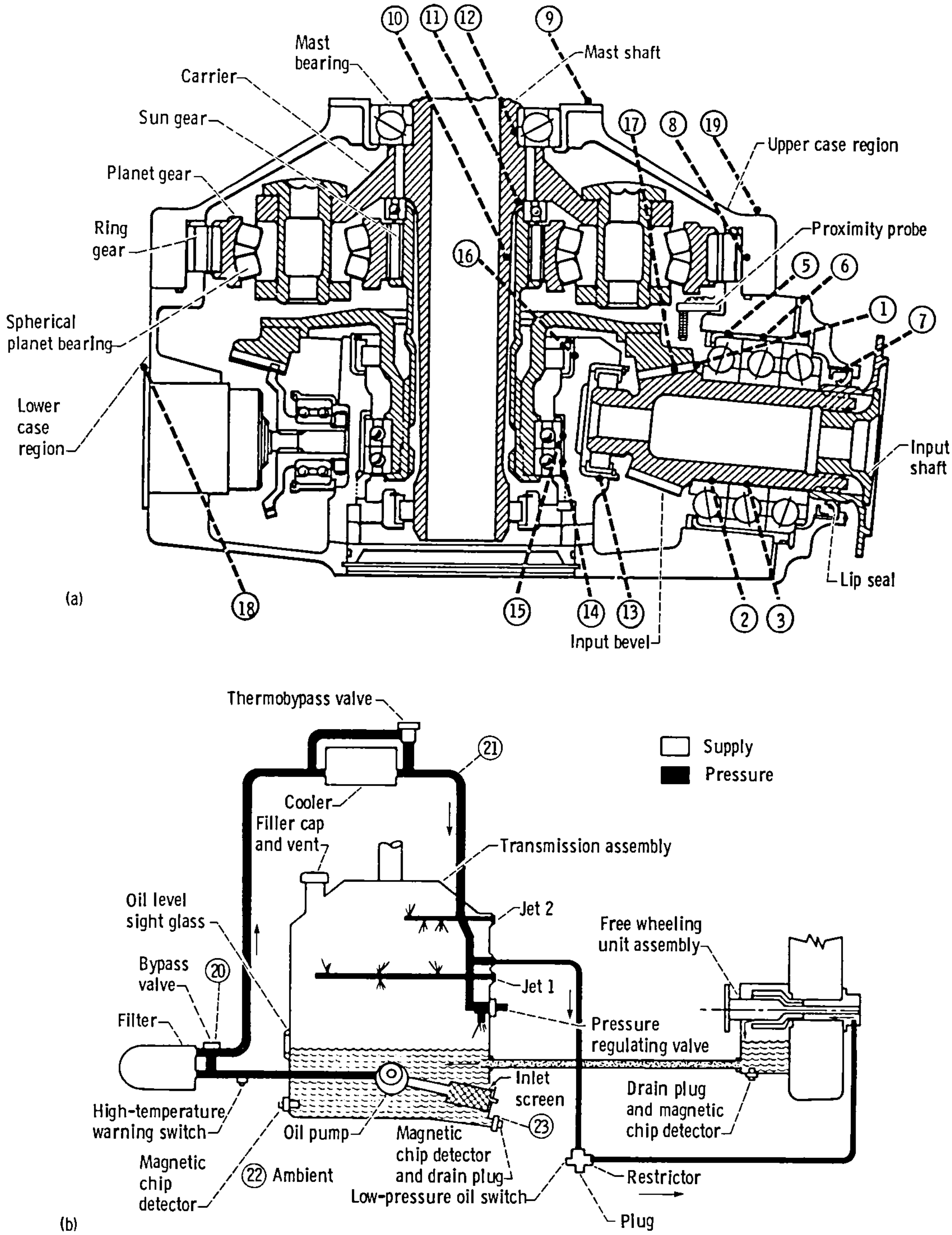
The spherical roller bearings used in the planetary had an integral geared outer-ring design, with two rows of 13-mm-diameter rollers and 12 rollers per row. The bore was 31.8 mm (1.25 in.) and the nominal contact angle was 15°. The bearing is shown in figure 2 and more complete specifications are given in table I.

Computer Programs

Two computer programs were used. Planetsys (ref. 17) was developed to simulate the thermomechanical performance of a multistage planetary power transmission. The major function of the program is to compute, for any of the six possible kinematic inversions for a planetary system, the performance characteristics of a planet bearing. This bearing can be either a cylindrical or a spherical roller bearing and can contain one or two rows of rollers. The outer ring can be rigid or flexible.

Spherbean (ref. 18) was created to simulate the thermomechanical performance of double-row spherical roller bearings under a variety of operating conditions. The program can analyze the effect of misalignment with outer-ring rotation and can include the centrifugal effects for a planetary application.

Both programs are capable of either steady-state or time-transient temperature mapping of an axisymmetric mechanical system. Program input includes bearing geometry, bearing material and lubricant properties, and bearing operating conditions (such as load, speed, and ambient temperature). When the programs are used for thermal analysis, additional input is required, since all



(a) Cross section of transmission showing location of instrumentation.
(b) Transmission lubrication system.

Figure 1.—OH58A main transmission.

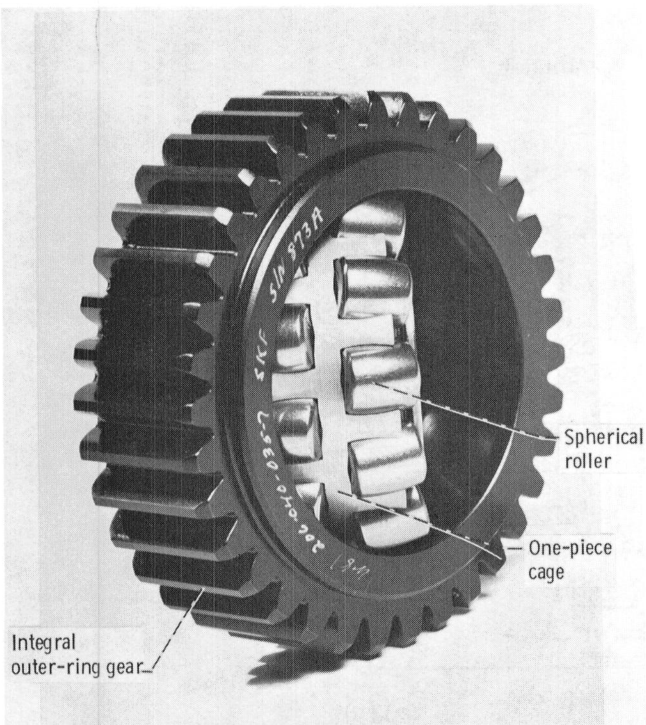


Figure 2.—Planet bearing.

TABLE I.—PLANET ROLLER BEARING SPECIFICATIONS

Parameter	
Outer ring	
Groove radius, mm (in.)	37.993 (1.4958)
Radius to neutral axis, mm (in.)	41.35 (1.628)
Gear pitch diameter, mm (in.)	100.37 (3.9516)
Number of gear teeth	35
Gear pressure angle, deg	24.6
Inner ring	
Groove radius, mm (in.)	38.080 (1.4992)
Bore diameter, mm (in.)	31.75 (1.250)
Nominal contact angle, deg	15
Flange angle, deg	No flange
Rollers	
Maximum diameter, mm (in.)	13.00 (0.5118)
Effective length, mm (in.)	11.48 (0.4520)
Overall length, mm (in.)	13.51 (0.5320)
Crown radius, mm (in.)	36.93 (1.454)
Number of rollers per row	12
Number of rows	2
Roller end radius	Flat
Cage	
Type	One-piece, roller-riding
Tangential pocket clearance, mm (in.)	0.203 (0.008)
Web thickness, mm (in.)	6.3 (0.25)
Bearing	
Cold diametral clearance, mm (in.)	0.0584 (0.0023)

the thermal nodes must be defined. The maximum number of nodes permitted is 100. The nodal system used with Spherbean for the steady-state analysis is shown in figure 3. The 37 air or metal nodes are shown in figure 3(a) and the 18 lubricant nodes in figure 3(b). For the transient analysis, all the lubricant nodes were removed and some of the remaining nodes were then renumbered. The nodal system for Planetsys was essentially the same.

Results and Discussion

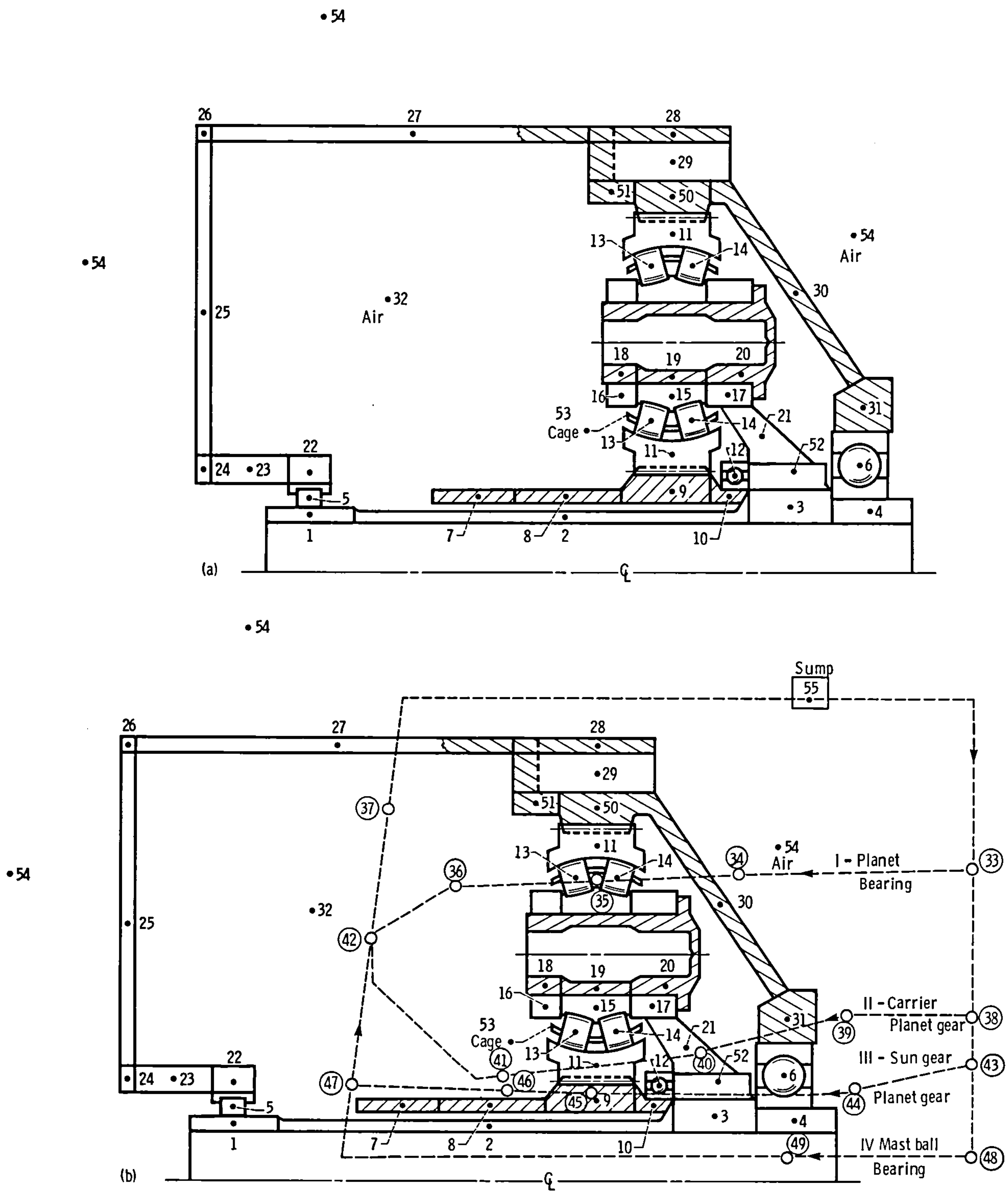
The analysis was accomplished in two parts. First, the programs (Planetsys and Spherbean) were run for comparison with the steady-state data from the OH–58 helicopter main rotor transmission. Then, using the values obtained as a starting point, the lubricant nodes were removed and each program was run in the transient thermal mode, computing temperatures as a function of time. In this manner, the situation was simulated where a transmission was operated after all the oil had drained out.

Steady-State Analysis

Use of the thermal analysis routines in either computer program requires a definition of each nodal area, type of heat transfer between nodes, and the corresponding heat-transfer coefficient. The values used for the conduction heat-transfer indexes are shown in table II, and the values used for convection indexes in table III. The oil flow indexes are shown in table IV. Some of the conduction indexes are derived for parallel or series-type heat flow through different materials as outlined in references 18 and 20. The flow indexes are based on the oil volume flow rate, a specific heat of 2175 W-sec/kg-°C (0.52 Btu/lb-°F), and a density of 900 kg/m^3 (56 lb/ft^3).

The convection indexes are based generally on the equations outlined in references 17 and 18. However, since the bearing is moving as part of a planetary system as well as rotating about its own centerline, it is not certain which equation applies. Using several of the equations suggested in the User's Manuals (refs. 17 and 18) produced values from 1000 to 7000 W/m^2-°C. The value of 2000 W/m^2-°C was chosen as most appropriate for index number 21.

The thermal routines also permit input of values of known temperature and constant generated heat. Thus, the oil inlet node (node 55, fig. 3) was set to 361 K (190° F) and the ambient air (node 54, fig. 3) to 300 K (80° F). The constant generated heat values are shown in table V. These include the mast support bearings and the planetary gear meshes. With the gear mesh, it was assumed that one-half of the heat generated went into the planetary gear and one-half went into the sun gear for the inner mesh. Likewise, it was assumed that one-half of the heat generated went into the planetary gear and one-half



(a) Metal and air nodes.
(b) Lubricant nodes and flow paths.

Figure 3.—Nodal system used with Spherbean computer program.

TABLE II.—CONDUCTION HEAT-TRANSFER INDEXES

Index number	Description	Value, W/m-°C
1	Bearing, bearing post parts—steel	45.0
2	Mast shaft—steel	38.0
3	Bottom main case, cast—magnesium	56.0
4	Top main case, forged—aluminum	138.0
5	Ring gear to bottom case—steel/magnesium	44.3
6	Ring gear to top case—steel/aluminum	65.6
7	Bottom case to top case (overlap)—magnesium/aluminum	80.0
8	Case joint (inside out)	121.6
9	Case joint (up and down)	63.8

TABLE III.—CONVECTION HEAT-TRANSFER INDEXES

Index number	Description	Value, W/m²-°C
21	Bearing to oil	2000
22	Bearing to air	79
23	Sun shaft to air	122
24	Planet post to air	65
25	Mast shaft to air	38
26	Main case, sides to outside air	7
27	Main case, bottom to outside air	3
28	Main case, top to outside air	6
29	Main case to inside air	30

TABLE IV.—FLUID FLOW INDEXES

Flow path[a]	Description	Value, W/°C
I	Total to all planet bearings	96
II	To carrier and planet gears	30
III	To sun gear, planet gears and carrier bearing	44
IV	To mast bearing	49

[a]See fig. 3(b).

TABLE V.—CONSTANT HEAT GENERATION INPUT

Node number (fig. 3)	Value assigned to node, W
5	10
6	25
9	188
11	[a]377
12	25
50	188

[a]Value is for Spherbean program. Due to program differences, the equivalent for Planetsys is 126 W.

into the ring gear for the outer mesh. In each case, the heat generated at the mesh was assumed to be 0.25 percent of the power being transmitted. The program difference noted in table V is that, with respect to the planetary nodes, Spherbean information is input as the total for all planets whereas Planetsys input information is for one planet.

Calculations were then made with both programs (Planetsys and Spherbean) using the abovementioned input for a transmitted power level of 150 kW (202 hp). The results are shown in table VI. Also shown are the experimental temperatures indicated by the thermocouple located near the corresponding node. The calculated values are very close to the experimental temperatures for both computer programs, generally within 2 K (4° F), which is less than a 1-percent difference. The calculated temperatures shown in table VI were determined using the NASA models for film thickness and traction as indicated in references 17 and 18. Reference 17 also notes that the program exists as an SKF version using SKF models for the film thickness and traction. Using the same input data with the SKF versions resulted in calculated temperatures that were approximately 1 K (2° F) higher than those shown in table VI, still less than a 1-percent difference compared to the experimental values.

Usually an important parameter in the calculation of bearing operating characteristics is the amount of lubricant in the bearing cavity (refs. 9 and 15). However, because of the relatively low bearing speeds in this application, little change in temperature was observed as the percent volume of the bearing cavity that is occupied by the lubricant was varied from 1 to 50 percent. The internal radial clearance of the bearing (at operating conditions) was varied from 0.061 to −0.013 mm (0.0024 to −0.0005 in.) and the temperatures changed only 1 K (2° F). Since these steady-state temperatures were very close to the experimental values, they were used as the starting temperatures for the transient solution.

TABLE VI.—STEADY-STATE TEMPERATURES

Node (fig. 3(a))	Description	Thermocouple (fig. 1)	Temperature, K		
			Experimental (ref. 20)	Calculated[a]	
				Planetsys[b]	Spherbean[b]
	Mast shaft—				
2	Lower	10	364	366	365
3	At carrier	11	364	366	365
4	At mast bearing	12	364	365	365
	Planet bearing—				
11	Outer ring			366	365
13,14	Spherical rollers			366	364
15	Inner ring			366	364
53	Cage			364	363
21	Carrier arm			365	364
27	Lower case—outer wall	18	364	362	361
	Upper case—				
28	At joint			366	366
29	At ring gear	8	364	366	366
31	At mast bearing	9	361	363	364
50	Ring gear			367	367

[a]Oil-inlet to transmission (node 33) and outside air (node 54) temperatures were set at the measured values of 361 and 300 K, respectively.
[b]Using NASA version of program. Results with the SKF version were approximately 1 K higher.

Transient Thermal Analysis

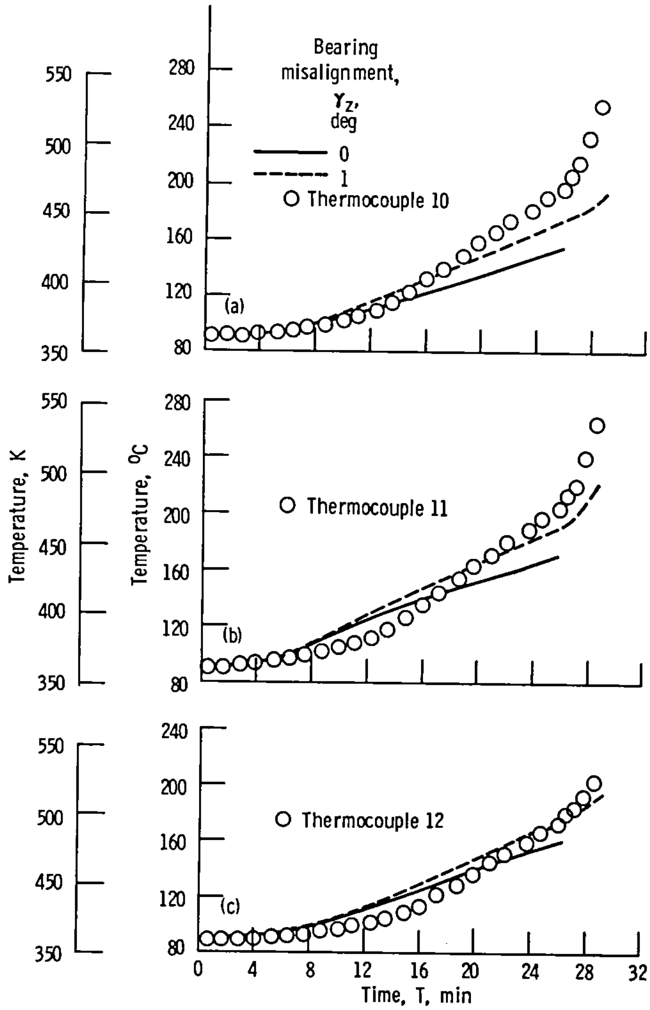
After the steady-state calculations were completed, the lubricant nodes were removed and the input data were prepared for operation of the computer programs in the transient thermal mode. In this manner, it was intended to simulate operation of the OH-58 main rotor transmission after all the oil drained. Since it took 1 1/2 minutes for the oil pressure to fall to zero in the experimental investigation of reference 20, the predicted temperatures were kept constant for 3 minutes and then allowed to change per the transient thermal solutions. (Note that the transient solutions assume dry friction.)

The first calculations made with Spherbean used a bearing misalignment angle of 0°, a gear mesh loss of 0.25 percent as noted for the steady-state calculations, and a dry friction coefficient of 0.075 for the roller to raceway contact (as used in ref. 21). These calculations were continued only to an elapsed time of 26 minutes, for it was apparent from the experimental data of reference 20 that a large change in the time rate of change of temperature took place at about 26 minutes.

The results for the mast shaft are shown in figure 4. Nodes 2, 3, and 4 (fig. 3(a)) are nearest thermocouples 10, 11, and 12 (fig. 1). For comparison, the results obtained using a bearing misalignment γ_z of 1° (starting at 5 min) are also shown. Generally the predicted values are fairly close to the measured temperatures for the first few minutes and then tend to deviate because of the differences in slope. Still, while node 2 shows a value quite a bit lower than that from thermocouple 10 after 18 minutes, node 4 is reasonably close to thermocouple 12 even after 24 minutes of elapsed time. The 1° misalignment produced a curve that was at first farther from the measured data and then tended to be closer at the higher elapsed times.

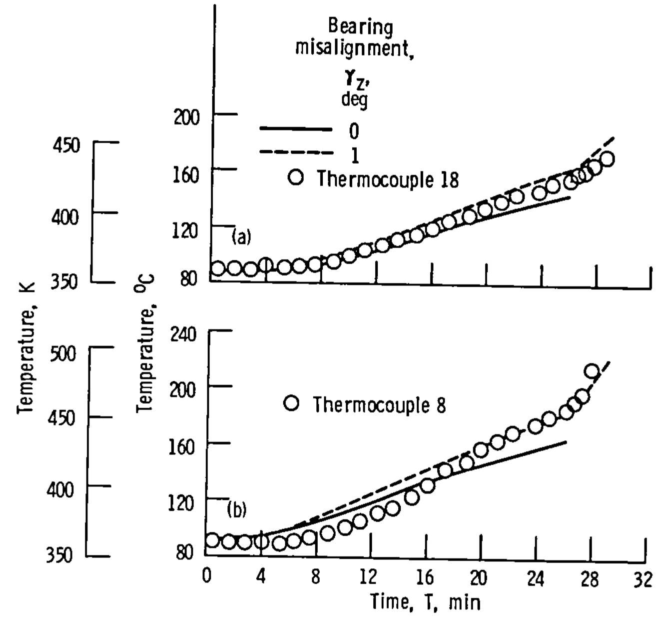
To explore the change in slope of the measured data at an elapsed time of 26 minutes, the input to the program was changed such that the gear mesh heat generation was 1 percent (instead of 0.25) and the friction coefficient was set at 0.2 for the raceway contacts. Those changes affected node 3 mast shaft values more than they affected either node 2 or 4, as expected. These results are also shown in figure 4. The fact that the changes in slope of the experimental data were a little higher than those calculated indicates that the changes occurring in the transmission at this point were at least as severe as the changes made to the Spherbean input data.

The results obtained for the transmission case are shown in figure 5. Here the predicted temperature for node 27 (fig. 3) is compared with the experimental values from thermocouple 18 (fig. 1) and node 28 is compared with thermocouple 8. The predicted values for the lower case are quite close to the experimental data, indicating that the lower case is not affected as much as the mast shaft by changes occurring in the planetary system. Understandably, the upper case comparison is very similar to the comparison of the mast shaft at the carrier



(a) Lower shaft. Node 2.
(b) At junction with carrier. Node 3.
(c) At mast bearing. Node 4.

Figure 4.—Computer predicted temperatures compared to experimental data, mast shaft area. Drain plug removed at $T=0$; oil pressure to zero at $T=1.5$ minutes.



(a) Lower case. Node 27.
(b) Upper case. Node 28.

Figure 5.—Comparison of computer-predicted temperatures with experimental values, transmission case. Drain plug removed at $T=0$; oil pressure to zero at $T=1.5$ minutes.

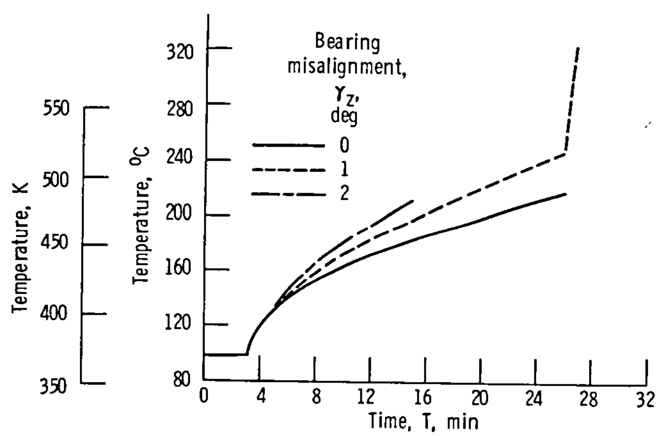
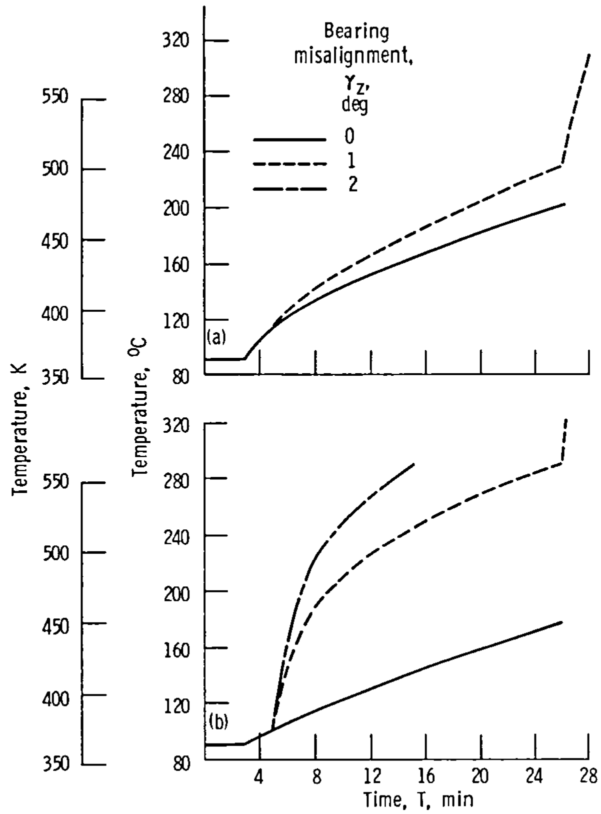


Figure 6.—Temperature of sun gear predicted by Spherbean. Node 9; drain plug removed at $T=0$.

junction (node 3), since both nodes 3 and 28 are directly connected to the planetary system.

While there were no thermocouples placed on either the sun gear or the planetary bearings, it is informative to observe the results of the Spherbean prediction for these areas. The calculated values of the sun gear temperature are shown in figure 6. The sun gear is predicted to be much hotter than the previous components—about 50 K (90° F) at an elapsed time of 20 minutes. The changes to the input data at 26 minutes, noted previously, produced a dramatic increase in temperature and in the rate of temperature change for the sun gear.

The predicted temperatures for the planetary spherical roller bearing are shown in figure 7. The outer ring temperatures were hotter than the previous components but cooler than the sun gear—about 16 K (30° F) cooler at the 20-minute mark. The outer ring also showed a large increase in the rate of temperature change when the program input was altered, as noted previously, at an elapsed time of 26 minutes. The bearing cage (fig. 7(b)), however, showed the most dramatic changes in temperature when the misalignment angle was changed from 0° to 1°. A calculation was also made for a misalignment angle of 2°. The aluminum bearing cage has the potential for rapid temperature change when the races become misaligned. And indeed reference 20 speculates that one of the bearings lost its cage around 7 min. Reference 20 also speculates that a second bearing lost its cage around 14 minutes.



(a) Outer ring. Node 11.
(b) Bearing cage. Node 36 (steady-state node 53 in fig. 3(a)).

Figure 7.—Predicted temperature for spherical planetary bearing. Drain plug removed at $T=0$.

The experimental results clearly show an increase in the rate of temperature change around 14 minutes (figs. 4 and 5). The measured temperatures just started to change around 5 minutes. The predicted results compare reasonably well, being generally conservative, up to 14 to 16 minutes, with the assumption of zero misalignment. And the predictions with a misalignment clearly show that the cage would have been in severe circumstances if the bearing races misaligned. Once the cage fails and the rollers turn sideways, the bearing computer program is no longer valid. Thus, it is not surprising that the predicted temperatures generally were too low after about 15 minutes.

Presently there are no well-proven criteria for predicting bearing failure. Certainly the loss of bearing internal radial clearance and the loss of material hardness, both of which are temperature related, will lead to bearing failure, but no one can predict exactly when it will occur. Perhaps the best we can do is to set a temperature limit. This is the approach taken in reference 21 where the time to failure was specified as the time required for any bearing component to reach 773 K (932° F). But that assumed steel components. Perhaps the aluminum should be limited to a lower temperature—for example, 478 to 533 K (400° to 500° F). The predicted temperature values after about 15 minutes then indicate how the transmission might have reacted had the cages not failed, causing changes in the bearing heat generation.

Without the cage failures, the transmission could have run much longer, based on the computed temperatures for zero misalignment. In fact, a straight-line extrapolation of the sun gear temperature indicates that it would have taken 2 hours for that component to reach 773 K (932° F). The same is true for the bearing outer ring and the mast shaft. However, the transmission case is made from aluminum and magnesium and thus subject to a different criterion. The same straight line extrapolation shows that the upper case would reach 478 K (400° F) in about 40 minutes and the lower case would take about 50 minutes. The higher limit of 533 K (500° F) would be reached in an additional 17 minutes for a total of 57 minutes for the upper case and 67 minutes for the lower case. Thus, it might be possible (depending on the stresses involved), for the outer case to become a time-limiting factor for the operation of the transmission after the loss of lubricant. Therefore, based on these comparisons and observations, Spherbean can be considered a useful tool in the analysis of a planetary transmission.

Planetsys was also used to predict the transient thermal operating characteristics of the transmission, using the same gear mesh loss of 0.25 percent. Bearing misalignment was not specifiable and thus remained at zero. When the calculations were made with the dry friction coefficient set at 0.075 as before, the temperatures of the housing and mast shaft nodes were about 5 K (9° F) higher than the corresponding Spherbean values at an elapsed time of 15 minutes. The sun gear node and the bearing nodes (except the cage) were 16 K (30° F) higher. With the friction coefficient set at 0.05 for Planetsys, the mast shaft and casing nodes were only about 2 K (4° F) higher while the sun gear and bearing nodes were about 7 K (13° F) higher than for Spherbean (with 0.075 coefficient), also at an elapsed time of 15 minutes. The difference in the results between the two programs is that Planetsys is predicting a much higher bearing heat generation than Spherbean for the dry friction mode. In both programs, the heat generation is calculated as the product of the friction force times the sliding velocity. Since the loading on each roller is about the same for both programs, the difference in heat generation is due to the difference in the predicted sliding velocities.

Since the value of 0.075 for a friction coefficient f compares with other references (e.g., ref. 21) and is a more reasonable number, it is concluded that Planetsys incorrectly predicts dry friction torque and that to utilize the program the lower value of 0.05 should be used. Additional Planetsys runs with $f=0.05$ showed that setting a roller skew angle of 1° produced results very close to those obtained with Spherbean at a 1° misalignment angle, except that the cage temperature remained low for Planetsys and did not show the potential for high temperature indicated by Spherbean (see fig. 7(b)).

Concluding Remarks

In general, the computer programs predicted temperatures that were very close to the corresponding experimental values. The steady-state temperatures calculated by both Planetsys and Spherbean were within about 2 K (4° F) of the measured data. It may therefore be speculated that the assumptions made were reasonably correct for these programs. Furthermore, the transient solutions were close, especially up to the apparent loss of a second aluminum bearing cage, around 14 minutes of elapsed time. On an absolute temperature basis, all the Spherbean-predicted temperatures were within 3 percent of the corresponding measured values at 15 minutes of elapsed time (with $\gamma_z = 0$ or no misalignment) and were actually within 9 percent even at 25 minutes of elapsed time. It is therefore also speculated that the assumptions made for the transient solutions were reasonable.

The Spherbean program did show that the cage could change temperature rapidly if misalignment were present. This change in temperature is even more spectacular if the friction coefficient is raised, as might be the case when aluminum from the cage transfers to the rollers (e.g., ref. 22). Since the aluminum material rapidly loses its tensile strength as temperatures rise above 366 K (200° F), this calculation showed that it would have been better not to have used aluminum cages for this application and thus tended to verify the conclusion reached in reference 20—namely, that steel cages should be used for that particular bearing.

Finally, the calculations indicate that the transmission probably would have run for about 2 hours had not the bearing cages failed, based on a temperature limit of 773 K (932 F). Since a modified OH-58 main rotor transmission that included bearings with steel cages was operated under the condition of loss of lubricant for over 2 hours in a subsequent test (ref. 20), perhaps the temperature limit criterion suggested by reference 21 is a reasonable value.

Summary of Results

The computer programs Planetsys and Spherbean were used to simulate the thermal performance of an OH-58 helicopter main rotor transmission. After a steady-state analysis was made, the lubricant nodes were removed and the temperatures were calculated as a function of time. In this manner the transient thermal analysis was made of a transmission operating after the oil had drained out. The calculated values of temperature were then compared with experimental data obtained previously. The transmission was operated at a power level of 150 kW (202 hp), which is a 75-percent power rating. Operation continued until complete failure, which occurred about 30 minutes after the oil drain plug was removed. The transmission contained 3 planets mounted on spherical roller bearings. Each bearing had two rows of 13-mm-diameter rollers, with 12 rollers per row. The bore was 31.8 mm (1.25 in.) and the nominal contact angle was 15°. The following results were obtained:

1. For the steady-state analysis, both Planetsys and Spherbean predicted mast shaft and upper and lower case temperatures that were within 1 percent of the corresponding experimental data.
2. For the transient thermal analysis, both Planetsys and Spherbean produced reasonable results. The temperatures predicted by Spherbean were within 3 percent of the corresponding measured values at 15 minutes of elapsed time and within 9 percent at 25 minutes, based on absolute temperatures.
3. Planetsys predicted higher temperatures than did Spherbean when using the same value (0.075) for the coefficient of friction. The mast and case temperatures were 5 K (9° F) higher while the sun gear and planet bearing were 16 K (30° F) higher, after 15 minutes of elapsed time. Using a 0.05 friction coefficient with Planetsys reduced the temperature differences to 2 and 7 K (4° and 13° F), respectively.
4. With zero misalignment, Spherbean predicted that the sun gear would be the hottest component at 20 minutes of elapsed time.
5. With a misalignment angle of 1° as input, Spherbean predicted a large increase in temperature of the spherical roller bearing aluminum cage, a component which failed during the experimental test.

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References

1. Pirvics, J.; and Kleckner, R. J.: Prediction of Ball and Roller Bearing Thermal and Kinematic Performance by Computer Analysis. Advanced Power Transmission Technology, G. K. Fischer, ed. NASA CP-2210, 1983, pp. 185-201.
2. Crecelius, W. J.: User's Manual for Steady State and Transient Thermal Analysis of a Shaft-Bearing System (SHABERTH). ARBRL-CR-00386, SKF Industries, Inc., 1978.
3. Gupta, P. K.: Dynamics of Rolling-Element Bearings. Part III: Ball Bearing Analysis. J. Lubr. Technol., vol. 101, no. 3, July 1979, pp. 312-381.

4. Kleckner, R. J.; Pirvics, J.; and Castelli, V.: High Speed Cylindrical Rolling Element Bearing Analysis "CYBEAN."—Analytic Formulation. J. Lubr. Technol., vol. 102, no. 3, July 1980, pp. 380-390.
5. Gupta, P. K.: Dynamics of Rolling Element Bearings, Part I: Cylindrical Roller Bearing Analysis. J. Lubr. Technol., vol. 101, no. 3, July 1979, pp. 293-302.
6. Conry, T. E.: Transient Dynamic Analysis of High-Speed Lightly Loaded Cylindrical Roller Bearings. Part I—Analysis. NASA CR-3334, 1981.
7. Crecelius, W. J.; and Kleckner, R. J.: PLANETSYS, A Computer Program for the Steady State and Transient Thermal Analysis of a Planetary Power Transmission System. SKF-AL-77P011, SKF Industries, Inc., 1977.
8. Kleckner, R. J.; and Pirvics, J.: Spherical Roller Bearing Analysis. J. Lubr. Technol., vol. 104, no. 1, Jan. 1982, pp. 99-108.
9. Coe, H. H.: Predicted and Experimental Performance of Large-Bore High-Speed Ball and Roller Bearings. Advanced Power Transmission Technology, G. K. Fischer, ed. NASA CP-2210, 1983, pp. 203-220.
10. Crecelius, W. J.; Heller, S.; and Chiu, Y. P.: Improved Flexible Shaft-Bearing Thermal Analysis with NASA Friction Models and Cage Effects. SKF-AL-76P003, SKF Industries, Inc., 1976.
11. Kleckner, R. J.; and Pirvics, J.: High Speed Cylindrical Roller Bearing Analysis, SKF Computer Program CYBEAN. Vol. I: Analysis. (AL78P022-Vol-1, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159460, 1978.
12. Kleckner, R. J.; and Pirvics, J.: High Speed Cylindrical Roller Bearing Analysis, SKF Computer Program CYBEAN. Vol. II: User's Manual. (AL78PO23-Vol-2, SKF Industries, Inc.; NASA Contract NAS3-20068.) NASA CR-159461, 1978.
13. Kleckner, R. J.; and Dyba, G.: High Speed Spherical Roller-Bearing Analysis and Comparison with Experimental Performance. Advanced Power Transmission Technology, G. K. Fischer, ed. NASA CP-2210, 1983, pp. 239-242.
14. Hadden, G. B.; et al.: Steady State and Transient Thermal Analysis of a Shaft Bearing System Including Ball, Cylindrical and Tapered Roller Bearings. (SKF-AT81D040, SKF Industries, Inc.; NASA Contract NAS3-22690.) NASA CR-165365, 1981.
15. Parker, R. J.: Comparison of Predicted and Experimental Thermal Performance of Angular-Contact Ball Bearings. NASA TP-2275, 1984.
16. Zaretsky, E. V.; Coy, J. J.; and Townsend, D. P.: NASA Transmission Research and its Probable Effects on Helicopter Transmission Design. NASA TM-83389, 1983.
17. Hadden, G. B.; et al.: Planetsys, a Computer Program for the Steady State and Transient Thermal Analysis of a Planetary Power Transmission System. (SKF-AT81D044, SKF Industries, Inc.; NASA Contract NAS3-22690.) NASA CR-165366, 1981.
18. Kleckner, R. J.; Dyba, G. J.; and Ragen, M. A.: Spherical Roller Bearing Analysis. SKF Computer Program "SPHERBEAN." Volume II: User's Manual. (SKF-AT81D007, SKF Industries, Inc.; NASA Contract NAS3-22807.) NASA CR-167859, 1982.
19. Zaretsky, E. V.: NASA Helicopter Transmission System Technology Program. Advanced Power Transmission Technology, G. K. Fischer, ed. NASA CP-2210, 1983, pp. 15-34.
20. Townsend, D. P.; Coy, J. J.; and Hatvani, B. R.: OH-58 Helicopter Transmission Failure Analysis. NASA TM X-71867, 1976.
21. Ragen, M. A.: Prediction of Time to Failure of the CH-53 Helicopter Power Input Module After Loss of Lubrication. SKF-AL79P013, SKF Industries, Inc., 1979.
22. Sargent, L. B., Jr.; Milz, W. C.; and Atkinson, R. E.: The Effect of Aluminum Transfer Upon the Friction Between Aluminum Alloys and Steel. Lubr. Eng., vol. 39, no. 11, Nov. 1983, pp. 706-711.

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15. Supplementary Notes

16. Abstract

Planetsys and Spherbean, two computer programs developed for the analysis of rolling-element bearings, were used to simulate the thermal performance of an OH-58 helicopter main rotor transmission. Both a steady-state and a transient thermal analysis were made. Temperatures thus calculated were compared with experimental data obtained from a transmission that was operated to destruction, which occurred about 30 min after all the oil was drained from the transmission. Both programs produced reasonable results. Temperatures predicted by Spherbean were within 3 percent of the corresponding measured values at 15-min elapsed time and within 9 percent at 25 min. Spherbean also indicated a potential for high bearing cage temperatures with misalignment and outer-ring rotation.

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