NASA Technical Paper 1196

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APRIL 1978

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National Aeronautics
and Space Administration

Scientific and Technical
Information Office

1978

PREDICTED AND EXPERIMENTAL PERFORMANCE OF JET-LUBRICATED 120-MILLIMETER-BORE BALL BEARINGS OPERATING TO 2.5 MILLION DN

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SUMMARY

Bearing performance characteristics, such as inner- and outer-race temperatures and friction power loss, can be predicted using recently developed computer programs. Two such programs were used, and the calculated values obtained were compared with the corresponding experimental data obtained previously for 120-millimeter-bore bearings. The bearings were operated at thrust loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb) and shaft speeds of 12 000, 16 700, and 20 800 rpm with lubricant flow rates of 3.8×10^{-3} and 8.3×10^{-3} cubic meter per minute (1.0 and 2.2 gal/min). The oil inlet temperature was maintained constant at 394 K (250° F).

The first computer program, one previously used in the design of the test bearings, predicted reasonable temperatures with proper consideration of required input data of housing and shaft end temperatures. However, this program severely underestimated the bearing power losses. The second program, called SHABERTH, also predicted reasonable race temperatures, but it did not require the input of housing and shaft end temperatures. Furthermore, SHABERTH provided a good estimate of bearing power loss. The bearing power loss predictions by both computer programs were a strong function of the value assumed for volume percent of the bearing cavity occupied by the lubricant.

INTRODUCTION

Bearings in current commercial aircraft turbine engines operate at speeds to 2.3 million DN (the speed parameter DN is the bearing bore in mm multiplied by the shaft speed in rpm). However, for some time, trends in gas turbine design have indicated that future engines may require bearings that can operate reliably at DN values of 3 million or higher (ref. 1). Therefore, there has been a great amount of work performed in the area of high-speed bearings in the last few years. Successful high-speed operation of 125-millimeter bearings was reported in reference 2. Reliable, long-life operation of 120-millimeter bearings at 3 million DN was reported in reference 3. The question of how to design bearings for high-speed applications is increasingly being answered by computer studies (ref. 4). There are currently several comprehensive computer programs in use that are capable of predicting rolling bearing operating and performance characteristics. 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. However, very little data have been published which compare computer predictions with actual bearing performance.

The bearings used in reference 3 were designed using the results of calculations made by the computer program first described in reference 5 and subsequently updated by reference 6. Therefore, the objectives of the research reported herein were (1) to calculate the operating characteristics of the 120-millimeter test bearings described in references 3 and 7 by using the computer programs of references 6 and 8, (2) to compare the calculated values of inner- and outer-race temperatures and bearing power loss with the corresponding experimental data from reference 7, and (3) to determine the effect on the calculated values of using two different traction models in the computation by using the program of reference 8.

BEARING TEST DATA

The experimental work reported in reference 7 was performed on the high-speed bearing tester described in detail in reference 9. 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 as shown in figure 1. The lubricant used was a tetra ester, 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-millimeter bore, split inner race with fifteen 20.6-millimeter-(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 three bearing thrust loads, these being 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb), and at three shaft speeds, 12 000, 16 700, and 20 800 rpm. The oil inlet temperature was held constant at 394 K (250° F).

COMPUTER PROGRAMS

The computer programs described in references 6 and 8 are capable of calculating the thermal and kinematic performance of high-speed ball bearings. This calculation includes the determination of inner- and outer-race temperatures and the bearing power loss. Since the test bearing design was essentially based on the early calculations made using reference 6, the first comparisons of predicted and experimental values were performed using the combined load computer program of reference 6. This program is hereinafter referred to as COMB. Later comparisons were made using the different and somewhat more comprehensive bearing-shaft computer program described in reference 8. This program is hereinafter referred to as SHABERTH. The performance characteristics that are compared are the inner-race temperature, the outer-race temperature, and the bearing friction torque as converted to power consumption.

Combined Load Program (COMB)

Using COMB (ref. 6) to obtain bearing performance predictions requires, as part of the input data, the temperatures of the bearing housing and test shaft at both the oil-inlet and oil-outlet ends as well as an estimate of the volume percent of the bearing cavity that is occupied by the lubricant. The bearing cavity is the space between the races that is not occupied by the cage or the rolling elements. The values assumed for these variables can affect the predicted race temperatures.

Bearing-Shaft Program (SHABERTH)

Using SHABERTH (ref. 8) to predict the bearing performance also requires the input of an estimate of the lubricant volume in the bearing cavity, but, unlike the COMB program (ref. 6), the oil inlet-end housing and shaft temperatures are not required to be a fixed value. Therefore, these end temperatures were left floating, to be calculated by the thermal routines in the computer program.

More extensive input is required by SHABERTH than by COMB for a thermal analysis type calculation. The reason is that all the temperature nodes are fixed in the COMB thermal routines whereas the temperature nodes must be defined for SHABERTH with an allowable maximum number of 100. Since the main objective of the present calculations with SHABERTH was to compare the results using the two different traction models that are available with the program, a relatively simple thermal grid system was chosen, using only 17 nodes for the ball bearing. The model in the COMB program (ref. 6) uses 32 nodes. The traction models are outlined briefly in appendix A and explained in detail in reference 8. It should be noted here that because of the simple nodal system the lubricant flow rate is not included directly in the thermal calculations with SHABERTH. The COMB program, however, does use the flow rate directly.

RESULTS AND DISCUSSION

To effect a direct comparison of predicted and experimental bearing performance, the computer programs were run at the stated operating conditions of the bearings tested in reference 7. Essentially, these conditions consisted of three thrust loads (6672, 13 350, and 22 240 N (1500, 3000, and 5000 lb)), three speeds (12 000, 16 700, and 20 800 rpm), and two lubricant flow rates (3.8×10^{-3} and 8.3×10^{-3} m^3/min (1.0 and 2.2 gal/min)). The first calculations were done with the COMB program. Then the comparisons using SHABERTH were made.

Combined Load Program (COMB)

As a prerequisite to using COMB, the shaft and housing end temperatures must be arbitrarily chosen. In addition, the percent of lubricant in the bearing cavity must be assumed. To determine how the race temperatures vary with the assumed values of shaft and housing end temperature, the program was run for several sets of end temperatures at the 6672-newton (1500-lb), 12 000-rpm condition. The lubricant flow rate was 3.8×10^{-3} cubic meter per minute (1.0 gal/min). The lubricant volume in the bearing cavity was assumed to be 5 percent. The results of these calculations are shown in figure 2. The race temperatures are seen to vary linearly with the end temperatures. Furthermore, for a given end temperature, the inner race temperature was calculated to be higher than the outer race for these conditions.

To determine how the race temperatures and bearing power loss vary with the assumed value of percent lubricant in the bearing cavity, the program was run for several values of percent lubricant in the bearing cavity at the 6672-newton (1500-lb), 12 000-rpm condition. Based on previous experience, the end temperatures were arbitrarily assumed to be 28 and 42 kelvin degrees (50 and 75 Fahrenheit degrees) higher than the oil inlet temperature, or 436 K (325° F) at the housing and 422 K (300° F) at the shaft. The oil flow rate was 3.8×10^{-3} cubic meter per minute (1 gal/min). The results are shown in figure 3.

The race temperatures (fig. 3(a)) increased with increasing lubricant volume. This would be expected since the fluid drag on the balls would increase with the amount of liquid available. The oil volume range recommended by the authors of reference 6 was approximately 2 to 10 percent. Over this range the temperature change appears to be almost linear and rather small, about 5 kelvin degrees (10 Fahrenheit degrees) for the conditions calculated.

The bearing power loss (fig. 3(b)) increases with increasing lubricant volume in the bearing cavity. The rate of change is significant. The predicted bearing power loss increased by 100 percent over the small recommended volume range. The power loss varies almost linearly with lubricant volume up to a volume of 30 percent.

The program was run to determine the effect of thrust load on bearing race temperature and power loss using the shaft and housing end temperatures of 422 and 436 K (300° and 325° F) selected previously. The lubricant volume was set at 5 percent. Calculations were made for a lubricant flow rate of 3.8×10^{-3} cubic meter per minute (1 gal/min) and a shaft speed of 12 000 rpm. The results, compared with the experimental data of reference 7, are shown in figure 4. The predicted race temperatures (fig. 4(a)) are slightly low at the lower load and quite close at the higher load. Adjustments to the end temperatures at each point could have brought the calculated temperatures into almost exact agreement with the experimental data at each point. The predicted power loss (fig. 4(b)) is seen to be about one-half the measured value.

The effect of oil flow rate on race temperature and power loss was determined. The lubricant volume in the bearing cavity was set at 5 and 10 percent. The results for a 6672-newton (1500-lb) thrust load at 12 000 rpm are plotted in figure 5.

The predicted values of race temperatures (fig. 5(a)) are seen to decrease as the flow rate increases. The calculated values are reasonably close to the experimental values at the higher flow rates.

The experimental bearing power loss (fig. 5(b)) increases linearly with flow rate over this flow range. However, the calculated results predict the power loss to be almost constant with flow rate, even though it is likely that the volume percent would increase somewhat with increasing flow rate. The calculated power losses are quite low, especially at the higher flow rates.

The effect of shaft speed on race temperatures and power loss was determined and compared to experimental data. The results for a 6672-newton (1500-lb) thrust load are shown in figures 6 and 7 for two lubricant flow rates. Figure 7 also shows comparisons at a 22 240-newton (5000-lb) thrust load. The predicted temperatures (figs. 6(a) and 7(a)) do not increase with speed to the same degree as the experimental data. From these comparisons it becomes fairly obvious that the assumption of constant housing and shaft end temperatures where bearing speed is a variable is probably not realistic and can lead to substantial underestimates of bearing temperature. (A further discussion is presented in appendix B.)

The predicted power loss increases with speed (figs. 6(b) and 7(b)). The rate of increase appears to parallel the experimental data. However, the absolute values of predicted power loss were less than 50 percent of the experimental values.

Based on these comparisons it appears that COMB can produce acceptable estimates of temperature but relatively low values of bearing power loss. The program can be an excellent tool to compare the performance of bearings having different design or operating parameters. However, the program is limited in its present form in predicting absolute values of temperature and power loss without empirical modifications.

Based on previous experience (ref. 3), the program gives reasonably good predictions of kinematics for 120-millimeter-bore ball bearings under lubricated conditions. As a result, no effort was made to investigate further these dependent variables.

Shaft Bearing Program (SHABERTH)

Unlike COMB, SHABERTH does not require shaft and housing end temperatures as program input. These values are outputs of the program and are calculated based on system heat-transfer characteristics and bearing power loss. However, the percent of lubricant in the bearing cavity must be assumed. The COMB program contained a Newtonian lubricant traction model. The SHABERTH program exists as two versions, and each version incorporates a different traction model. These models, described in appendix A, are referred to as the NASA traction model and the SKF traction model after each of the organizations which developed the respective models. The two versions of the program are therefore called SHABERTH/NASA and SHABERTH/SKF. The calculations of the elastohydrodynamic (EHD) film thickness and contact traction forces are the only differences between the two versions (see refs. 8 and 10).

Calculations were made for the 6672-newton (1500-lb) thrust load case at 12 000, 16 700, and 20 800 rpm. The lubricant volume was set at 1.0 percent (the authors of ref. 8 recommend a maximum value of 2 to 3 percent for the program). The results, using both versions of SHABERTH, are shown in figure 8 compared to the experimental data with the oil flow rate of 3.8×10^{-3} cubic meter per minute (1 gal/min). The previous results using the COMB program are also shown for comparison. The calculated values of inner and outer race temperatures are lower than the experimental data for both versions of the SHABERTH computer program, with the NASA model resulting in slightly higher temperatures than the SKF model. There was no really significant difference between the SHABERTH temperature predictions and those of the COMB program. The calculated bearing power loss (fig. 8(c)) is very close to the measured value for the NASA model and again slightly higher than with the SKF model. The SHABERTH program predicted values of power loss significantly closer to the measured values than did the COMB program.

To compare with the experimental data taken at the oil flow rate of 8.3×10^{-3} cubic meter per minute (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 percent lubricant should increase with increased flow rate. The results are shown in figure 9. The previous results from the COMB program (obtained from fig. 7) are also shown for comparison purposes. The calculated race temperatures are seen to be reasonably close to the experimental data. The calculations with the NASA model again produced temperatures that were slightly higher than with the SKF model. The bearing power losses calculated with the NASA model resulted in fairly good agreement with the experimental data. The values were slightly higher than those calculated with the SKF model and, also, were about a factor of 2 times the values obtained with the COMB program.

A comparison of figures 8 and 9 shows that the experimental temperature data went down 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 in 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 using the NASA version at one condition for several other values of percent lubricant. The results for the 6672-newton (1500-lb) thrust load, 16 700 rpm shaft speed are shown in figure 10. 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 8 and 9 indicates that for these operating conditons the values of percent lubricant in the bearing cavity assumed for the comparison calculations were reasonably correct.

Since the race temperatures and bearing losses do agree as shown when using SHABERTH, it is interesting to note the values calculated for the end temperatures with this program and then to compare them with those assumed for input in the COMB program. This comparison is shown in figure 11 with the values used in the calculation for the 6672-newton (1500-lb) thrust load, 8.3×10^{-3} cubic meter per minute (2.2 gal/min) oil flow rate data. It can be seen that the end temperatures assumed for the COMB program were, in general, close to the mean of those calculated by the SHABERTH program. Furthermore, those calculated with the NASA model were slightly higher than those calculated with the SKF model. Both models resulted in a difference between the housing and shaft end temperatures, particularly at the higher speeds. Values obtained from equation (B1) are also shown. Note that while the equation temperatures are somewhat higher, the trend with speed agrees very well with the SHABERTH calculated values.

The SHABERTH program (as noted in ref. 8), when using the NASA version, uses a modification to the film thickness calculation as proposed by Loewenthal, et al. in reference 11. Therefore, the lubricant film thicknesses as calculated by both versions of the program for the same 6672-newton (1500-lb) load, 8.3×10^{-3} cubic meter per minute (2.2 gal/min) oil flow case as before were plotted as shown in figure 12. The values calculated by the COMB program are also shown for comparison. The film thicknesses calculated by SHABERTH are considerably lower than those from COMB (by a factor of 2) at each speed condition. The values with the SKF model are about 50 percent higher than those from the NASA version. It is possible that this difference in film thickness accounts for most of the difference between the two versions in the power loss calculation. The SKF model is based on fraction of asperity contact, which in turn would be dependent on the film thickness. Also, the high values of film thickness indicated for the COMB program would partly account for the low values of calculated bearing power loss noted for that program.

Since higher bearing loads should result in thinner films, the two versions of SHABERTH were compared again at the 13 350- and 22 240-newton (3000- and 5000-lb) load conditions. The results are shown in figures 13 and 14, where the inner race temperature and bearing power loss are plotted as functions of bearing thrust load for shaft speeds of 12 000 and 16 700 rpm. The SKF version values were again lower than those values obtained with the NASA version; however, more importantly, the SKF version shows little change in temperature or power loss with an increase in thrust load. The NASA values do increase with thrust load and give a reasonably close prediction of the experimental data. The combination of smooth bearing surfaces and thicker EHD films result in very little asperity contact with the SKF version for all the conditions calculated.

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, it can be speculated 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.

The largest unknown quantity of the input data required for SHABERTH is the volume percent of lubricant in the bearing cavity. The values chosen for these calculations were in the range suggested by the authors of reference 8. The reason for the difference in range of lubricant volume percent between COMB and SHABERTH is not clear.

As stated previously, since the SHABERTH-calculated power loss did compare very well with the experimental data, it can be concluded that the values of percent lubricant in the bearing cavity used are correct for this program. However, since how these values vary with oil flow and/or shaft speed is still not clear, work needs to be performed in this area.

SUMMARY OF RESULTS

Computer programs were used to predict bearing inner- and outer-race temperatures and friction power loss over a range of operating conditions, and the results were compared with experimental data obtained previously. The 120-millimeter bore bearings were operated at thrust loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb) and at shaft speeds of 12 000, 16 700, and 20 800 rpm with jet lubrication flow rates of 3.8×10^{-3} and 8.3×10^{-3} cubic meter per minute (1.0 and 2.2 gal/min). The oil inlet temperature was maintained constant at 394 K (250° F). The following results were obtained:

1. The combined load program (COMB) can predict reasonable bearing race temperatures with proper input data, but this program severely underestimates bearing power loss.

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

3. The bearing-shaft program predicted slightly higher bearing power losses using the NASA version than when using the SKF version.

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.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, January 12, 1978,
505-04.

APPENDIX A

TRACTION MODELS USED IN COMB AND SHABERTH COMPUTER PROGRAMS

The following is a brief description of the traction models used in the computer programs utilized in this report. A detailed explanation of each model is given in the respective reference cited. In all cases the tractive force is determined by calculating values over a small elemental area and then summing the elemental values over the entire contact area.

COMB Program (ref. 6)

The model in the COMB program is essentially Newtonian viscous fluid friction. A shear stress τ is calculated from

$$\tau = \eta \frac{v}{h} \tag{A1}$$

where

η lubricant viscosity, $N\text{-}sec/m^2$ ($lb\text{-}sec/in^2$)

h contact film thickness, m (in.)

v sliding velocity, m/sec (in/sec)

The value of shear stress is influenced greatly by the value of η. The value of η is determined by both temperature and pressure. The COMB program used an equation of the form

$$\eta = s_1 \exp\left(\sqrt{s_2 + s_3 P}\right) \tag{A2}$$

where P is Hertz pressure, and s_1, s_2, and s_3 are constants to relate the viscosity with pressure for a given temperature. The film thickness was calculated from an expression formulated by Archard and Cowking (ref. 12).

SHABERTH/NASA Program (ref. 8)

The NASA version of the SHABERTH program uses the model proposed by Allen et al. (ref. 13) wherein the lubricant shear stresses are represented in two regimes.

At low shear stresses, the same equation (A1) applies, with the viscosity calculated from

$$\eta = \eta_o e^{\alpha P} \tag{A3}$$

where

η_o viscosity at atmospheric pressure, $N\text{-sec/m}^2$ (lb-sec/in^2)

α pressure-viscosity coefficient, m^2/N (in^2/lb)

P Hertz pressure, N/m^2 (lb/in^2)

At higher values of shear stress, the following equation is used:

$$\tau = fP \tag{A4}$$

where

f lubricant factor

P contact pressure, N/m^2 (lb/in^2)

The lubricant factor f is used to limit the shear stress τ at high pressures and high shear rates. Essentially, equation (A4) is used when the stress values of equation (A1) exceed those of equation (A4).

The film thickness required for equation (A1) is calculated according to the modifications suggested by Loewenthal, et al. (ref. 11).

SHABERTH/SKF Program (ref. 10)

The SKF version of the SHABERTH program uses a traction model that accounts for lubricant shear and asperity interaction. The portion of the contact load carried by the asperities is determined according to Tallian (ref. 14). The film shear coefficient is calculated according to Chiu (ref. 15). The traction F then is calculated from

$$F = Q_{ehd}\mu_{ehd} + Q_{asp}\mu_{asp} \tag{A5}$$

$$Q = Q_{ehd} + Q_{asp} \tag{A6}$$

where

Q total normal load, N (lb)

Q_{ehd} load carried by EHD film, N (lb)

Q_{asp} load carried by asperities, N (lb)

μ_{ehd} friction coefficient that develops from lubricant shear

μ_{asp} asperity friction coefficient

The film thickness is calculated according to Archard and Cowking (ref. 12). These two calculations, for the film thickness and contact traction, are the only differences between the NASA and SKF versions of the SHABERTH program.

APPENDIX B

USE OF COMB PROGRAM WITH VARYING END TEMPERATURES

From figures 6 and 7 it may be reasonably concluded that it is not valid to input constant end temperatures when the shaft speed is increased for this test configuration, wherein the ends of the shaft and housing are fairly close to the bearing. Furthermore, from these figures it is seen that the inner-race temperature can become higher than the outer-race temperature. Therefore, it is probably not valid to assume that the housing end temperatures should always be higher than the shaft end temperatures. To predict an end temperature that would vary with shaft speed, the following equation was empirically formulated:

$$\left.\begin{aligned} T_{end} &= \left(\frac{DN}{10^6} - 0.5\right) \times 56 + T_{oil} \quad (K) \\ T_{end} &= \left(\frac{DN}{10^6} - 0.5\right) \times 100 + T_{oil} \quad (^{o}F) \end{aligned}\right\} \tag{B1}$$

where T_{end} is either the housing or shaft end temperature, T_{oil} the temperature of the lubricating oil at the inlet to the bearing, and DN the product of the bearing bore in millimeters times the shaft speed in rpm.

End temperatures as calculated from equation (B1) were then used as input, and the COMB computer program run for three shaft speeds (12 000, 16 700, and 20 800 rpm) and thrust loads (6672 and 22 240 N (1500 and 5000 lb) with a lubricant flow rate of 8.3×10^{-3} cubic meter per minute (2.2 gal/min). The volume of the lubricant in the cavity was set at 10 percent. This set of data was chosen since the test bearings had been operated to 20 800 rpm with the higher oil flow rate. The comparison of the calculated values with the experimental data is presented in figure 15. The predicted temperatures are reasonably close to the experimental data over the test range, generally within 5 percent. However, the calculated values of bearing power loss are consistently low compared with the measured data. The difference was usually a factor of 2 to 3. Nevertheless, the trends of the calculated values are correct.

It should be noted here that in all cases the end temperatures input to the computer program were higher than the race temperatures the program calculated. This would imply heat transfer from the end of the housing and shaft toward the respective bearing races. This is not possible, and the fact that high end temperatures are required as input to correctly predict the race temperatures implies that the bearing power losses as calculated by the program are too low. This implication is verified by the aforementioned comparison of predicted and measured values of bearing power loss.

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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)($^\circ$F))	2340 (0.54)
Thermal conductivity at 477 K (400° F), J/(m)(sec)(K), (Btu/(hr)(ft)($^\circ$F))	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

[a]From reference 7.

TABLE II. - 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, μcm (μin.)	2.5 (1)
Raceway surface finish, μcm (μin.)	5 (2)

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

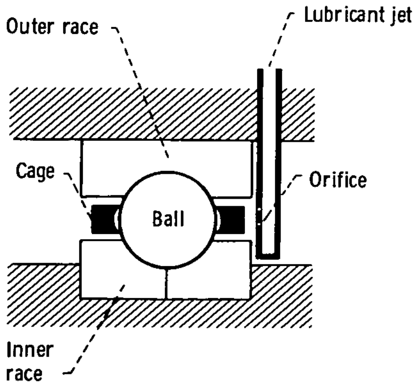


Figure 1. - Bearing lubrication. Number of jets, two per bearing; dual orifice; inner-land riding cage.

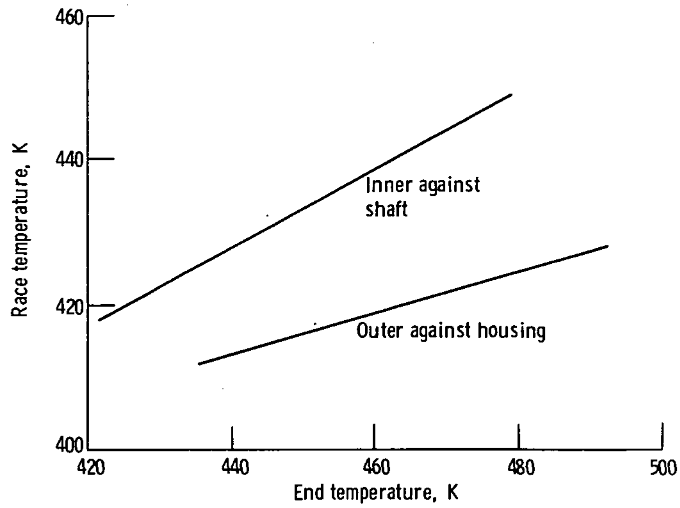
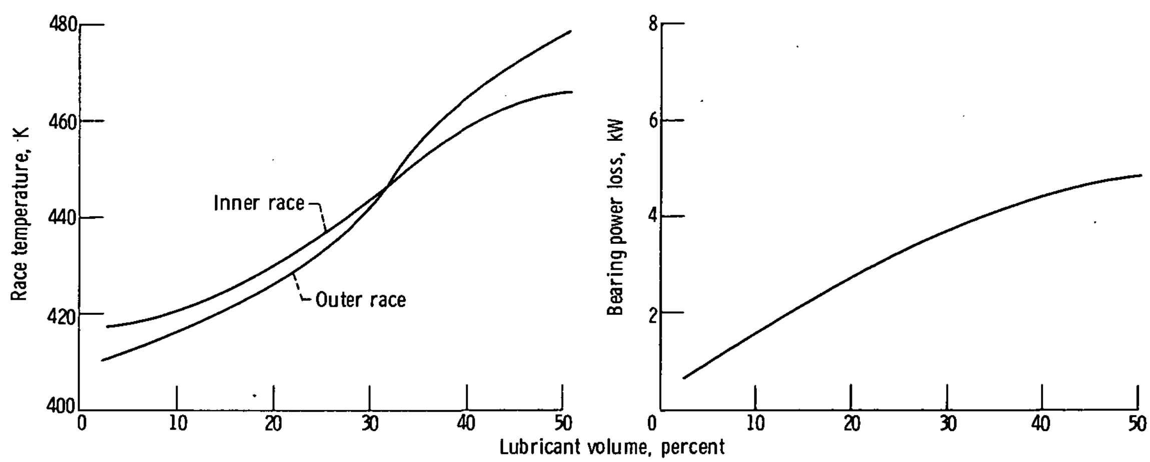


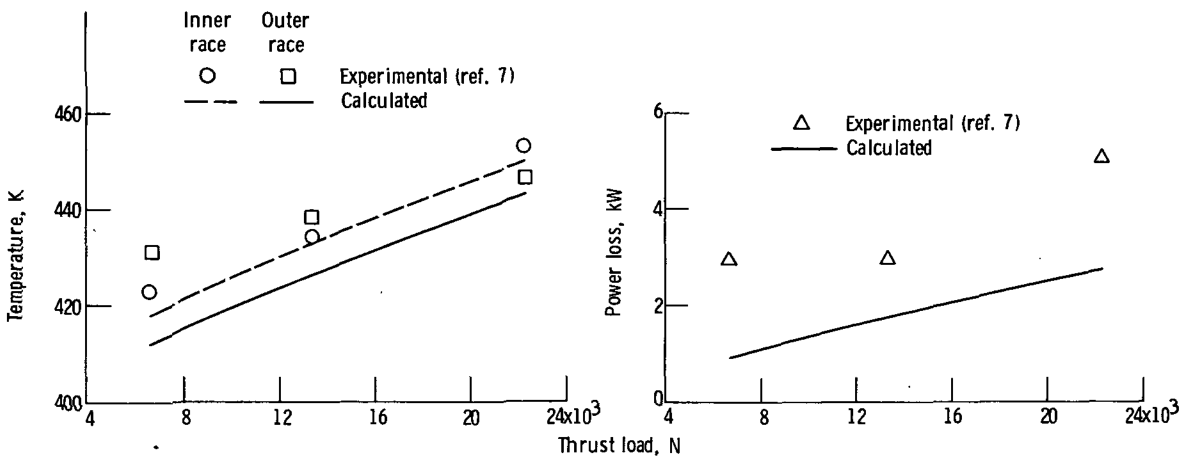
Figure 2. - Calculated inner- and outer-race temperatures as functions of shaft and housing end temperatures, respectively. Thrust load, 6672 newtons (1500 lb); shaft speed, 12 000 rpm; lubricant flow rate, 3.8×10^{-3} cubic meter per minute (1 gal/min); volume of lubricant, 5 percent. COMB program.



(a) Temperature.

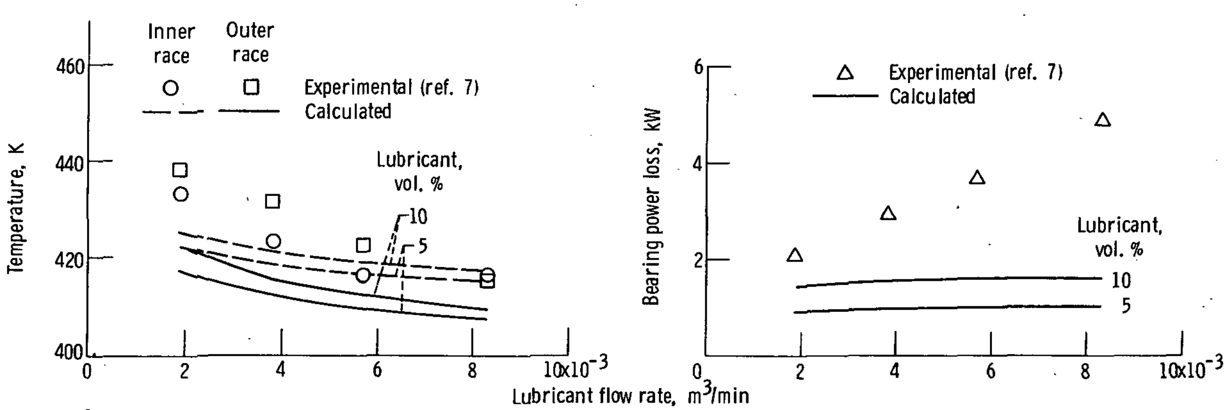
(b) Bearing power loss.

Figure 3. - Calculated values of bearing operating characteristics as functions of lubricant volume fraction. Thrust load, 6672 newtons (1500 lb); shaft speed, 12 000 rpm; lubricant flow rate, 3.8×10^{-3} cubic meter per minute (1 gal/min); housing/shaft end temperatures, 436/422 K. COMB program.



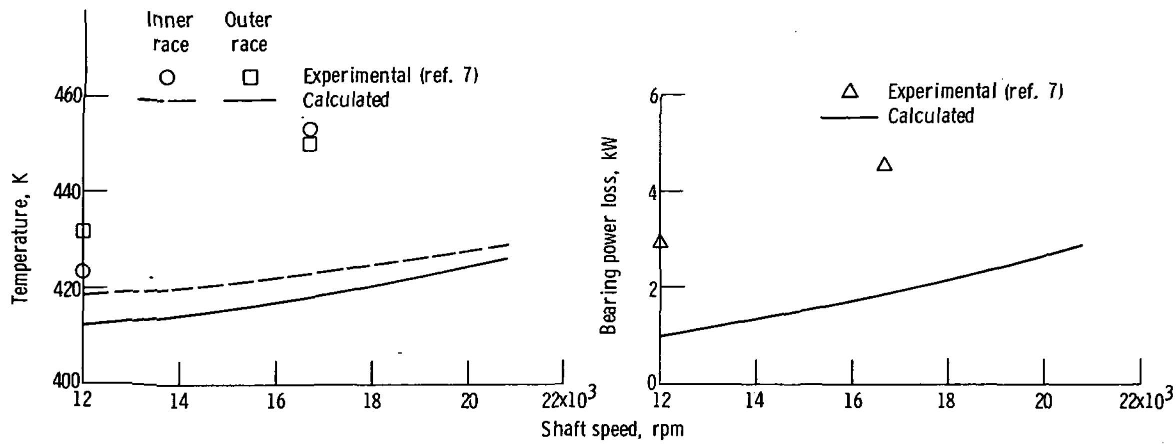
(a) Race temperature. (b) Bearing power loss.

Figure 4. - Comparison of calculated and experimental values of bearing operating characteristics as functions of thrust load. Shaft speed, 12 000 rpm; lubricant flow rate, 3.8×10^{-3} cubic meter per minute (1 gal/min); volume of lubricant, 5 percent; housing/shaft and temperatures, 436/422 K. COMB program.



(a) Race temperature. (b) Power loss.

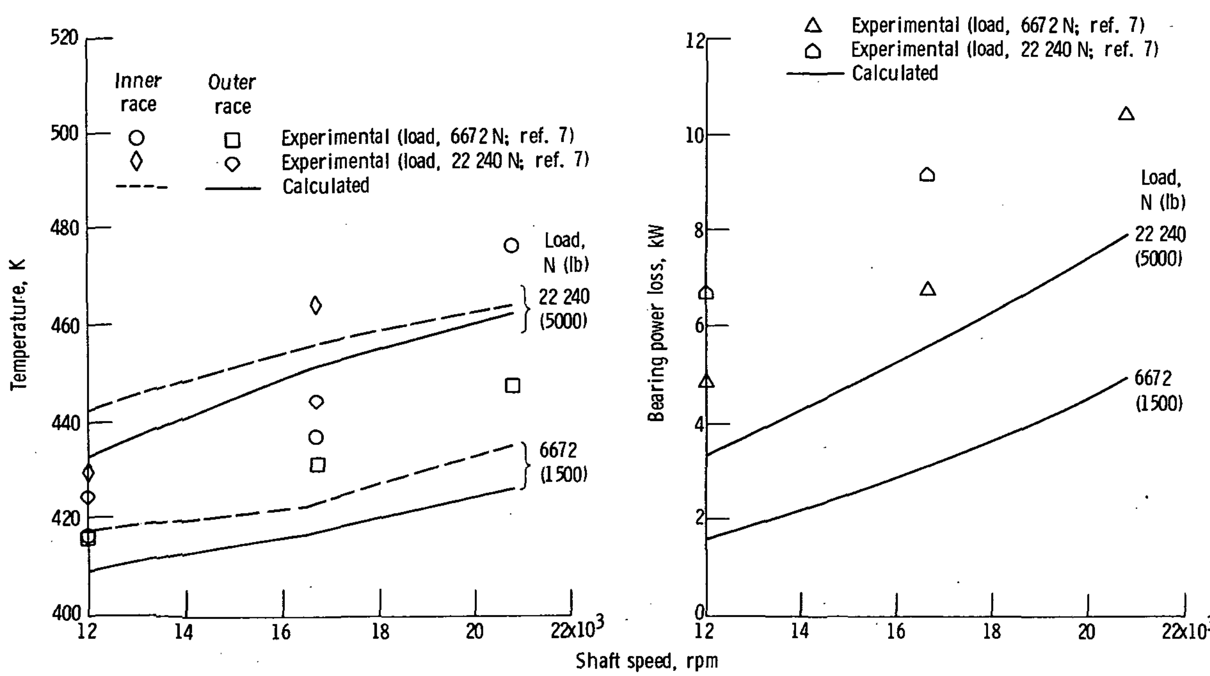
Figure 5. - Comparison of calculated and experimental values of bearing operating characteristics as functions of lubricant flow rate. Thrust load, 6672 newtons (1500 lb); shaft speed, 12 000 rpm; housing/shaft end temperatures, 436/422 K. COMB program.



(a) Race temperature.

(b) Power loss.

Figure 6. - Calculated and experimental values of temperature and power loss as functions of shaft speed. Thrust load, 6672 newtons (1500 lb); lubricant flow rate, 3.8×10^{-3} cubic meter per minute (1 gal/min); volume of lubricant, 5 percent; housing/shaft end temperatures, 436/422 K. COMB program.



(a) Race temperatures.

(b) Power loss.

Figure 7. - Comparison of calculated and experimental values of bearing operating characteristics as functions of shaft speed for two thrust loads. Lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 10 percent. COMB program.

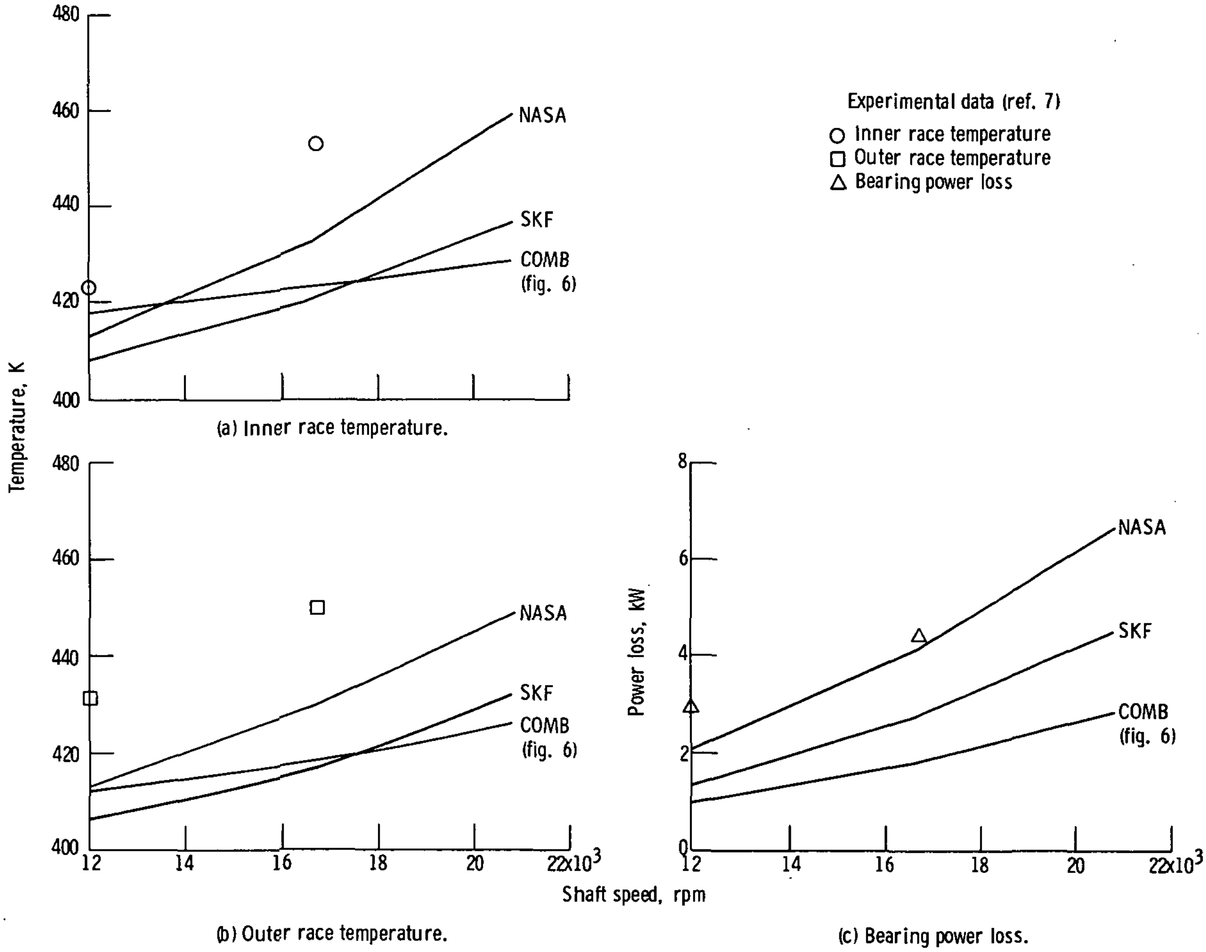
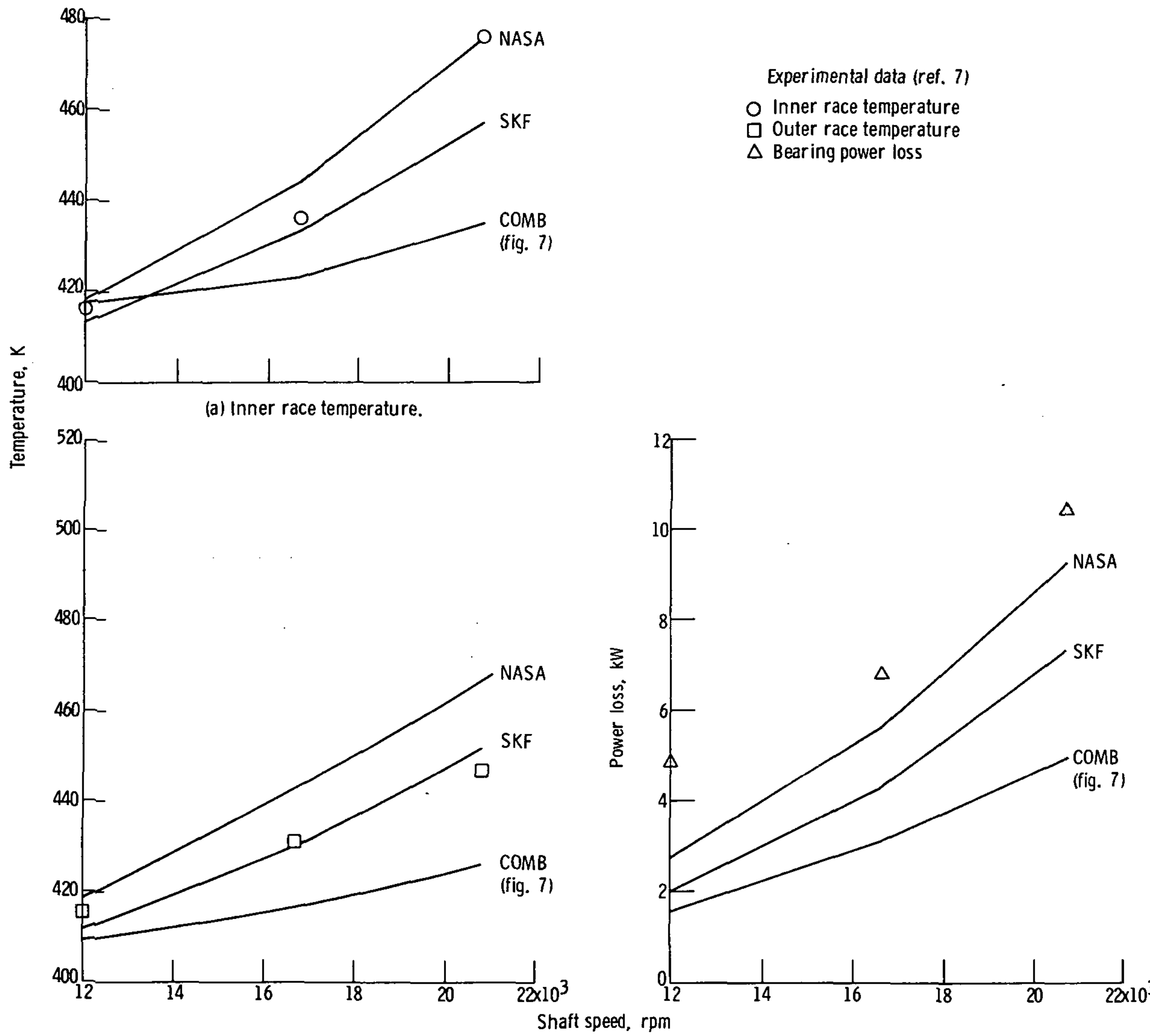
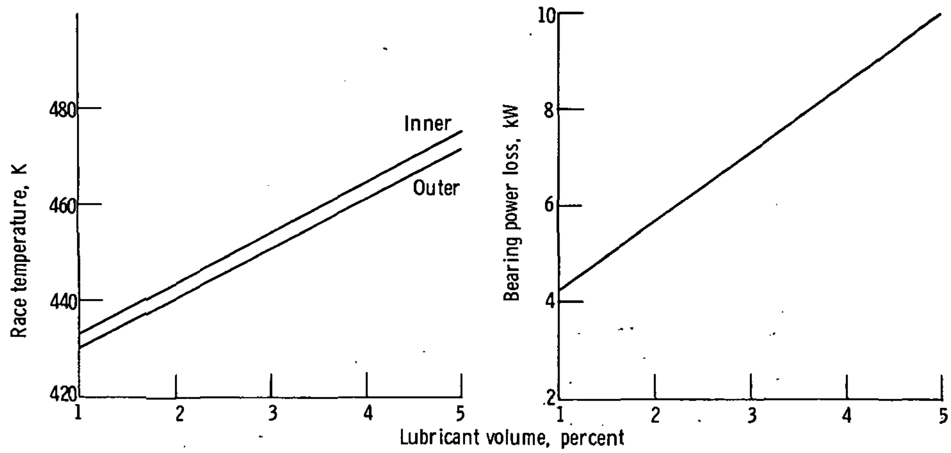


Figure 8. - Comparison of calculated and experimental bearing operating characteristic data as functions of shaft speed using two versions of SHABERTH computer program. Thrust load, 6672 newtons (1500 lb); lubricant flow rate, 3.8×10^{-3} cubic meter per minute (1 gal/min); volume of lubricant, 1.0 percent.



(b) Outer race temperature. (c) Bearing power loss.

Figure 9. - Comparison of values of calculated and experimental bearing temperatures and power loss as functions of shaft speed using two versions of SHABERTH computer program. Thrust load, 6672 newtons (1500 lb); lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 2.0 percent.



(a) Race temperatures.

(b) Power loss.

Figure 10. - Calculated bearing operating characteristics as functions of volume percent lubricant in bearing cavity using SHABERTH computer program. Thrust load, 6672 newtons (1500 lb); shaft speed, 16 700 rpm.

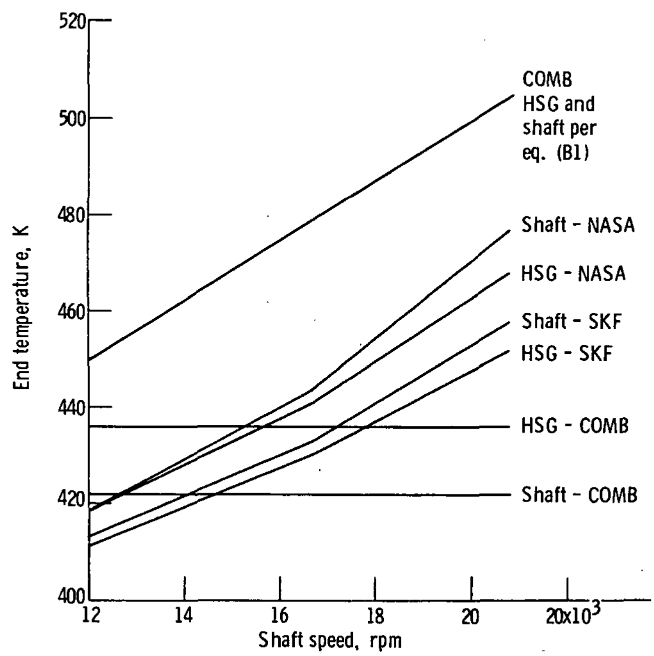


Figure 11. - Comparison of end temperatures as calculated by two versions of SHABERTH with values assumed as input to COMB and with values from equation (B1). Thrust load, 6672 newtons (1500 lb); lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 2 percent for SHABERTH.

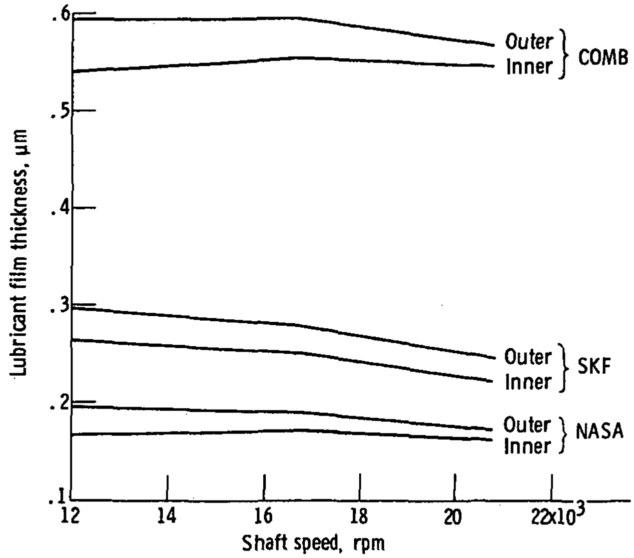
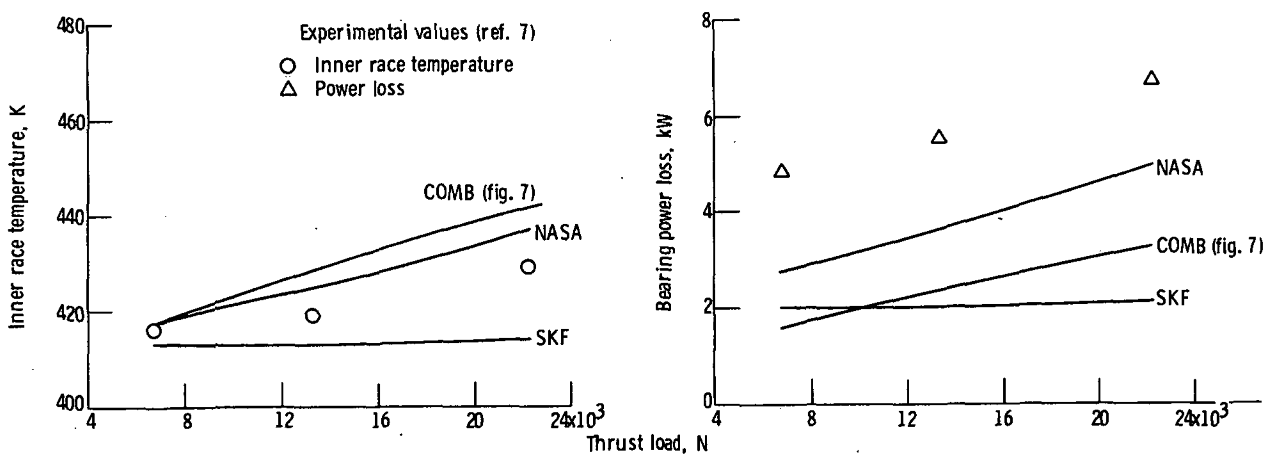


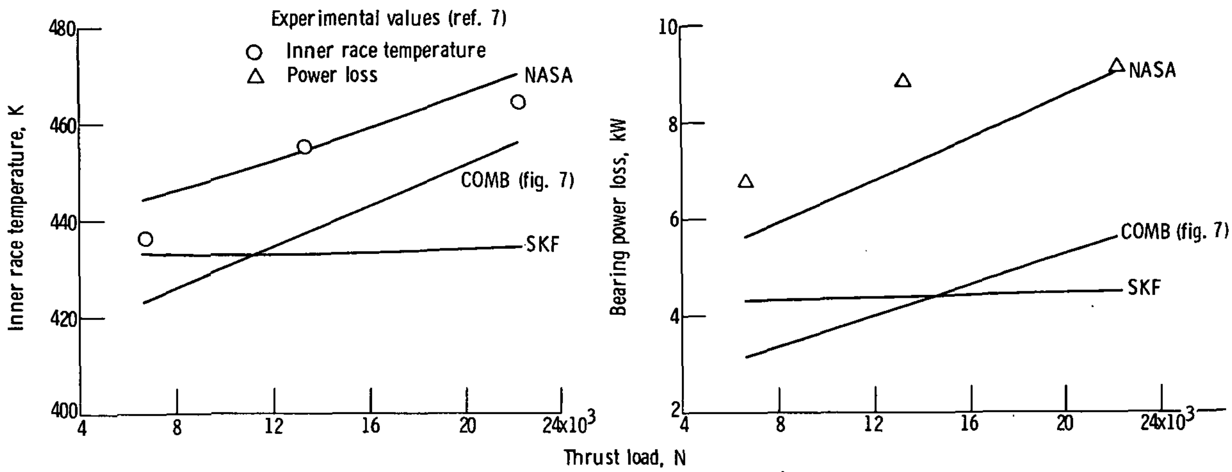
Figure 12. - Values of elastohydrodynamic (EHD) film thickness at inner- and outer-race-ball contacts as function of speed as calculated by COMB and two versions of SHABERTH computer program. Thrust load, 6672 newtons (1500 lb); lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 10 percent for COMB, 2 percent for SHABERTH.



(a) Inner race temperatures.

(b) Power loss.

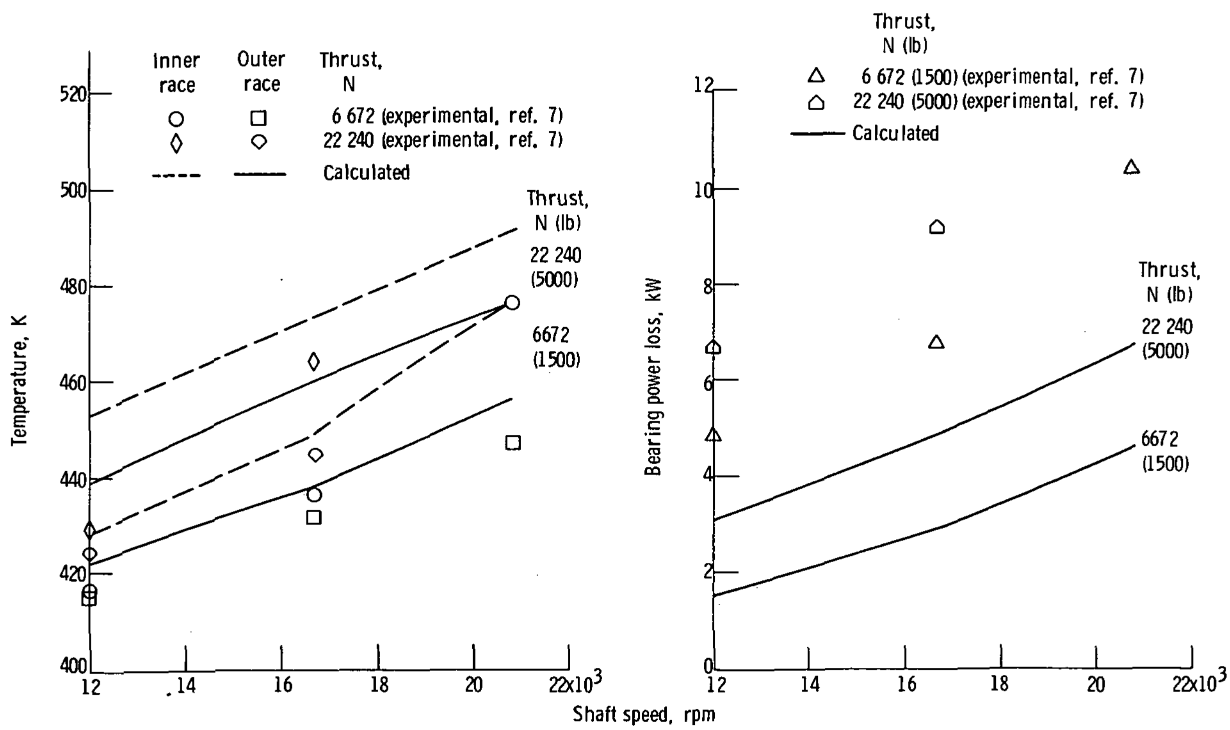
Figure 13. - Comparison of calculated and measured values of race temperatures and bearing power loss as functions of thrust load using two versions of SHABERTH. Shaft speed, 12 000 rpm; lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 2 percent.



(a) Inner race temperatures.

(b) Power loss.

Figure 14. - Comparison of measured and calculated values of bearing operating characteristics as functions of thrust load using two versions of SHABERTH. Shaft speed, 16 700 rpm; lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 2 percent.



(a) Race temperatures.

(b) Power loss.

Figure 15. - Comparison of calculated and experimental values of bearing operating characteristics as functions of shaft speed for two thrust loads. Lubricant flow rate, 8.3×10^{-3} cubic meter per minute (2.2 gal/min); volume of lubricant, 10 percent; housing/shaft end temperatures per equation (B1). COMB program.

1. Report No. NASA TP-1196	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle PREDICTED AND EXPERIMENTAL PERFORMANCE OF JET-LUBRICATED 120-MILLIMETER-BORE BALL BEARINGS OPERATING TO 2.5 MILLION DN		5. Report Date April 1978
		6. Performing Organization Code
7. Author(s) Harold H. Coe and Erwin V. Zaretsky		8. Performing Organization Report No. E-9288
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135		10. Work Unit No. 505-04
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		13. Type of Report and Period Covered Technical Paper
		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

Bearing inner- and outer-race temperatures and friction power losses were calculated using two computer programs. The values obtained were compared with previously reported experimental data for 120-mm-bore bearings which operated at thrust loads to 22 240 N (5000 lb), shaft speeds to 20 800 rpm, and with two lubricant flow rates. One program severely underestimated the power loss, while the other, called SHABERTH, provided a good prediction of both race temperatures and power losses.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Ball bearing; Computer program; High speed; Bearing analysis; Jet lubrication; Experimental data; EHD	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	26	A03

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