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CARBON-PHENOLIC CAGES FOR HIGH-SPEED BEARINGS

Part III – Development of Numerical Models for Heat Generation and Temperature Prediction in Lightly Lubricated Bearings

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NOTICE

THIS TECHNICAL REPORT HAS BEEN REVIEWED AND IS APPROVED FOR PUBLICATION.

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14. ABSTRACT

This is the third part of a three-part series of reports by AFRL/PRTM to investigate carbon-phenolic bearing cages in high-speed, lightly lubricated bearings. This portion describes the bearing and thermal analysis models. A process has been developed to numerically investigate operating temperatures of a rolling element bearing which combines capabilities of the Advanced Dynamics of Rolling Elements (ADORE) software with finite element thermal analysis. Initial results are presented that describe experimental and predicted transient and steady-state temperature responses for a 30 mm ball bearing duplex pair operating with only a light coating of Pennzane® lubricant for two different rolling element materials (Si_3N_4, 52100 steel) in a bearing with 52100 steel races operating at a speed of 10,000 rpm using a conventional cotton-phenolic cage. Although preliminary result agreement appeared excellent, further analysis led to model modifications to account for the potential of thermal expansion to cause an increase or decrease in bearing load. This change in load can be significant for the relatively light load on a satellite bearing. The effect of modifying the shape of the traction curves used in the bearing analysis model has also been investigated.

15. SUBJECT TERMS

bearings, composite cages, control moment gyroscope

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Table of Contents

List of Figures

List of Figures (concluded)

List of Tables

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1. Introduction

Numerically investigating heat generation for specific bearing environment, geometry, material, and lubricant conditions provides the opportunity to design bearings for improved thermal performance. This is critical for bearing conditions where heat generation can lead to temperatures too extreme for reliable operation. The capability to determine heat generation in individual bearing contacts exists within the research level Fortran-90 software package, Advanced Dynamics of Rolling Elements (ADORE), developed by Gupta (1984). Previous investigations have demonstrated the potential for using ADORE to predict heat generation as a function of contact location (rolling element-outer race, rolling element-inner race, rolling element-cage pocket, and cage land-outer race) for several operating conditions. The code uses friction data, bearing geometry, material properties, shaft speed, and bearing load as inputs to solve the differential equations of motion for each bearing element in a 6 degree-of-freedom system. Applied forces and moments are computed from bearing element interactions, and the differential equations are integrated numerically with prescribed initial conditions. The Hertzian stress calculation (Timoshenko, 1951) is used to calculate the stress and contact area as a function of material properties and applied load. The ability to treat internal transient forces, such as those resulting from cage-race collisions, is critical to determination of heat generation. Absolute velocities of two elements in contact are used to determine slip, defined as relative sliding velocity normal to the contact load. A traction-slip relation input into the program for the specific lubricant and operating conditions is used to determine a traction coefficient, which, multiplied by the normal load, gives the traction force. Traction forces at the ball/race interface greatly influence orbital accelerations of the rolling elements, which determine the extent of ball/cage collisions and cage instabilities. ADORE solutions include rolling element positions, accelerations, loads, and heat generation.

Forster and Givan (1999) used ADORE in one of the first investigations of heat generation in vapor lubricated bearings. They outlined an approach consisting of friction testing in concentrated contacts, experimental bearing torque measurements to calculate power loss at high speeds, and comparison with bearing performance predictions. Brown and Forster (2000, 2002) further exercised the ADORE software to develop thermal boundary conditions for use in ANSYS finite element thermal analyses to investigate bearing operating temperatures under mist lubrication conditions and the thermal performance benefits of using a lightweight carbon-carbon cage to reduce centrifugal loading in 30-mm bore bearings. Brown et al. (2002) accomplished preliminary investigations of the potential to use ADORE and ANSYS for lightly lubricated bearing analysis. Much of the current report is taken directly from this reference. Detailed thermal analysis of the heat generation requires geometry information for the supported shaft and bearing housing along with lubricant and environment conditions.

As part of the current NRO-sponsored effort, innovative bearing designs are currently being investigated that use coated race components, lightweight carbon composite cage materials, and high performance synthetic lubricants to allow rotating component performance of devices to be extended to high speeds and loads with reduced frictional loss from cage interactions (Sanders et al., 2000, Forster et al., 2001). Thermal modeling of novel bearing systems is being accomplished to investigate thermal performance benefits of reduced heat generation and improved heat dissipation. The ANSYS finite element heat transfer analysis can be applied to specific environments by considering convection heat transfer modes and including heat transfer to the surrounding structural components via conduction and radiation. Validated analytical results can be particularly beneficial in determining the speed limits of traditional phenolic cages based on thermal performance. Results may also lead to tailoring of cage properties for high-speed bearing applications.

Comparisons of numerical bearing analysis predictions with AFRL/PRTM experimental results have been accomplished. Baseline investigations used a 30-mm bore duplex pair fabricated from 52100 steel with a cotton-phenolic cage. To lubricate the bearing, cages have been saturated with lubricant that is then dispersed to the bearing during operation by the centrifugal force from rotation. While this system has been adequate to date for relatively low speeds (10,000 rpm), problems may occur in applications where rotor speeds are increased by 2 - 3X. This speed increase will greatly affect bearing cage issues as the heat generation from the cage increases as a second order effect due to centrifugal loading of the cage land and increased ball-cage collision forces. An area of particular concern is the localized heating of conventional cotton-phenolic bearing cages at these interfaces due to poor heat transfer characteristics. It will be difficult for these cages to dissipate increased heat loads, and the net result will be localized hot spots that may compromise the cage integrity and degrade the lubricant. Testing has then been accomplished using a carbon-phenolic cage. The thermal analysis indicated that these cages would not be as susceptible to hot spots. The final series of testing investigated CRU-20 bearing races with Si_3N_4 ceramic rolling elements using both cotton-phenolic and carbon-phenolic cages.

As the AFRL/PRTM experiments use minimal lubrication, a technique that reduces churning effects and removes the primary mode for heat rejection in conventional bearings, the results have provided an excellent opportunity to accomplish bearing thermal performance validation. Friction data available from AFRL/PRTM testing of the cotton-phenolic and carbon-phenolic material has been used in the ADORE bearing analyses. The experimental methods used and the measured friction data are described in Part I of this series of reports. Modifications to the predicted/measured friction values have been investigated as part of the numerical analysis. The experimentally measured bearing torque has been used to determine heat generation for comparison with the analytical prediction for total bearing power loss. Experimental and predicted transient and steady-state temperature responses for the tested 30-mm bore bearing duplex pair operating with only a light coating of Pennzane® lubricant (Sanders et al., 2000) are presented for two different rolling element materials (Si_3N_4, 52100 steel) operating at a shaft speed of 10,000 rpm. These results demonstrate agreement of the numerical predictions and experimental results. Although models used in the preliminary comparisons have been modified slightly, the results indicate the potential of the analysis process to improve bearing design through the evaluation of thermal conditions and determination of cooling requirements for future bearing system designs. Results describing model comparisons with the final series of bearing tests are considered to be the most representative of the tested conditions. These experiments also used a light coating of Pennzane® lubricant on bearings with CRU-20 races, Si_3N_4 balls, and either a cotton-phenolic or a carbon-phenolic cage. Both steady-state and transient results are presented.

2. Experimental Design

The capability to experimentally measure torque and temperature for 206 size bearings under lightly lubricated conditions exists within the AFRL/PRTM laboratories. The high-speed test rig described by Forster and Givan (1999) has been modified for conversion to duplex pair testing. A more complete description of the experimental setup is provided in Part II of this series of reports. The test bearings are mounted on a shaft driven by an integral air turbine. In the initial series of testing, the bearing load is due to the designed axial preload of 270N (65 lb) with the duplex pair flush mounted. Following initial test result analysis, it was determined the potential for change in load due to thermal expansion could be minimized by incorporating a spring-load mechanism rather than relying on the clamping load. Torque measurement is accomplished using a proximity probe to measure the displacement of a rod connected to the outer race housing and restraining the rotation of the outer races through a spring. Displacement of the rod corresponding to an applied torque has been calibrated by hanging known masses at a specific rod location and recording the displacement (voltage) measurement. This torque measurement system has worked well with the tested duplex pair configuration. Further modifications are being considered to damp the vibrations of this measurement method to improve response at higher shaft speeds. Type K thermocouples are mounted between the outer race and the housing and on the housing chamber. A slip ring for inner race temperature measurement has also been installed but has not provided the desired accuracy especially for speeds greater than 10,000 rpm. A telemetry system has been ordered to provide improved capability for temperature measurement of the inner race and shaft. Mounting of the duplex pair required the design and fabrication of a new nose piece, duplex bearing holder and miscellaneous installation/removal fixtures. Various test hardware and schematics are shown in Figures 1 - 5.

Figure 1. Duplex Bearing Pair (52100 Steel Races, Cotton-Phenolic Cage, Si_3N_4 Rolling Elements)

Figure 2. Shaft Nose Piece where Bearings are Mounted and Portion of Outer Race Housing

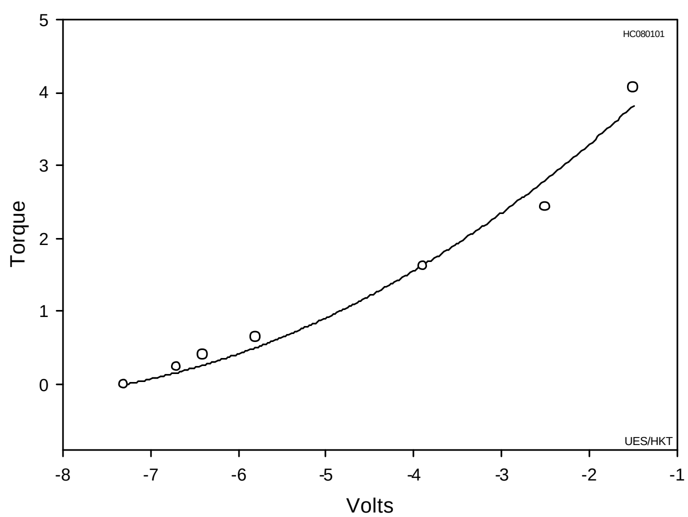

Figure 3. Calibration Curves Developed for Torque Measurement Based on Displacement

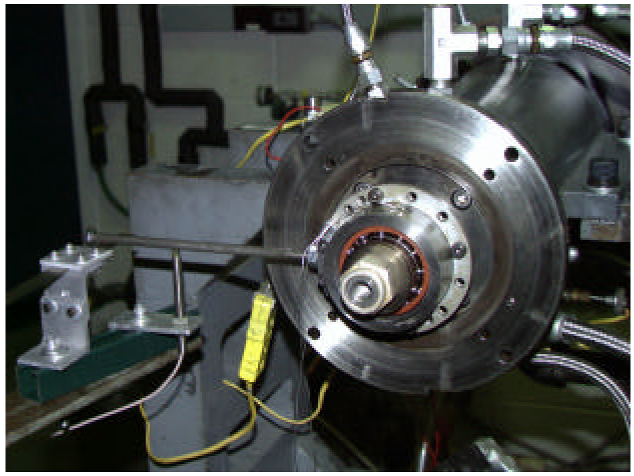

Figure 4. View of High-Speed Test Rig with Bearings Mounted and Proximity Probe

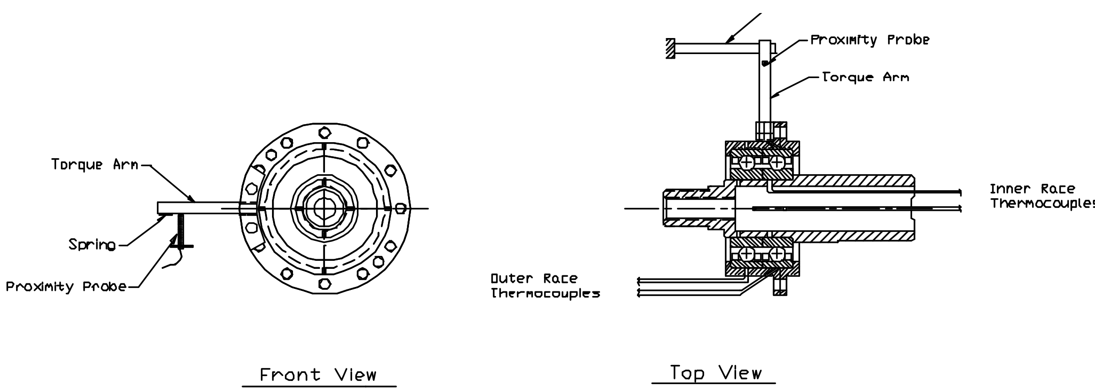

Figure 5. Schematic of Duplex Pair Flush Mounting and Torque Measurement System

3. Numerical Model Development (ADORE and ANSYS)

A numerical modeling process for bearing analysis has been developed which can be used to illustrate thermal performance benefits of using specific cage, ball, or raceway material with a given lubricant. This process consists of use of ADORE 4.0 bearing models to develop thermal boundary conditions for the ANSYS finite element analysis software package. This method provides the capability to evaluate improved thermal management designs. Geometry for the experimentally tested duplex bearing (Table 1) has been used to model a single bearing in ADORE with the specified loading for shaft speeds of interest.

Table 1. Bearing Geometry
Class 206, ABEC 7, Single Outer Land Guided Cage

Number of balls	11		
Contact angle	15°	Cage outer diameter (in)	2.049
Outer race curvature factor	0.5175	Cage inner diameter (in)	1.800
Inner race curvature factor	0.5300	Cage land clearance (in)	0.011
Pitch diameter (in)	1.81	Cage pocket clearance (in)	0.016
Ball diameter (in)	0.375	Cage width (in)	0.590
Axial preload (lb)	65	Radial load (lb)	5.000

Previous work by Brown (2001) described cage imbalance as a critical factor for bearing heat generation. For the purpose of the current investigation, a value of 0.20 g-cm has been assumed for the 206 size bearing. Churning effects can also be a significant parameter affecting bearing heat generation, especially for the fully flooded lubrication condition. However, churning effects should be small for the lightly lubricated condition, and they have been omitted in this study. Other primary factors considered in the ADORE analyses are the bearing materials and the lubricant/material traction data. The models have used constant material properties, as listed in Table 2. This has been considered adequate for the relatively small steady-state temperature increases at 10K rpm. 52100 tool steel, M50, or CRU-20 have been used as the bearing race material and either 52100 steel or Si_3N_4 ceramic for the rolling elements. The cage has been modeled as either a cotton-phenolic or carbon-phenolic composite. The finite element thermal analysis uses T15 steel properties for both the shaft and the bearing housing.

Table 2. Material Properties of Bearing Components

Material	CRU-20	T15	M50	52100	Si_3N_4	Cotton-Phenolic	Carbon-Phenolic
Elastic modulus (GPa)	235	215	205	205	320	8	62.3
Poisson ratio	0.28	0.30	0.29	0.29	0.26	0.05	0.07
Thermal expansion coefficient (x10^6 °C^{-1})	11.2	9.0	11.2	11.5	2.9	18	0.7
Thermal conductivity (W/m-K)	20.7	21.3	37.0	46.6	29.3	0.375	3.25
Thermal conductivity (W/m-K) (radial)							0.85
Specific heat (J/kg-K)	429	400	425	425	1100	1465	989
Density (kg/m^3)	8172	8190	7971	7810	3160	1400	1400

Prediction of bearing heat generation is highly dependent on accurate representation of friction. For lightly loaded, high-sliding conditions of cage-land and cage-pocket contacts, the friction coefficient has been estimated as 0.225 based on AFRL/PRTM experiments using the rolling contact friction tester described by Forster and Givan (1999). Figure 6 represents some of the experimental friction data from Part I of this series of reports available for the cage-land sliding speed of interest. Traction data for the ball-race contacts has been based on experimental data (Figure 7) provided for Pennzane® in a fully flooded condition.

Typical ADORE traction curves used for the ball-pocket, cage-race, and ball-race contacts are shown in Figure 8. This data displays the general behavior of traction with relative slip. Traction initially increases with increasing slip, but as shear heating in the lubricant becomes significant at high slip velocities, it peaks and begins to drop. The relationship is as follows:

$$k = (A + Bu)\exp(-Cu) + D$$

where κ is the traction, *u* is relative slip velocity, and *A*, *B*, *C*, and *D* are empirical coefficients describing the lubricant. For cage contacts, the maximum coefficient of 0.225 at approximately 5% slip drops to 0.2 at infinite slip. For the ball/race contacts, a peak value of 0.125 at 5% slip and 0.12 at infinite slip has been used to account for the starved conditions. Values are based on mist phase lubrication of MIL-L-7808, a high-quality synthetic aircraft lubricant indicating traction values ~ 4X flooded conditions.

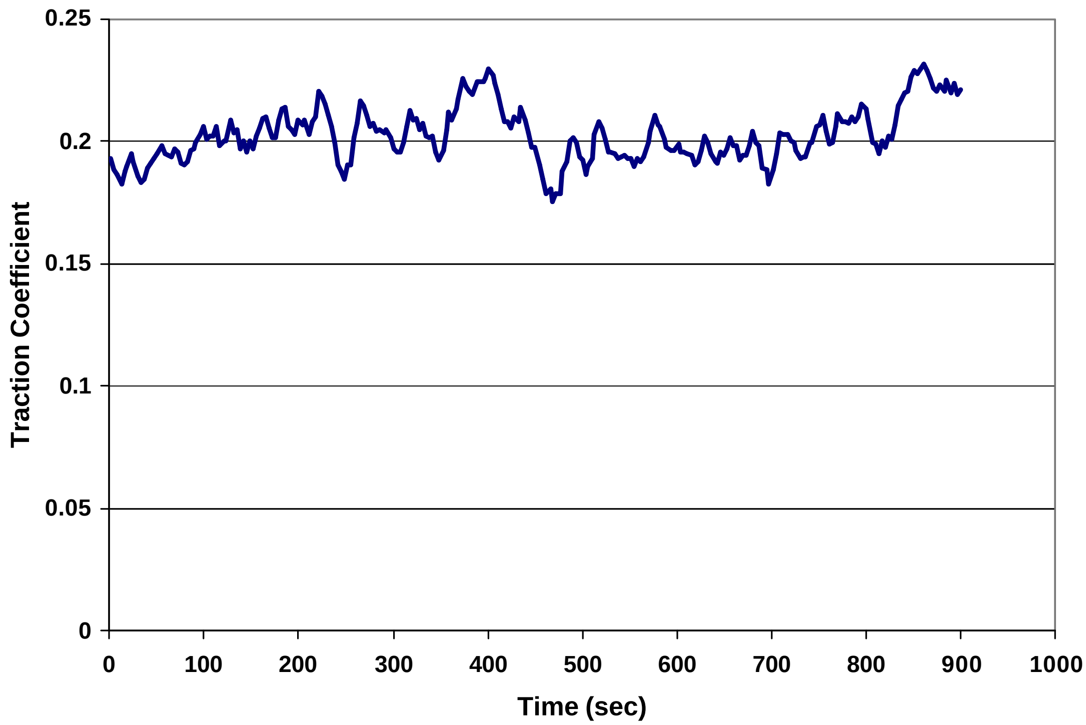

Figure 6. Friction Data for Cotton-Phenolic Cages
(Light Lubrication, 15 m/s Rolling Speed)

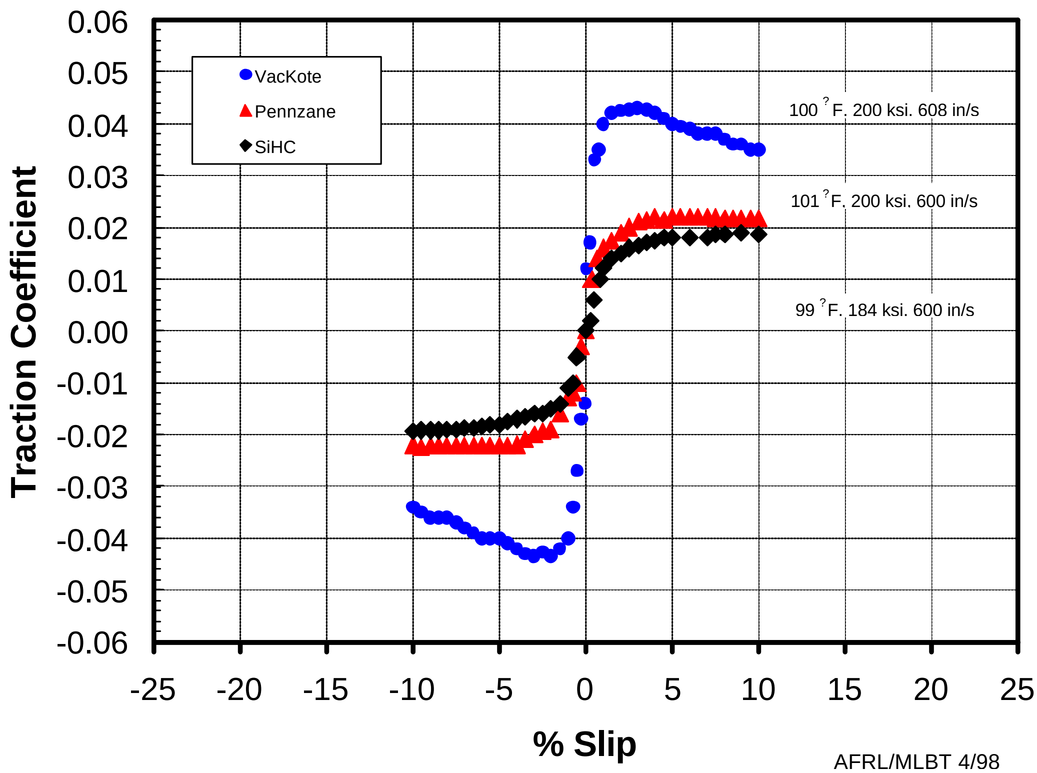

Figure 7. Pennzane® Lubricant Traction Data (Fully Flooded Conditions)

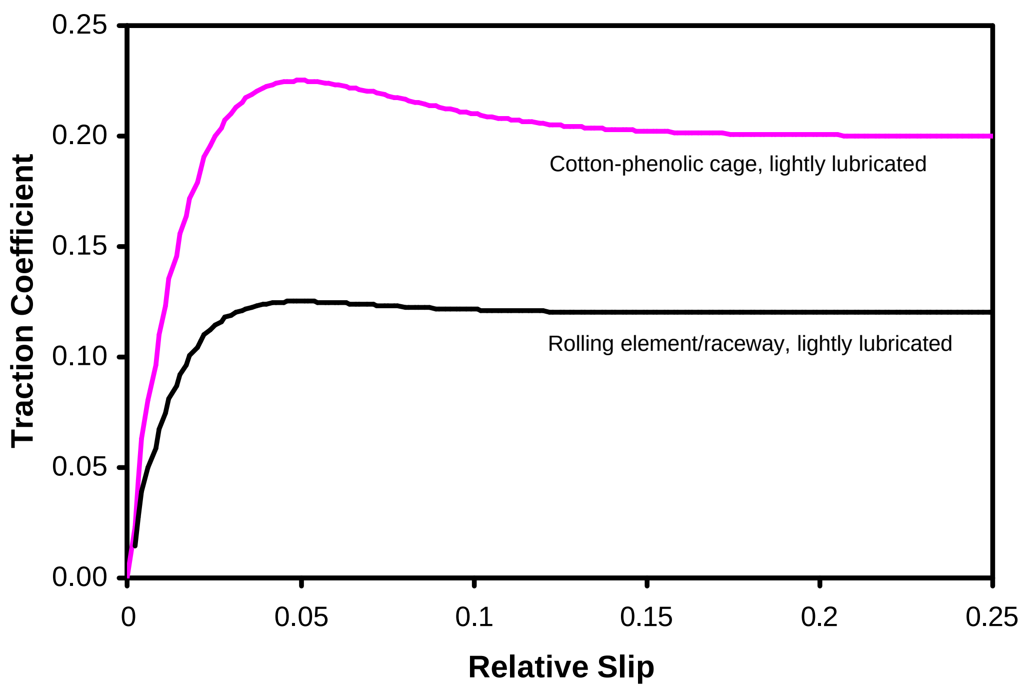

Figure 8. ADORE Traction Curves for Cotton-Phenolic Cage Contacts; ADORE Rolling Element-Raceway Traction Curve Accounting for Lubricant Starvation Effects

Following review of the model results, it has been observed that the value of the relative slip where the maximum traction occurs can significantly affect the predicted heat generation. Use of the modified form of the traction curves may more realistically predict the heat generation at both the lower and higher tested bearing speeds. This finding is supported by the work of Wedeven (1974) investigating increased traction coefficients for the same slide-to-roll ratio in highly starved contacts. Figure 9 depicts changes in traction curve shapes at low relative slip. This is further discussed in the experimental results comparison section.

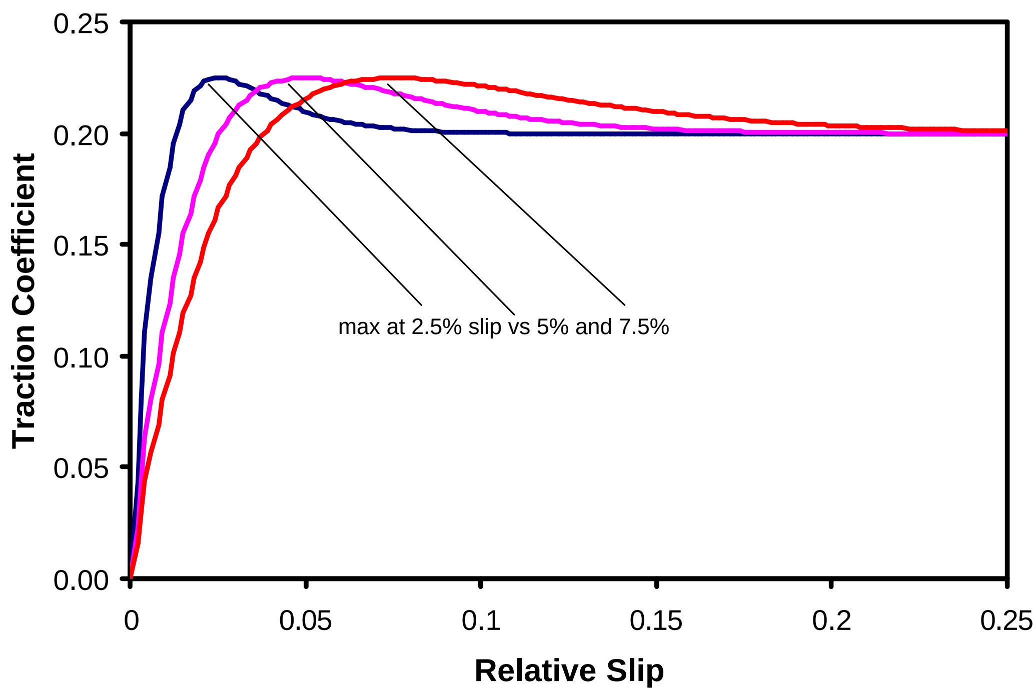

Figure 9. Moving Location of Maximum Traction in Terms of Relative Slip to Model Starvation

A 3-D ANSYS finite element thermal model of the bearing has been developed for a symmetric section of the duplex pair bearing geometry containing a single ball for each bearing along with portions of the shaft and bearing housing as shown in Figure 10. The model, which uses constant properties, accounts for heat conduction within the solid regions of the bearing and convection heat transfer from the bearing structure. There has been no attempt to model conduction within the lubricant due to the near-starvation conditions. The model provides detailed temperature data allowing evaluation of the effect of bearing material properties. ADORE total time-averaged heat generation values for bearing contact locations have been averaged over the number of rolling elements and used as boundary conditions in ANSYS for the symmetric thermal model. Heat loads have been applied to the model as shown in Figures 11 - 13. Heat generated at the cage/outer race contact area has been applied to the model using surface effect elements that allow for volumetric heat generation along the contact area. Heat flux boundary conditions have been applied to the ball track area along both the inner and outer raceway. A nodal heat flow boundary condition has been used in the ball pocket. Line contact elements, with areas of contact determined from ADORE, have been used to transfer heat by conduction from one raceway to the other through the rolling element. The modeled ball track and cage/race contact areas have been maintained constant in the analyses so that the same finite element model could be used for all conditions. This has eliminated the effect of mesh density on the thermal result variations. Heat loads have been adjusted to account for the difference in the model areas and the calculated contact areas from ADORE. The solid model has been meshed using approximately 30,000 3-D 20 node thermal elements with a single degree of freedom at each node. These elements are well suited to model the curved boundaries existing in the bearing geometry. Heat generated at the ball/raceway locations has been assumed to initially go into the raceway and then either transferred by conduction into the race material, conducted away from the race through the rolling element, or removed by convection to the ambient rig environment.

Heat transfer coefficients of 30 - 60 W/m^2-K for forced air convection at the test rig conditions (10,000 rpm) have been used. These values have been used for the exposed areas of the shaft, housing and the bearing. Variations have been based on the general arrangement of the experimental rig. The shaft temperature has been fixed at 20°C at a location 0.048 m upstream of the bearing due to the relatively cold conditions of the air turbine driving the shaft. Future efforts to measure the shaft and inner race temperatures will verify this assumption. A Newton-Raphson iterative solution procedure has been used to obtain convergence, and a Jacobi Conjugate Gradient solver has been used for the matrix solution.

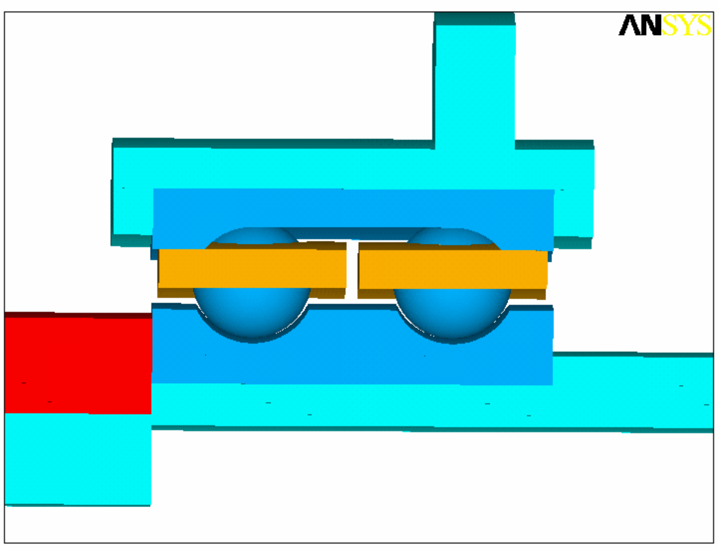

Figure 10. Schematic of Solid Model of Bearing Symmetric Section with Single-Land Outer Race Guided Cage

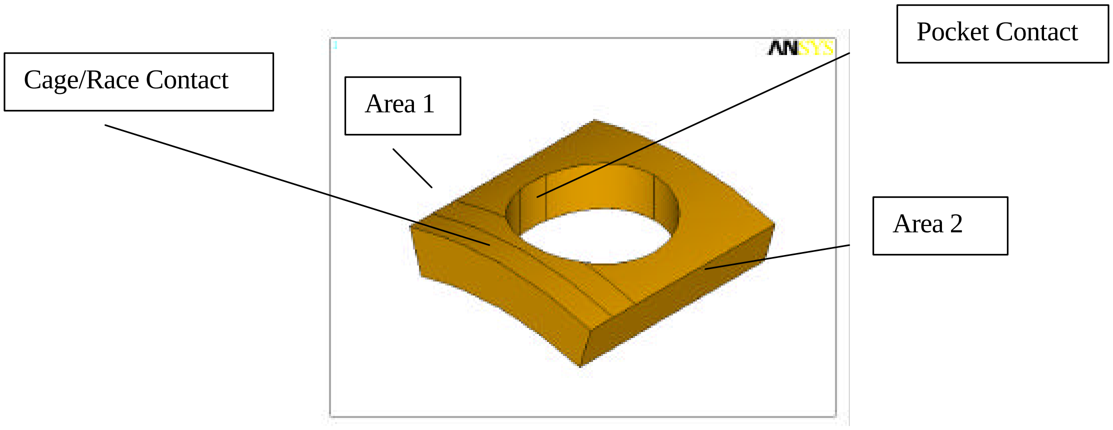

Figure 11. Section of Bearing Cage Showing Ball/Pocket Contact Area and Cyclic Temperature Conditions (Area 1 Temp = Area 2 Temp)

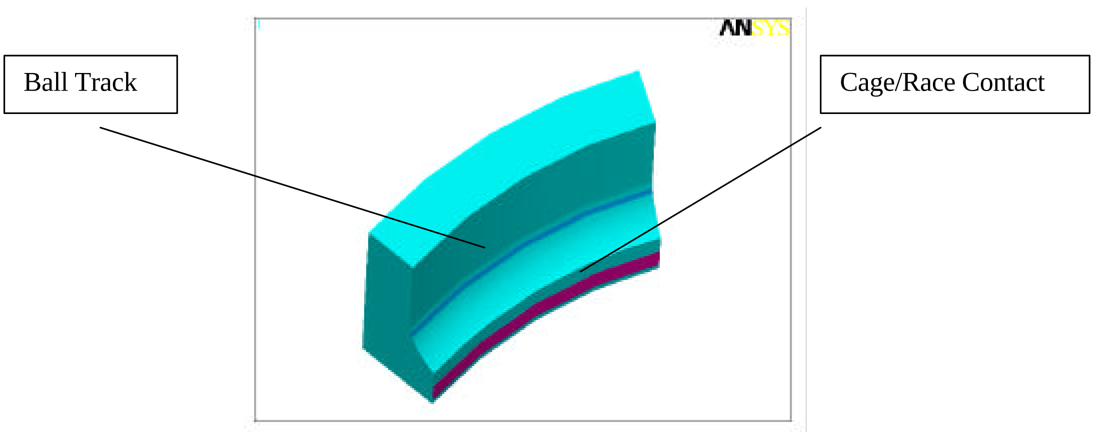

Figure 12. Bearing Outer Race Ball Track and Cage/Outer Race Contact Area

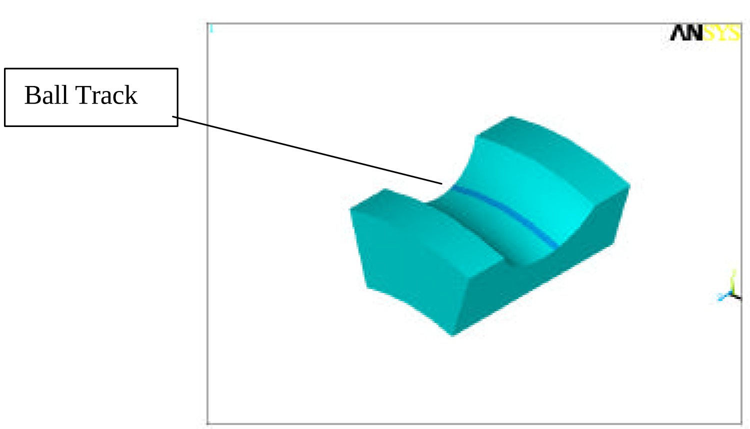

Figure 13. Bearing Inner Race Ball Track

4. Numerical Results/Comparison with Experimental Data

The bearings described previously have been analyzed using ADORE for two different ball materials, with an applied preload of 270N (65 lb), and operating at a shaft speed of 10,000 rpm. The runs have used traction coefficient data presented earlier of 0.225 maximum for cage contacts and 0.125 maximum for the ball/race contacts. ADORE calculates a maximum inner race Hertzian contact stress of 1.1 GPa for metal balls and 1.25 GPa for ceramic balls and a maximum outer race Hertzian contact stress of 0.85 GPa for the metal and 0.94 GPa for the ceramic. Key ADORE outputs for thermal analyses are dynamic contact locations, areas, and heat generation. Figure 14 presents transient heat generation during various contacts for the lightly-lubricated 206 size bearing with cotton-phenolic cage and Si_3N_4 balls operating at 0.3 MDN (MDN = bearing bore size in mm x shaft speed in rpm divided by 10^6). Contact between the rolling element and the inner and outer raceway is relatively constant while contact within the cage pocket and between the cage and outer race is much more intermittent. Data has been time-averaged, as shown in Figure 15. It is this information for the various bearing design conditions that has been used for evaluation of bearing performance.

Table 3 provides predicted time-averaged steady-state heat loads calculated for rolling element materials of interest. Time-averaged contact areas have been used for comparison with the finite element model areas. Data for the ceramic ball bearing is also provided at speeds up to 40,000 rpm to show the large increase in heat generation with speed. Based on these results, the duplex pair with ceramic rolling elements in the AFRL/PRTM experiments generates approximately 9.3 W at 10K rpm and 45.2 W at 20K rpm. These values can be compared with experimental measurements using the assumption that the bearing heat generation, generally termed power loss, can be determined using the measured friction torque, M, and shaft speed, ω. ($Q = M * \omega$)

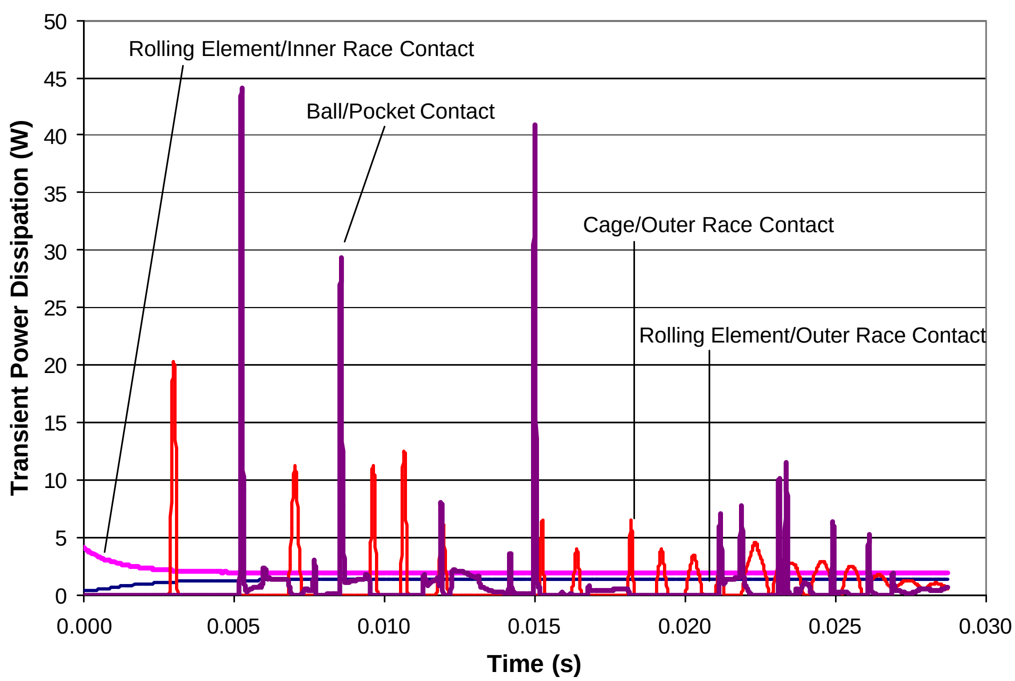

Figure 14. ADORE Predicted Instantaneous Heat Generation Data (206 Size Bearing, Cotton-Phenolic Cage, Si_3N_4 Balls, Lightly Lubricated, 0.3 MDN)

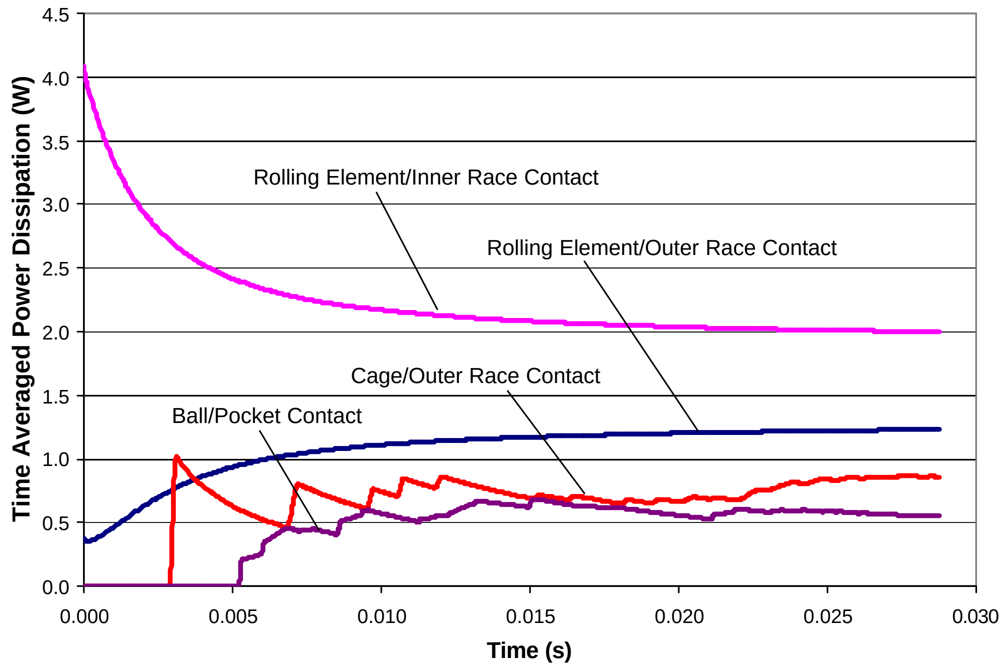

Figure 15. ADORE Predicted Time-Averaged Heat Generation Data (206 Size Bearing, Cotton-Phenolic Cage, Si_3N_4 Balls, Lightly Lubricated, 0.3 MDN)

Table 3. Time-Averaged Heat Generation (W) based on ADORE Single Bearing Analysis

Ball Material	Speed (rpm)	Outer Race	Inner Race	Pocket	Cage Land	Total (Single)	Total (Duplex)
52100	10000	1.69	3.16	0.43	0.95	6.27	12.54
Si_3N_4	10000	1.23	2.00	0.70	0.87	4.64	9.28
Si_3N_4	20000	3.38	7.60	3.21	8.20	22.59	45.18
Si_3N_4	40000	13.37	44.24	28.04	83.33	169.65	338.30

Figures 16 - 17 provide excellent experimental data for comparison with the numerical bearing analysis and finite element thermal model predictions. The experimental data represent test results for three different bearings. Two of the bearings used Si_3N_4 balls and the other used 52100 steel balls. The torque measurements in Figure 16 represent test results for two runs with each bearing. The outer race temperature data in Figure 17 is from the same set of tests with only one of the repeat test results being shown. It appears, based on the data in these two figures, that the bearing with the 52100 rolling elements generates the most heat, as is predicted by ADORE. Using $Q = M * \omega$, with an average value of 0.015 N-m for the 52100 bearing and 0.010 N-m for the ceramic rolling element bearing, leads to measured heat generation values of 12 W and 9 W, respectively. These results demonstrate good agreement of the ADORE predictions from Table 3 with the experimentally determined values.

The experimental steady-state temperatures from Figure 17 have been used for comparison/ validation of the ANSYS model using the ADORE predicted heat loads. Figures 18 - 19 provide steady-state bearing temperatures obtained using the finite element model. For the evaluated conditions, the predicted outer race temperatures are consistent with the experimental results, 46.5°C for the Si_3N_4 balls and 55°C for the 52100 balls. The potential to develop large temperature gradients within the cage material, as shown in these figures, is one of the reasons that novel cage materials are being investigated for use at increased bearing operating speeds. The predicted bearing temperature with a carbon-phenolic cage for these conditions, shown in Figure 20, demonstrates a nearly isothermal cage with a slightly reduced outer race operating temperature. No carbon-phenolic cages were tested using clamped loading. As such, there is no experimental comparison of the different cage operating performance under these conditions.

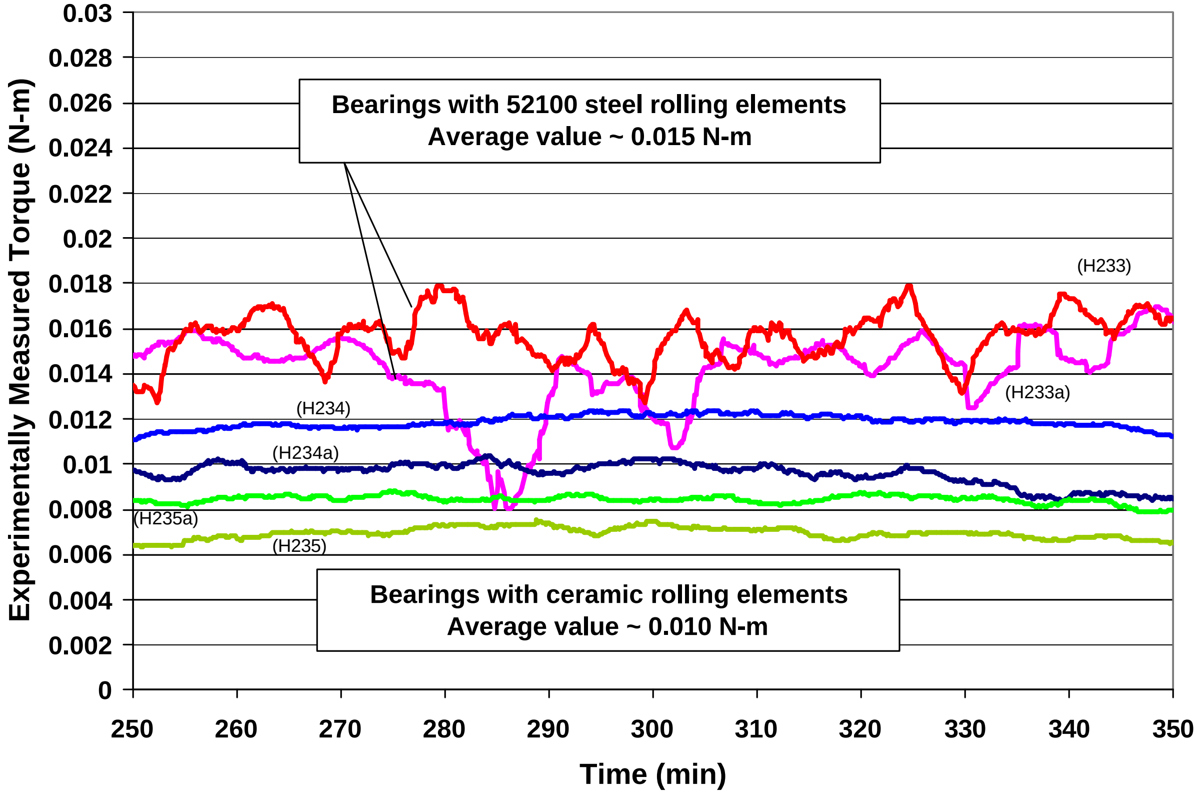

Figure 16. Experimental Torque Measurements (206 Size Duplex Pair, Cotton-Phenolic Cage, 10K RPM, Refer to Part II for Test Numbering)

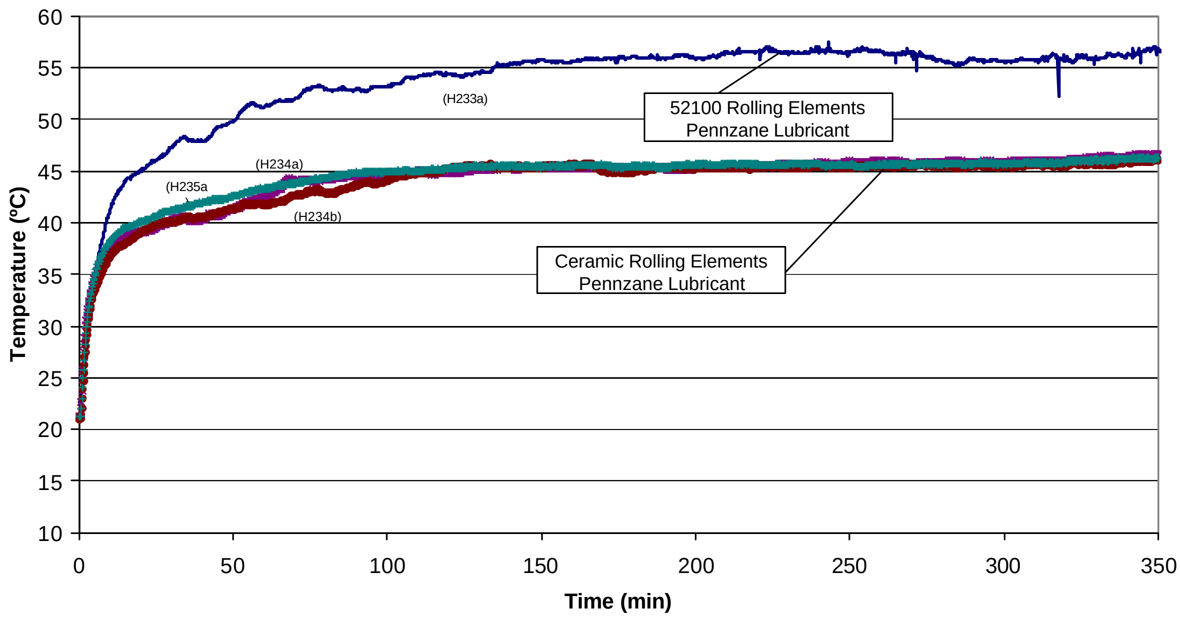

Figure 17. Experimental Outer Race Temperature Measurements (206 Size Duplex Pair, Cotton-Phenolic Cage, 10K RPM, Refer to Part II for Test Numbering)

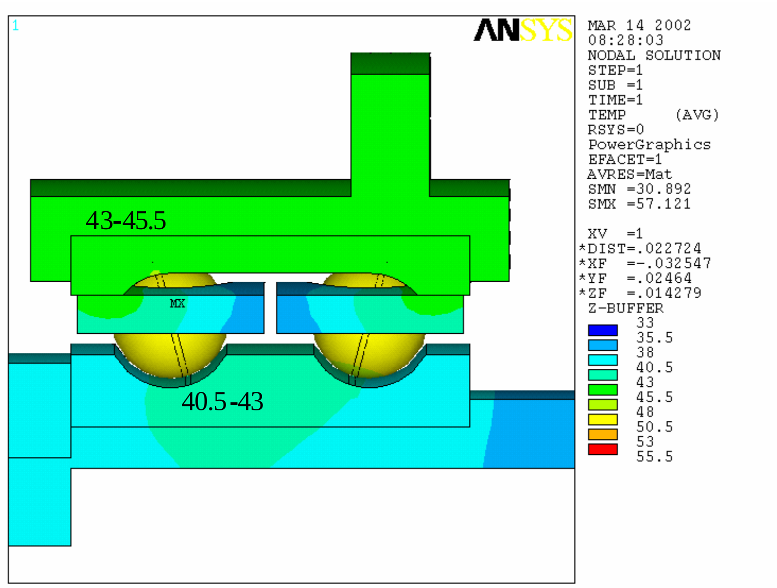

Figure 18. ANSYS Steady-State Temperature (206 Size, Si_3N_4 Balls, Cotton-Phenolic Cage, 10K RPM)

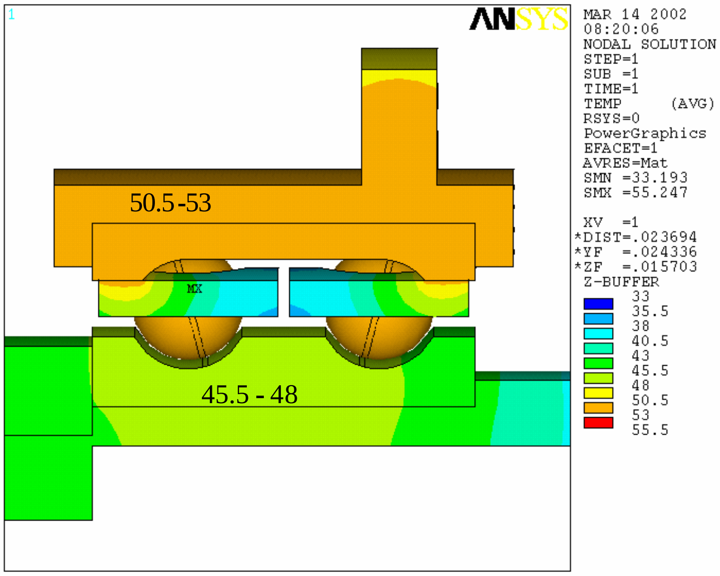

Figure 19. Steady-State Temperature (206 Size, Metal Balls, Cotton-Phenolic Cage, 10K RPM)

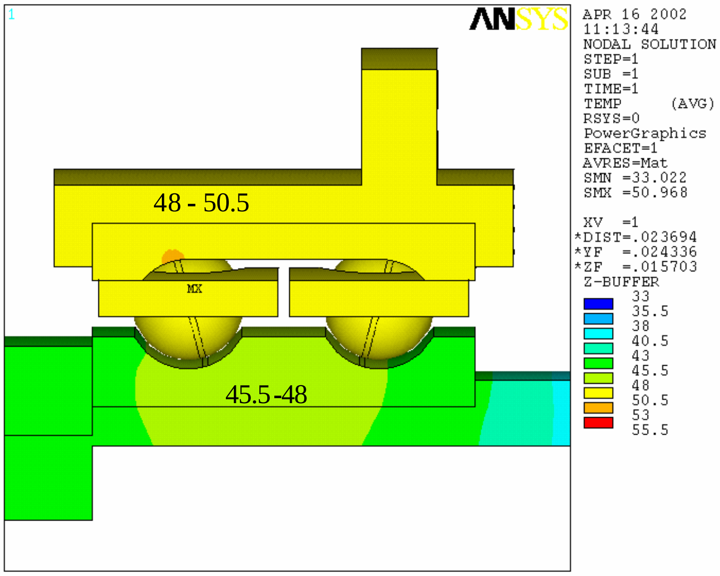

Figure 20. ANSYS Steady-State Temperature (206 Size, Metal Balls, Carbon-Phenolic Cage, 10K RPM)

Transient thermal analyses have also been accomplished for comparison with the tested bearing outer race temperature for the startup time period of 25 minutes. This data, shown in Figure 21, displays excellent agreement with the measured temperatures and provides a preliminary validation of the methods used to model the bearing thermal response. The inset figures show the hot spot within the cage pocket and the heated ball track region. Additional experimental results for the inner race temperature measurements will strengthen the validation.

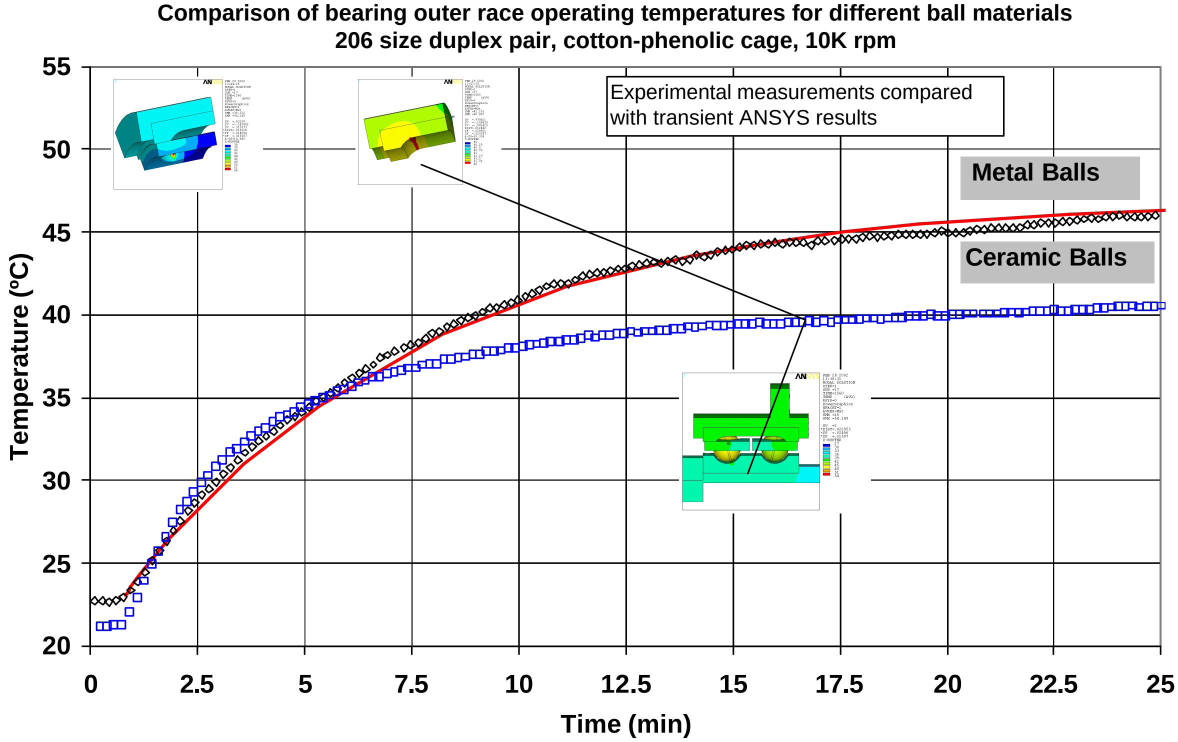

Figure 21. Comparison of Transient ANSYS Results with Measured Outer Race Temperatures

Although agreement between ADORE-ANSYS predictions and experiment appears excellent for the cases presented, there has been significant concern regarding the effect of thermal expansion on the bearing clamped loading. A process is available using ADORE to determine the effect of changes in bearing operating temperature on bearing load due to thermal expansion. The method first accomplishes the bearing analysis at room temperature with applied loads at a desired speed. ADORE generates a quasi-dynamic initial load, L_i, derived from the initial preload, centrifugal loads, and any other loads on the system. ADORE also calculates a corresponding initial displacement, $X_i = L_i / K$, where K is axial stiffness. ADORE analysis can then be accomplished using the prescribed displacement, X_i, and temperatures of interest for the housing, shaft, cage, rolling element, inner race, and outer race. Using the specified temperatures, along with the material coefficients of thermal expansion, ADORE will calculate an additional displacement due to temperature, X_T. ADORE will also calculate a new bearing load L_T using the bearing axial stiffness and the combined displacement, $L_T = K (X_i + X_T)$. For a preloaded bearing, the load L_T is the combination of applied loads to the bearing and thermal loads due to expansion inside of the bearing. Using an ANSYS temperature profile corresponding to an experimentally measured outer race temperature, (outer race = 113ºF, inner race = 108ºF, housing = 109ºF, shaft = 104ºF, cage = 104ºF, ball = 122ºF), as an input to ADORE, the calculated load is observed to be reduced from 65 lbf to 31.3 lbf. This calculation is the basis for the 60 lbf and 30 lbf spring loadings that have been selected as reported in Part II. At the present time, it is noted that the modeled ADORE loads for the initial test conditions are likely larger than actually existed. Therefore, either maximum traction coefficient, shape of the traction versus relative slip curve, or the bearing environment conditions require modification to account for the experimentally observed bearing heat generation.

Based on experimental data, it is not believed that the maximum traction coefficients are much larger than modeled. The effect of changing maximum traction coefficient location and the cage imbalance on ADORE predicted heat generation for 10K rpm with the reduced loading is shown in Table 4. All cases used a maximum traction coefficient of 0.225 and are for the Si_3N_4 rolling elements. Comparison with Table 3 data for Si_3N_4 balls at 5% slip for 10K rpm demonstrates approximately the same total heat generation. However, the ball-race contacts are now a much larger fraction of the heat generation than the cage.

Table 4. Effect of Traction Coefficient Maximum Location versus Relative Slip on Adore Predicted Heat Generation

Case	Imbalance (g-cm)	Relative Slip Location	Total Heat (W)	RE/OR (W)	RE/IR (W)	Pocket (W)	Cage (W)
1	.20	2%	9.54	3.36	5.37	0.25	0.51
2	.20	2.5%	8.05	2.96	4.67	0.26	0.15
3	.20	5%	5.10	1.80	2.82	0.23	0.20
4	.40	2.5%	8.46	2.96	4.67	0.34	0.55

The best experimental comparison between the bearings operating with cotton-phenolic cages versus carbon-phenolic cages is shown in Figure 22. There are several factors which require noting for this series of tests. These tests have all used CRU-20 races and have been conducted using a spring-loading mechanism to load the duplex pair bearings. The spring-loaded design has been used to maintain a more constant load on the bearings than in the clamped-loaded design. The measured steady-state temperatures with spring loading are higher than with clamped loading. This implies that the loading is being better maintained. It has also been observed that although the bearing temperatures did increase with increased operating speed, they did not increase as dramatically as predicted. Modifications to the traction curves that change the location of the maximum traction coefficient as a function of relative slip have been shown to affect this condition. This is due to higher heat generation at the ball-race contact for low relative slip reduces the significance of the cage-land contact factor. The cage-land contacts can then be modeled with possibly a lower maximum traction coefficient. However, there is still a speed at which cage-land heat generation becomes dominant (~ 25K rpm in these tests) and leads to dramatically increased outer race temperatures. It is also noted that the carbon-phenolic cage tests actually ran slightly hotter than did the cotton-phenolic bearing cage tests. This has been attributed to debris as described in Part II of this series of reports.

ANSYS steady-state predictions for this series of tests are provided in Figures 23 - 26. Figure 23 shows predicted outer race temperature of 63°C for the carbon-phenolic cage test at 10K rpm and 65 lbf load when using the location of maximum slip changed to 2.5%. This is approximately the value measured experimentally. However, the ANSYS model significantly over predicts (Figure 24) the cotton-phenolic cage results. Modifications to the maximum traction coefficient for the cotton-phenolic cage to 0.165 led to the results in Figure 25. Increasing the speed to 12.5K rpm led to the results for the cotton-phenolic test shown in Figure 26. Here, the operating temperature is increased slightly above that for the 10K rpm case.

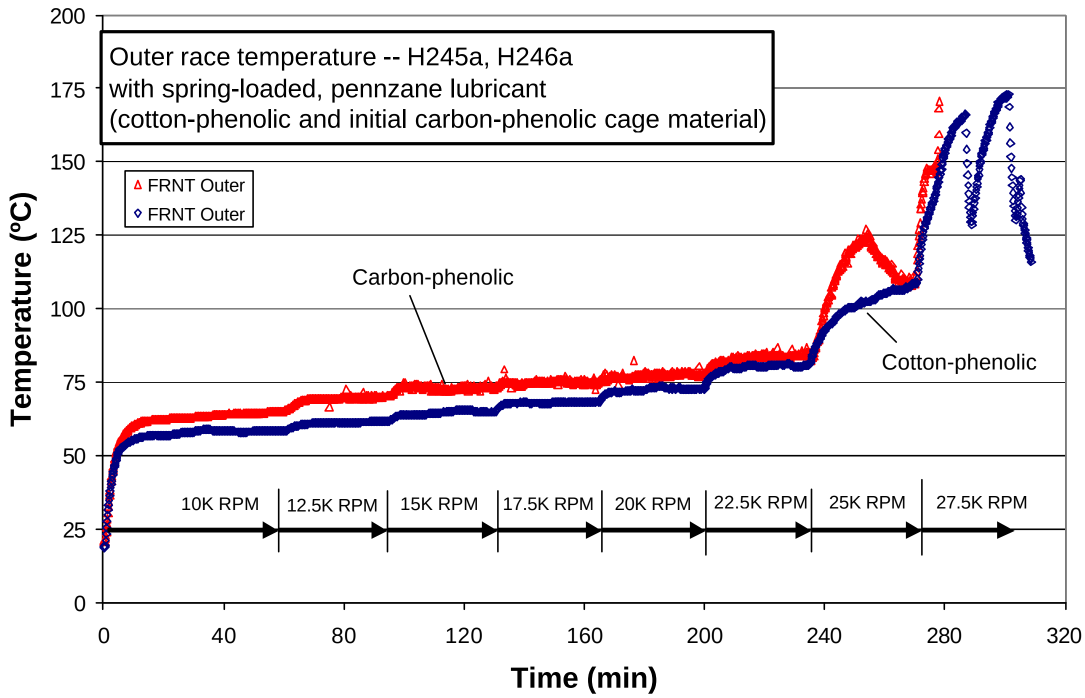

Figure 22. Comparison of Experimentally Measured Outer Race Temperatures for Carbon-Phenolic and Cotton-Phenolic Cages (206 Size Duplex Pair, CRU-20 Races, Si_3N_4 Balls)

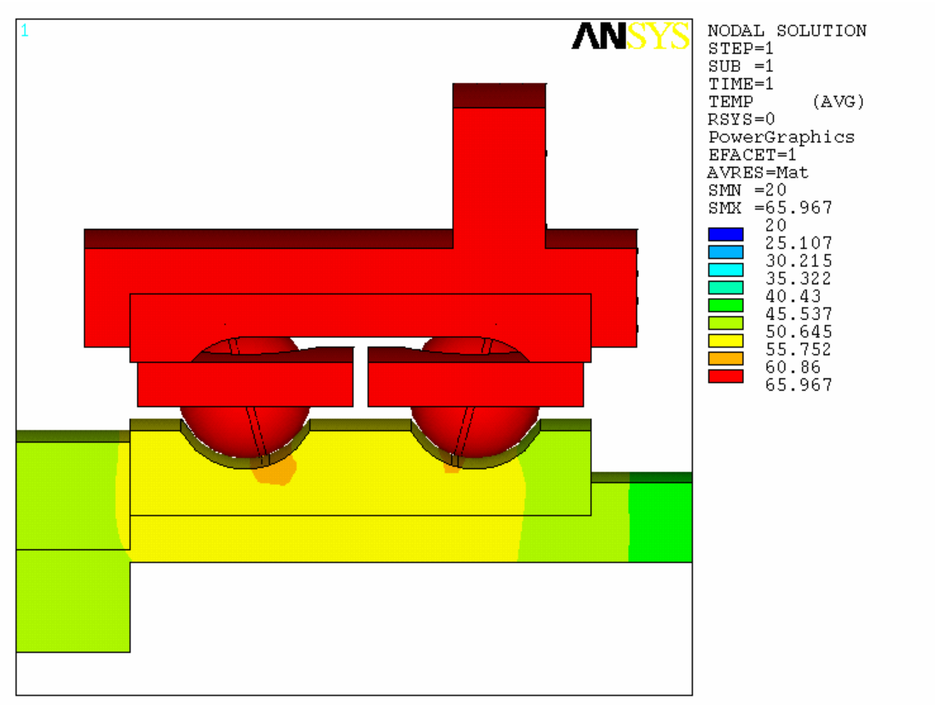

Figure 23. ANSYS Steady-State Temperature, °C, (206 Size Duplex Pair, Si_3N_4 Balls, Carbon-Phenolic Cage, 10K RPM, 0.225 Max Traction Coefficient at 2.5% Slip)

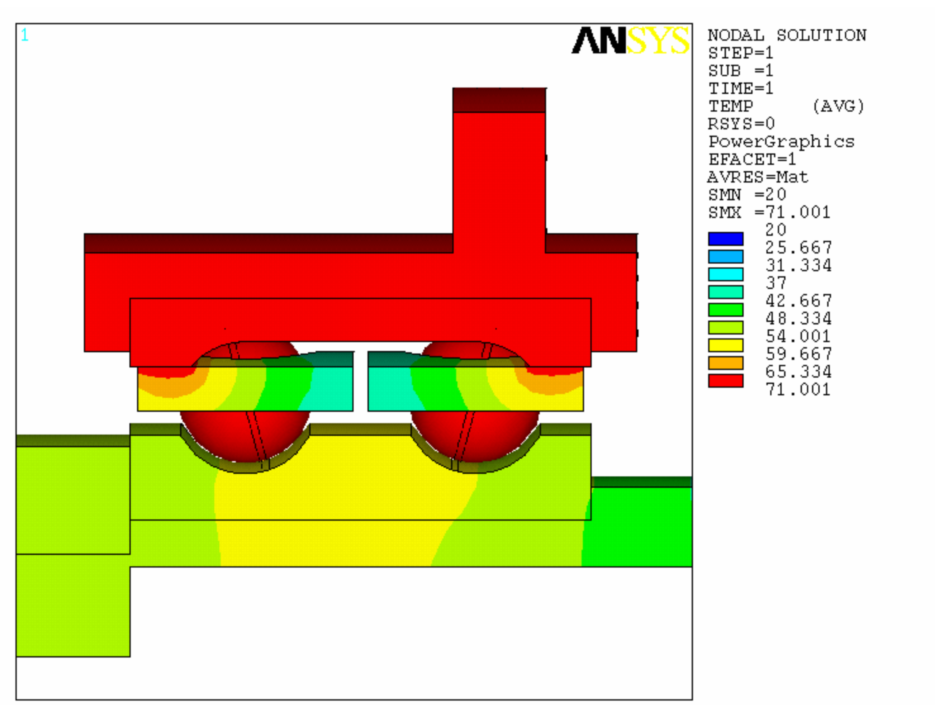

Figure 24. ANSYS Steady-State Temperature, °C, (206 Size Duplex Pair, Si_3N_4 Balls, Cotton-Phenolic Cage, 10K RPM, 0.225 Maximum Traction Coefficient at 2.5% Slip)

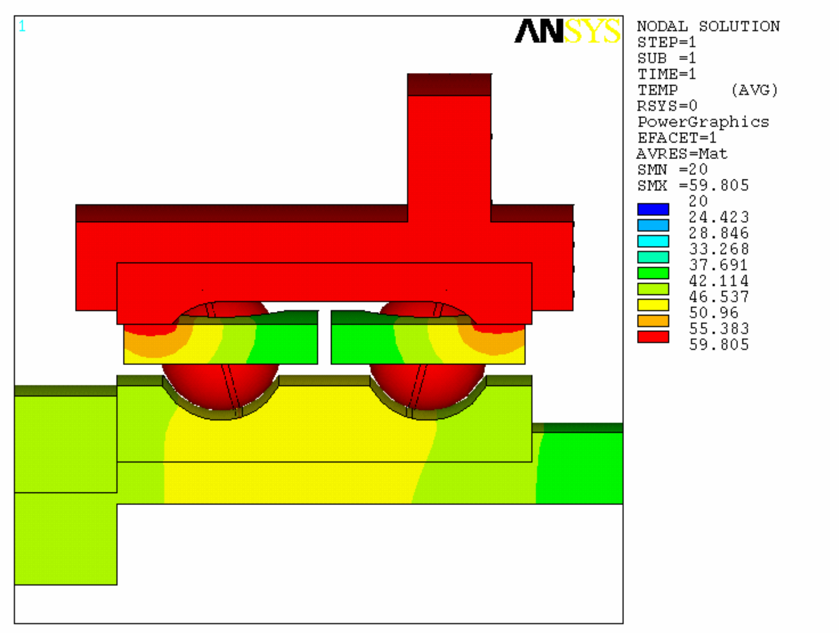

Figure 25. ANSYS Steady-State Temperature, °C, (206 Size Duplex Pair, Si_3N_4 Balls, Cotton-Phenolic Cage, 10K RPM, 0.165 Maximum Traction Coefficient at 2.5% Slip)

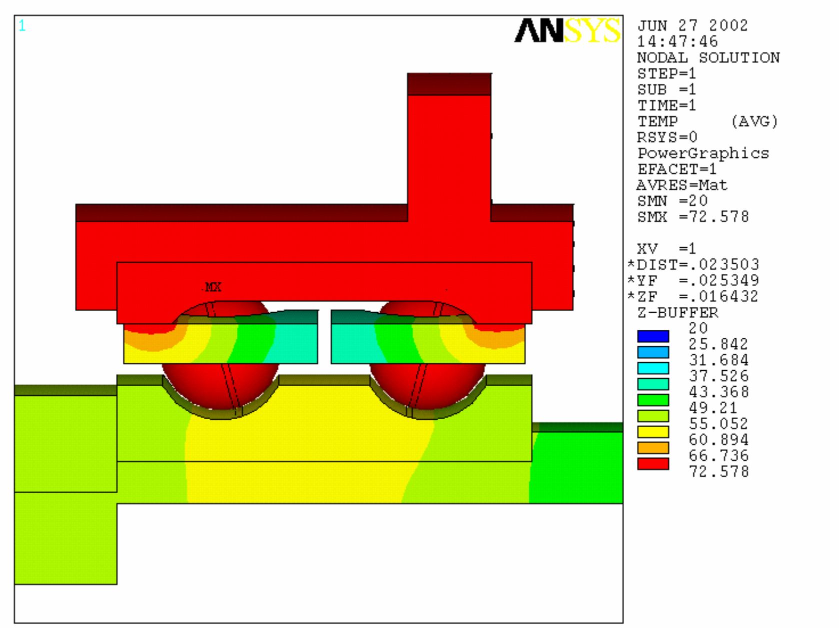

Figure 26. ANSYS Steady-State Temperature, °C, (206 Size Duplex Pair, Si_3N_4 Balls, Cotton-Phenolic Cage, 12.5K RPM, 0.165 Maximum Traction Coefficient at 2.5% Slip)

5. Summary

It is apparent that increases in bearing speed can significantly increase heat generation and that this heat generation can lead to temperatures too extreme for reliable operation. A process has been developed to numerically investigate operating temperatures of a rolling element bearing which combines capabilities of ADORE bearing analysis software with finite element thermal analysis. Experimental efforts within AFRL/PRTM, sponsored by NRO to investigate benefits of novel cage materials in satellite bearing applications, have provided an opportunity to accomplish bearing thermal performance validations. These experiments have used minimal lubrication, which reduces heat transfer by convection, the primary mode for heat rejection in conventional bearings. Results have been presented that describe experimental and predicted transient and steady-state temperature responses for a 30-mm ball bearing duplex pair operating with only a light coating of Pennzane® lubricant for two different rolling element materials (Si_3N_4, 52100 steel) in a bearing with 52100 steel races operating at a speed of 10,000 rpm using a conventional cotton-phenolic cage. Although agreement of these results appeared excellent, further analysis led to model modifications to account for the potential of thermal expansion to cause an increase or decrease of bearing load which can be significant for the relatively light load on a satellite bearing. Modifications to the experimental rig were also made to account for the thermal expansion effects by incorporating a spring-loading mechanism rather than relying on the clamped load. Investigations of modifications to the shape of the traction curves used in the bearing analysis model and the affect on predicted heat generation helped answer questions concerning the expected large contribution of cage heating with increases in speed that was not observed until a failure criterion was reached. Final model and experimental results have been presented to compare the performance of a carbon-phenolic cage relative to a cotton-phenolic cage operating in 30-mm bearing with CRU20 steel races. To accurately simulate the transient thermal response of the tested bearings, it is important that the heat loads be applied with the correct magnitude/rates and at the correct time. There are several factors which affect the predicted loads, including traction coefficients, cage imbalance, and any changes during heating/startup such as changes in lubricant viscosity. Realistic material property values for density, thermal conductivity, and specific heat, along with an accurate description of the bearing environment, including heat transfer coefficient variations with rotation speed, are also critical. Although the current series of bearing tests conducted by AFRL/PRTM did not demonstrate the numerically predicted benefits of the carbon-phenolic cage relative to the cotton-phenolic cage, they did lead to a critical review of the methods used in bearing and thermal analyses. The numerical results also led to the conclusion that debris generated during testing with the carbon-phenolic cage increased the apparent traction coefficient, and control of this debris is a topic that will be addressed in future efforts. Results indicate the potential of the analysis process to improve bearing design through the evaluation of thermal conditions and determination of cooling requirements for future bearing system designs. Future efforts to obtain experimental results for higher shaft speeds, including inner race temperature measurements, will strengthen the validation. Additional research is being conducted to improve the level of model fidelity in the area of specifying heat transfer coefficients as a function of operating speed for the various bearing designs and environments. Research to develop traction data for the Pennzane® lubricant under starved conditions should also be accomplished. Additional validation efforts will be conducted both for higher bearing operating speeds and for larger bearings.

6. References

Brown, J.R. (2001), "Numerical Analysis of Cage Imbalance Effects on Bearing Heat Generation," *Proceedings of the 36th Intersociety Energy Conversion Engineering Conference*, Vol. 2, pp. 1201-1206.

Brown, J.R. and Forster, N.H. (2000), "Operating Temperatures in Mist-Lubricated Rolling Element Bearings for Gas Turbines," AIAA 2000-3027.

Brown, J.R. and Forster, N.H. (2002), "Numerical Investigation of the Effect of Carbon-Carbon Composite Cages on High-Speed Bearing Operating Temperatures," *Tribology Transactions*, Vol. 45, No.3, pp. 411-417.

Brown, J.R., Forster, N.H., Givan, G.D., and Gerardi, D.T. (2002), "Experimental Validation of Numerically Predicted Heat Generation in Lightly-Lubricated Rolling Element Bearings," *ASTM International Symposium on Rolling Element Bearing Program and Papers*, pp. 29-38.

Forster, N.H. and Givan, G.D. (1999), "Computer Modeling of Heat Generation in Vapor Lubricated Bearings for Gas Turbines," NATO Research and Technology Organization, RTO-MP-8, AC/323(AVT)TP/9, pp. 1-9.

Forster, N.H., Rosado, L., Shih, W.T, and Brown, J.R. (2001), "The Development of Carbon-Carbon Composite Cages for Rolling Element Bearings, *Tribology Transactions*, Vol. 45, No.1, pp. 127-131.

Gupta, P.K. (1984), Advanced Dynamics of Rolling Elements, Springer-Verlag, New York.

Sanders, J.H., Cutler, J.N., Miller, J.A., and Zabinski, J.S. (2000), "In vacuuo tribological investigations of metal, ceramic, and hybrid interfaces for high-speed bearing applications," *Tribology International*, Vol. 32, pp. 649-659.

Timoshenko, S. and Goodier, J.N. (1951), Theory of Elasticity, McGraw-Hill, New York.

Wedeven, L.D. (1974), "Traction and Film Thickness Measurements Under Starved Elastohydrodynamic Conditions," *Journal of Lubrication Technology*, 74-Lub-30, pp. 1-9.