Influence of Oil Lubrication on Spacecraft Bearing Thermal Conductance

Yoshimi R. Takeuchi*, Matthew A. Eby*, Benjamin A. Blake*, Steven M. Demsky* and James T. Dickey*

Abstract

Increasing demands on bearing performance and a lack of thermal conductance data for bearings in space applications motivated The Aerospace Corporation to study heat transfer across angular-contact ball bearings for space systems. Tests were conducted under controlled conditions including rotational speed, temperature, axial load, and vacuum environment. Bearings with Nye Pennzane SHF2001 synthetic oil were compared with dry (non-lubricated) bearings. These comparisons show that dry and oil lubricated bearings vary in thermal conductance by up to an order of magnitude. Experimental measurements also indicated that sensitivity to other variables, such as axial load and temperature, depends on whether the bearing is dry or oil lubricated, and whether it is in a static or dynamic (rotating) state. Mechanisms of heat transfer are discussed for each of these states.

Introduction

In contrast to typical terrestrial applications, the absence of convection shifts the focus for thermal analysis of rotational space hardware. In vacuum environments, conductance through the bearings often provides the primary heat transfer path between the shaft and the housing. As such, temperature predictions for rotating components, such as satellite instruments, or the bearing itself require knowledge of the bearing thermal conductance. However, published literature provides little help on the subject, and thus bearing thermal conductance is usually the significant unknown in the development of a thermal model of a rotational system in space. When available, engineers often use heritage information for comparable systems with similar bearings. Significant uncertainty arises with the advent of design changes, different bearing geometry, different lubricant type or quantity, or dissimilar operational conditions. An absence of thermal conductance data leads to challenges in using thermal models for guidance in the design process.

Existing literature yields limited thermal conductance information for static and low-speed bearings [1-9] and none for high-speeds. Yovanovich [1,2] developed a mathematical model for the thermal conductance of a non-lubricated, static (non-rotating) bearing. Experimental work followed, including studies on spacecraft bearings performed by Stevens and Todd [3], from the European Space Tribology Laboratories (ESTL). They measured thermal conductance across a bearing, up to a maximum speed of 2,500 RPM, using an experimental setup designed by Dell et al [4-5]. ESTL continued to study this subject over the years, focusing on static or low-speed and large thin cross-section bearings [6-8].

Demand on bearing performance has grown and thermal concerns have increased as systems have reached higher speeds. Current momentum wheels and control moment gyroscopes typically operate at 6,000-9,000 RPM [10]. Future wheels, including energy storage flywheels, are envisioned to run at even higher speeds, above 15,000 RPM. In addition to the need for high-speed data, low-speed applications could benefit from a greater range of experimental data, and additional sources of counter verification from other researchers.

To address these concerns, an experiment was designed to assess thermal conductance of bearings in vacuum, at speeds ranging from 0 to 6,000 RPM. Controlled studies were conducted by varying variables such as axial load, thermal boundary conditions, and rotational speed, individually for parametric studies. The tests identified variables of importance for different bearing conditions, including dry, lubricated, static, and dynamic states. Finally, qualitative theories for the mechanisms of heat transfer for each of these conditions emerged.

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Nomenclature

G – conductance across the bearing
N – number of balls in the bearing
Ri – thermal resistance at the contact between the inner race and one ball
Ro – thermal resistance at the contact between the outer race and one ball
Rb – thermal resistance across inner to outer race of one ball
k, – thermal conductivity of the inner race material
k, – thermal conductivity of the ball material
k0 – thermal conductivity of the outer race material
a – major axis of a Hertzian contact area
a, – major axis of the Hertzian contact area between the inner race and one ball
a0 – major axis of the Hertzian contact area between the outer race and one ball
b – minor axis of a Hertzian contact area
b, – minor axis of the Hertzian contact area between the inner race and one ball
b0 – minor axis of the Hertzian contact area between the outer race and one ball

Experiment

Reference [11] provides a detailed description of the experimental setup and measurement techniques. All tests were conducted in vacuum environments of at least 1x10⁻⁶ Torr (approximately 1.3x10⁻⁵ Pa). As a summary, the experimental design matrix is outlined as follows:

1. Vary rotational speeds between 0 and 6,000 RPM. The maximum speed of 6,000 RPM is typical of a spacecraft control moment gyroscope (CMG) or momentum wheel.
2. Apply a constant pure axial load ranging from 40 to 129 N.
3. Accommodate bearings of different sizes, namely the 101 and 204-size ball bearings.
4. Test dry or oil lubricated bearings.
5. Vary the average bearing temperature.

Oil lubricated bearings were tested in either the virgin or fully run-in states. A fully run-in bearing was established by continually running the bearing at a constant speed until heat generation, torque, and thermal conductance remain unchanged (about a week of continual operation).

Test Bearings

A 52100 steel bearing and a hybrid bearing, consisting of silicon nitride balls and 52100 steel races, were tested. Table 1 provides the specifications of the two different angular contact ball bearing sizes used. Table 2 summarizes the relevant material properties.

Table 1. Ball Bearing Specifications

<table>
<thead>
<tr>
<th>Bearing Input</th>
<th>204-size</th>
<th>101-size</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Diameter</td>
<td>47</td>
<td>28</td>
<td>mm</td>
</tr>
<tr>
<td>Inner Diameter</td>
<td>20</td>
<td>12</td>
<td>mm</td>
</tr>
<tr>
<td>Ball Diameter</td>
<td>7.94</td>
<td>4.76</td>
<td>mm</td>
</tr>
<tr>
<td>Number of Balls</td>
<td>10</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Angle of Contact</td>
<td>15</td>
<td>15</td>
<td>degree</td>
</tr>
<tr>
<td>Cage Material</td>
<td>Phenolic</td>
<td>Phenolic</td>
<td></td>
</tr>
<tr>
<td>Cage Type</td>
<td>H-type, non-separable</td>
<td>H-type, non-separable</td>
<td></td>
</tr>
<tr>
<td>Cage Land</td>
<td>Outer</td>
<td>Outer</td>
<td></td>
</tr>
</tbody>
</table>
Table 2. Ball Bearing Material Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Silicon Nitride</th>
<th>52100 Steel</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic Modulus</td>
<td>310</td>
<td>210</td>
<td>GPa</td>
</tr>
<tr>
<td>Poisson's Ratio</td>
<td>0.27</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>30</td>
<td>26.6</td>
<td>W/m-K</td>
</tr>
</tbody>
</table>

Results

Following the experimental matrix, tests were performed to establish the influence of rotational speed, axial load, and temperature on bearing thermal conductance. While results for oil-lubricated bearings prove most relevant to a spacecraft application, the influence of axial load on the thermal conductance of a static dry bearing is already well known analytically [1-2]. Thus, the experimental results for a static dry bearing provide an excellent means of verifying the experimental technique. Further testing extends the results to static and dynamic (constant rotational speed) lubricated bearings, and reveals drastic shifts in the magnitude and trend of bearing thermal conductance relative to axial load and temperature.

Dry (Non-Lubricated) Static Bearings

The analytical relationship between thermal conductance and axial load in a non-lubricated, non-rotating bearing is well known. Yovanovich [1-2] developed the best-known analytical method, deriving the equation for bearing thermal resistance. A comparison with this model was used to verify the experimental approach, before proceeding to the more complex cases of lubrication and motion.

Figure 1 provides a comparison of experimental and theoretical bearing thermal conductance as a function of axial load at 20 °C. Both experiment and analysis agree closely in both magnitude and trend. The Figure shows that thermal conductance of the dry static bearing responds to axial load to the 1/3 power.

![Graph](image)

Figure 1. Comparison of Analytical Calculation of Thermal Conductance with Experiment for a Static Dry 204-Size Hybrid Bearing at 20 °C
Yovanovich's analysis provides an explanation for the observed trend. His model assumes the mechanism of heat transfer through a non-lubricated static bearing is pure conductance, from the hotter race, through the ball, and to the cooler race (Figure 2) and the driving source of resistance across the bearing arose from the thermal constriction region between the ball and the race contacts. This recognition presumes perfect contact (no asperities), and negligible resistance within the ball and both races.

As such, Yovanovich calculated the thermal resistance across a dry static bearing by modeling the ball and races as semi-infinite half-planes with the Hertzian contact area modeled as the thermal constriction region. The basic equations to calculate thermal resistance across each of the ball to race contacts are:

\[
\begin{align*}
R_i &= \Psi_i/4k_i a_i + \Psi_i/4k_b a_i \quad \text{inner race to ball thermal resistance} \\
R_o &= \Psi_o/4k_o a_o + \Psi_o/4k_o a_o \quad \text{ball to outer race thermal resistance} \\
R_b &= (R_o + R_i) \quad \text{total thermal resistance across the ball}
\end{align*}
\]

Where \( \Psi \) is a non-dimensional geometric factor defined as:

\[
\Psi_n = \frac{2}{\pi} \int_0^{\pi/2} \frac{d\theta}{\left(1 - \frac{a_n^2}{b_n^2} \sin^2 \theta\right)^{1/2}} \quad \text{where } n = i \text{ or } o
\]

The Hertzian contact ellipse, and associated major and minor axes (a and b), can be calculated by a number of existing programs based on classical Hertzian theory, such as BRGS10C [12]. Here, the major and minor axes are related to the applied axial load to the 1/3 power. Thus, the total Hertzian contact area is related to the applied axial load to the 2/3 power.

\[
A_{Hertz} = \pi \cdot \frac{a \cdot b}{4}
\]

Once the resistance across each ball is known, the conductance is determined by taking its inverse. Scaling this result by the number of balls in the bearing yields the total bearing conductance (Equation 6).

\[
G = N \cdot \frac{1}{R_b}
\]

Equation 6 assumes that each ball provides an equivalent thermal pathway as a result of pure axial load, a condition consistent with our experimental setup. As the influence of axial load cancels out in the non-

![Figure 2. Thermal Resistance Across One Bearing Ball](image-url)
dimensional parameter $Y$, only the influence on the Hertzian contact ellipse major axis remains in the thermal conductance equations of 1-3. Thus, analytically, conductance is sensitive to axial load to the $1/3$ power. Both the magnitude and trend predicted by the analysis compared well with the experimental measurements.

Figure 3 shows the effect of the average bearing temperature (average between the inner and outer races) on thermal conductance of a dry (non-lubricated) 204-size hybrid bearing for three different axial loads. The graph shows that the conductance of the dry bearing is not strongly responsive to temperatures, especially when compared to the effect of axial load. This is attributed to the weak dependencies of the relevant material properties (thermal conductivity and Young's modulus) of the steel races and ceramic balls to temperature.

![Graph showing thermal conductance vs. temperature for different axial loads.](image)

**Figure 3. Effect of Average Temperature on Thermal Conductance of a Static Dry 204-Size Hybrid Bearing**

**Oil Lubricated Static Bearings**

Yovanovich's model is a good approximation for some applications, such as dry lubricated bearings with little motion, but once lubrication is introduced into the system, bearing thermal conductance can change significantly. Figure 4 plots the static thermal conductance of a 204-size hybrid bearing in three lubrication states; dry, virgin oil lubricated, and oil lubricated after run-in. The exercised bearing was fully run-in at 6000 RPM, then brought back to 0 RPM for testing. Thermal conductance was measured for three axial loads, over a range of temperatures for each lubrication state.

The significantly higher thermal conductance of the oil-lubricated bearings, in comparison with the dry, warrants attention. To explain the difference between dry and oil lubricated bearings, Figure 5 depicts one of the ball-to-race contact regions. The mechanism of heat transfer is still pure conductance, but the large increase suggests that the lubricant meniscus surrounding the ball contributes a significant heat transfer path by increasing the constriction area at the ball to race interface. Lubricant could also potentially reduce the thermal resistance due to asperity contact at the metal-to-metal contact region; however, the close agreement between the Yovanovich model and the experiment for dry bearings suggest that this represents a secondary effect.

The large increase in thermal conductance implies that the heat path provided by the lubricant meniscus ultimately dominates, masking the influence of the Hertzian contact area. While the thermal conductance across dry bearings are driven by the size of the Hertzian contact ellipse, and thus proves sensitive to
axial load, the masking of the Hertzian contact area by the meniscus explains the minimal influence of axial loading upon the conductance of the oil-lubricated bearing. The dominance of the meniscus also explains the observation that oil lubricated bearings were found to be more sensitive to temperature, a result attributable to the higher temperature dependencies of the lubricant material properties.

![Figure 4. Effect of Average Temperature and Axial Load on Conductance of a Static 204-Size Hybrid Bearing, for Dry and Oil Lubricated Bearings](image)

The conductance values of the virgin bearing are slightly higher than those of the fully run-in bearing, as apparent in Figure 4. This observation was attributed to lubricant loss during the run-in process, as the ball pushed excess oil out of its pathway and centrifugal forces displaced lubricant from the ball. This means that the degree of difference will be dependent upon the initial amount of lubrication and the maximum run-in speed of the bearing.

For space applications, bearings are typically lubricated once, and this lubricant may deplete over the mission duration due to various reasons, such as lubricant migration or run-in. Knowing this, one may construct bounds on static or slow moving conductance values expected throughout the spacecraft mission life. Yovanovich establishes the lower extreme bound, and test measurement of a virgin bearing establishes the upper bound. Lubricant loss can occur throughout the life of the bearing for various reasons including run-in and lubricant migration, meaning the end-of-life conductance would be somewhere between these bounds. This observation, however, does not hold for dynamic bearings.
Oil Lubricated Dynamic Bearings

Figures 6-8 show the influence of average bearing temperature on conductance at different rotational speeds. Each figure represents a different axial load applied to an oil-lubricated bearing that had been run-in at 6,000 RPM.

**Figure 6.** Thermal Conductance of a 101-Size Steel Bearing for a 40-N Axial Load

**Figure 7.** Thermal Conductance of 101-Size Steel Bearing for 84.5-N Axial Load
Several observations arise from these results. First, a distinct difference exists between static and dynamic bearings. Once in motion, the thermal conductance increased with temperature in a linear manner over the data range explored. Moreover, the slope and magnitude increased with axial load. To separate the response to each individual variable, Figure 9, shows the influence of axial load on the bearing at 6000 RPM and at a temperature of 20 °C. Unlike static lubricated bearings that are insensitive to axial loads, the dynamic lubricated bearing responds in a linear manner over the test range.

Figure 8. Thermal Conductance of 101-Size Steel Bearing for 129-N Axial Load

Figure 9. Effect of Axial Load on a Lubricated 101-Size Steel Bearing at 20 °C
Figure 10 re-plots the data to investigate the influence of rotational speed at a constant temperature (20°C). There is a distinct difference between 0 RPM and motion, which becomes more pronounced with increasing axial loads. Furthermore, contrary to intuition, at a given temperature and axial load the thermal conductance proved insensitive to rotational speeds below the run-in speed. The difference between static and dynamic bearings reflects a difference in the predominant heat transfer mechanism.

For moving bearings, conductance through the bearing is not the primary mechanism of heat transfer. Figure 11 illustrates a ball with a lubricant film and a meniscus at the ball to race contact. The dominant mechanism of heat transfer is most likely mass transport, where the lubricant at the meniscus of the hotter race picks up heat, transports it with the ball as it rotates, then deposits heat at the cooler race. The film thickness on the ball becomes a dominant player as it determines the amount of heat transport that occurs. The ball film thickness should not be confused with the elastohydrodynamic (EHD) film thickness, which in the context of our argument has no influence on heat transport.

As the minimum rotational speed tested was 3000 RPM, it stands to reason that there may be a transition region between 0 RPM and 3000 RPM, where speed does influence bearing conductance. Looking through existing literature, Stevens and Todd from ESTL [3] explored the influence of speed on bearing thermal conductance at low speeds. An example of their findings, depicted in Figure 12, indicate an initial decline in thermal conductance, followed by an increase at higher speeds.
Figure 12. Effect of Low Speeds on a 42-mm OD Bearing Lubricated with 11.4 mg of BP135 Oil with 40-N Load; Study by Stevens and Todd [3]

Of additional relevance, Stevens and Todd [3] conducted tests multiple times with consecutively increasing or decreasing speeds. The result was a reduction in conductance occurring with additional revolutions. This is an effect that we also observed when comparing virgin and fully run-in bearings, again most likely due to the effect of excess lubricant being displaced over time. Our observations, and the literature results, indicate that this effect eventually diminishes and tests become repeatable when the bearing reaches a fully run-in state.

Discussion

The objective of this research was to acquire a fundamental understanding of bearing thermal conductance, to establish which variables influence that property, and to determine the dominant mechanisms of heat transfer. Research results indicate that bearing thermal conductance was influenced by a number of interdependent variables, underpinned by the lubrication and the dynamic state of the bearing. Figure 13 provides a global example by plotting the influence of axial load on a 101-size steel bearing at 20 °C. The large differences in trend and magnitude between the dry and lubricated, static and dynamic bearings illustrate that the thermal analyst needs to recognize the assumptions underlying experimental data or analytical models of bearing thermal conductance.

Metal-to-metal contact, through the Hertzian contact ellipses, provided the heat transfer mechanism for dry static bearings. Experimentally, dry static bearings were found to be sensitive to axial load to the 1/3 power, as predicted by the Yovanovich analytical model, and relatively insensitive to temperature. Yovanovich’s method establishes a lower bound for any bearing.

The presence of lubrication can drastically change the bearing thermal conductance. The menisci contributed an additional thermal pathway that overshadowed the conductance through the metal-to-metal Hertzian contact area. In contrast to dry bearings, the influence of axial load was negligible for static lubricated bearings, but the effect of temperature proved significant.
Mass transport of oil was considered the dominant mode of heat transfer in the lubricated dynamic bearing. Both axial load and temperature all affected this mechanism of heat transfer, and ultimately the bearing thermal conductance.

Figure 13. Effect of Axial Load on Thermal Conductance of a 101-Size Steel Bearing at 20 °C
Conclusion

This paper has shown that lubrication or motion affects bearing thermal conductance in both magnitude and sensitivity to operational conditions, such as temperature and axial load. In practice, engineers typically use heritage information from applications using similar bearings within a comparable system to obtain an estimate of bearing thermal conductance. But in some cases heritage information is altogether lacking due to a new or a one-of-a-kind system. Engineers have often used analyses for a first-order approximation of bearing thermal conductance, but the basic analytical equations apply to a non-lubricated, static bearing. Information available on other bearing types is sometimes used as well. However, drastic difference in both thermal conductance magnitude and sensitivity to variables such as axial load and temperature is dependent on bearing lubrication and state of motion. As a consequence, engineers need to evaluate where bearing conductance data came from, what conditions it represents, and whether those conditions reflect the application at hand before using the data to predict temperatures of a rotational device in space.

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References

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