Observations of Spacecraft Bearing Lubricant Redistribution Based on Thermal Conductance Measurements

Yoshimi R. Takeuchi*, Peter P. Frantz** and Michael R. Hilton*

Abstract

The performance and life of precision ball bearings are critically dependent on maintaining a quantity of oil at the ball/race interface that is sufficient to support a robust protective film. In space applications, where parched conditions are intentionally the norm, harsh operating conditions can displace the small reserves of oil, resulting in reduced film thickness and premature wear. In the past, these effects have proven difficult to model or to measure experimentally. This paper describes a study addressing this challenge, where bearing thermal conductance measurements are employed to infer changes in lubricant quantity at the critical rolling interfaces.

In the first part of the paper, we explain how the lubricant's presence and its quantity impacts bearing thermal conductance measurements. For a stationary bearing, we show that conductance is directly related to the lubricant quantity in the ball/race contacts. Hence, aspects of bearing performance related to oil quantity can be understood and insights improved with thermal conductance data. For a moving bearing, a different mechanism of heat transfer dominates and is dependent on lubricant film thickness on the ball.

In the second part of the report, we discuss lubricant quantity observations based on bearing thermal conductance measurements. Lubricant quantity, and thus bearing thermal conductance, depends on various initial and operating conditions and is impacted further by the run-in process. A significant effect of maximum run-in speed was also observed, with less oil remaining after obtaining higher speeds. Finally, we show that some of the lubricant that is displaced between the ball and race during run-in operation can be recovered during rest, and we measure the rate of recovery for one example.

1.0 Introduction

Bearing life and performance is critically dependent on lubricant. Heat transfer is also dependent on lubricant in space, therefore the two are linked. This paper will show bearing thermal properties depend on lubricant and its quantity, then, show how the conductance measurements can be used to infer lubricant behavior.

The requirements for operation of space mechanisms present bearings a very different thermal environment than mechanisms used in a terrestrial environment. In terrestrial applications, convection dominates the cooling mechanism. If air is not enough to cool it, the bearing is typically flooded with lubricant for additional cooling. Thus, bearing thermal conductance tends to be a second or third order effect in most terrestrial applications.

However, in the vacuum of space, essentially no air is present and flooding with lubricant is not feasible. Furthermore, in most cases, the bearing must operate with parched lubricant quantities and perform for years under these conditions. In the absence of convection, bearing raceway temperatures are a product of bearing thermal conductance, heat generation, and the operational environmental temperature. In most

* The Aerospace Corporation, El Segundo, CA
** The Aerospace Corporation, Colorado Springs, CO

vacuum situations, thermal conductance through the bearing and housing becomes a primary driver in maintaining stable temperatures.

Bearing thermal conductance and heat generation are dependent on a complex interaction of secondary factors, including the bearing geometry, internal loads, materials properties, operating and environmental conditions, and lubricant distribution. Heat generation within the bearing is the thermal energy gained and is equal to the mechanical energy lost based on the energy conservation concept. Thus, heat generation is the product of torque and rotational bearing speed. Since torque and speed are commonly measured values, bearing heat generation tends to be a commonly known quantity. Thermal conductance represents the bulk effective heat transfer between inner and outer race; the inverse of thermal resistance. The value of this parameter tends to be poorly known. Bearing thermal conductance and heat generation are dissimilar and respond to thermal and mechanical environments in very different ways. By measuring the lesser known parameter, thermal conductance, a different set of observations on bearing lubricant behavior can be made.

This paper will show that for static bearings, thermal conductance depends on lubricant distribution in the ball to race contacts. For example, a meniscus of oil collected at the interface between a ball and race provides for a larger effective area of contact and heat transfer between the two stationary bodies. This paper will also show that for a bearing in motion, the mechanism of heat transfer differs. The lubricant distribution on the ball dominates the thermal conductance measurement.

In the first part of the paper, we describe the use of thermal conductance measurements to establish the mechanisms of heat transfer, which are sensitive to lubricant distribution. Once the relation between bearing thermal conductance and the lubricant is demonstrated, we show that thermal conductance measurements can be used as a qualitative indicator of lubricant quantity, to monitor behavior during the run-in process, and to deduce the impact of lubricant recovery during bearing rest times.

1.1 Nomenclature

G – (W/°C) bulk effective conductance across the bearing
A_i – (m^2) ball to inner race Hertzian contact area
A_o – (m^2) ball to outer race Hertzian contact area
A – (m^2) average inner and outer ball to race Hertzian contact area
A_{0i} – (m^2) area due to lubricant in ball to inner race contact area meniscus
A_{0o} – (m^2) area due to lubricant in ball to outer race contact area
A_0 – (m^2) average inner and outer lubricant areas
P – (N) axial load

R_i – (°C/W) thermal resistance from the inner race to the ball in a bearing
R_o – (°C/W) thermal resistance from the outer race through the ball of a bearing
R_b – (°C/W) total thermal resistance across a single bearing ball

a_i – (m) major axis of the Hertzian contact ellipse between the ball to inner race
b_i – (m) minor axis of the Hertzian contact ellipse between the ball to inner race
a_o – (m) major axis of the Hertzian contact ellipse between the ball to outer race
b_o – (m) minor axis of the Hertzian contact ellipse between the ball to outer race
a – (m) average of the inner and outer race major axis of the Hertzian contact ellipse
b – (m) average of the inner and outer race minor axis of the Hertzian contact ellipse

k_1 – (W/m-°C) thermal conductivity of the race materials
k_2 – (W/m-°C) thermal conductivity of the ball material

2.0 Test Set Up

A test facility at The Aerospace Corporation was used to measure thermal conductance under controlled thermal and mechanical conditions [1]. The test rig allows the measurement of thermal conductance across a single bearing under controlled axial loads and speeds, and provides the ability to vary and monitor the thermal boundaries, all under a vacuum environment. The speed refers to the angular velocity of the inner race, where the shaft is driven by a motor. The other race is held stationary. A single test bearing is held under a constant load applied by the axial load device. Photos of the test facility used to measure the larger bearing's thermal conductance and its fixture are shown in Figures 2.0.1 and 2.0.2 respectively. A similar test set up was created to measure small and medium size bearing thermal conductance.

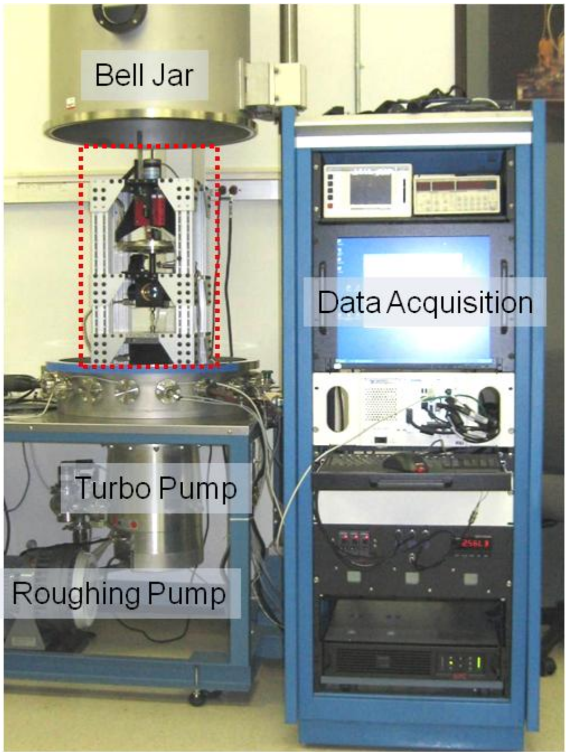

Figure 2.0.1 Test Facility, Including the Electronics, and Part of the Open Vacuum Chamber

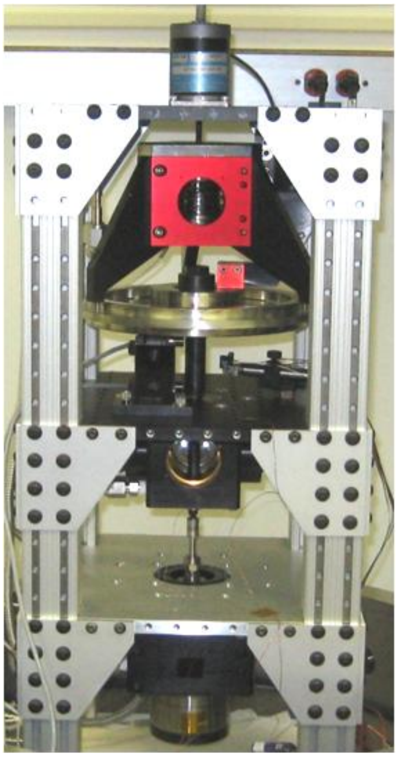

Figure 2.0.2 Bearing Test Fixture

The sizes, geometry, and material information for the bearings that were tested are listed in Table 2.0.1. They are all conventional angular contact bearings with inner ring piloted phenolic retainers. Bearing sizes range from 28 mm to 62 mm outer diameter (OD). None of the test bearings are sealed or shielded.

Table 2.0.1 Bearing sizes, materials, and lubricant quantities

Sample Description	OD (mm)	Race material	Ball material	Dry Bearing weight (g)	Initial lubricant weight (mg)
Large bearing, steel	62	52100 steel	52100 steel	228.54	130
Large bearing, hybrid	62	52100 steel	Silicon Nitride	188.05	130
Medium bearing, steel	47	52100 steel	52100 steel	63.4076	96.9
Small bearing, steel	28	52100 steel	52100 steel	19.8058	49.9

The lubricant used for the bearings was NYE 2001 Penzanne synthetic oil, Nye Lubricants, Fairhaven, MA. The bearings were cleaned and the retainers impregnated with the test oil [2]. A thin film of oil was cast on all surfaces from immersion in dilute solution, and excess oil was removed by centrifugation at 3000 rpm. The dry bearing weight is measured and compared with the weight after lubrication. The difference represents the lubricant weight. Table 2.0.1 documents the initial lubricant weight in each test bearing.

3.0 Test Results: PART 1 – Effect Of Lubricant On Bearing Thermal Conductance

The presence and location of lubricant affects bearing thermal properties. This relationship was explored here by first studying a base case of a dry (no oil), static (non-moving) bearing in vacuum. We then introduced oil into a static bearing to draw a comparison. Data was collected and an example compared the analytical predictions with that of experimental thermal conductance measurements for a small bearing size.

3.1 Test Method

All tests were conducted under steady state thermal environmental conditions, where the thermal and mechanical boundary conditions were fixed and temperatures were allowed to equilibrate. Bearing thermal conductance measurements in section 3.2 through 3.4 were made under static conditions, 0 rpm, and dynamic measurements in section 3.5 were made at 6000 rpm. One of the challenges of the dynamic tests is that the variables, temperature and speed, are not designed to operate independently. To resolve this issue, a matrix of tests was used to perform a parametric study, where both parameters were varied. This enabled the sequential isolation of single independent variables. In this section, the test data set presented is for the average bearing temperature of 20 °C, determined by interpolation over a range of test data.

3.2 Dry Static Bearing

Bearings are generally thermally insulative. The reason for the high thermal resistance is the bearing geometry. Two small contact areas are present, per ball, at the ball-to-inner race and ball-to-outer race interfaces generating a thermal constriction region, through which the heat must pass. A closed form analytical model to calculate the effect of the thermal constriction was developed by Yovanovich [3]. Yovanovich calculated the thermal resistance across a dry, static bearing by modeling the ball and races as semi-infinite half-planes with the elliptical Hertzian contact area serving as the thermal contact region. The basic equations to calculate thermal resistance across each of the ball-to-race contacts are:

$R_i = \Psi_i/4k_1a_i + \Psi_i/4k_2a_i$ — inner race to ball thermal resistance (3.1)

$R_o = \Psi_o/4k_2a_o + \Psi_o/4k_1a_o$ — ball to outer race thermal resistance (3.2)

$R_b = (R_o + R_i)$ — total thermal resistance across the bearing (3.3)

where Ψ is a non-dimensional geometric factor accounting for the ball contact areas and defined as:

$$\Psi_n = \frac{2}{\pi}\int_0^{\pi/2} \frac{d\theta}{\left(1 - \frac{a_n^2 - b_n^2}{a_n^2}\sin^2\theta\right)^{1/2}} \qquad (3.4)$$

Where the subscript *n* is designated either *i* or *o* for the inner or outer race. The Hertzian contact area is elliptical in shape and the geometry is represented by:

a_n – major Hertzian contact axis

b_n – minor Hertzian contact axis

The relationship between the major and minor axis dimensional geometries and load, P, is documented in the reference by Yovanovich [3]. A simplified summary:

$$a_n \propto P^{1/3} \tag{3.5}$$

$$b_n \propto P^{1/3} \tag{3.6}$$

Conductance is the inverse of the resulting calculated Yovanovich bearing thermal resistance equations.

$$G = \frac{1}{R_b} \tag{3.7}$$

Simplification is required to gain a basic understanding of the relationship between conductance and axial load and area. Substituting and simplifying with the following assumptions:

$$a_i \approx a_o \tag{3.8}$$

$$b_i \approx b_o \tag{3.9}$$

We obtain a relationship between bearing thermal conductance and load.

$$G \propto P^{1/3} \tag{3.10}$$

The Hertzian contact area is also defined as:

$$A_n = \frac{\pi}{4} \cdot a_n \cdot b_n \propto P^{2/3} \tag{3.11}$$

Thus,

$$A \approx A_n \propto P^{2/3} \tag{3.12}$$

and

$$G \propto \sqrt{\mathrm{A}} \tag{3.13}$$

Equation 3.13 also means that the dependence of conductance on contact area weakens as the contact area increases. As the contact area grows, conductance becomes less sensitive to further increases. Equation 3.10 shows a similar, but stronger attenuation for the effect of increasing preload on conductance.

The relationship between a bearing's thermal conductance and axial load is demonstrated in Figure 3.2.1. This figure compares measured and calculated conductance values for the small (Table 2.0.1), dry (non-lubricated), static (non-moving) 52100 steel bearing. The conductance analysis utilized the Yovanovich method. Note the conductance vs. axial loads relationship demonstrates the $P^{1/3}$ curve, where low values of axial loads result in high conductance sensitivity, but with higher loads the sensitivity gradually diminishes, evidenced by a decreased slope. A comparison between the calculated and measured values of thermal conductance indicates that the trends correlate well.

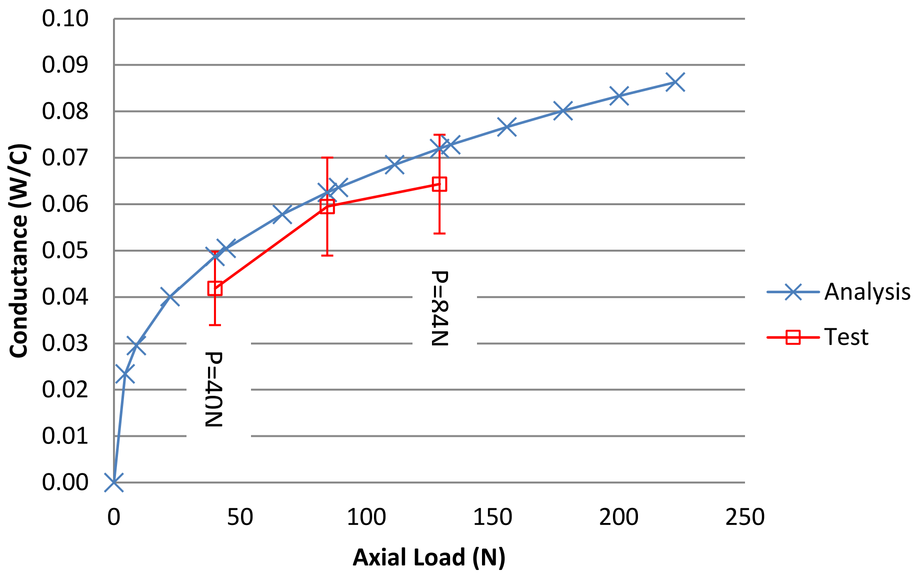

Figure 3.2.1 Theoretical and Experimental Conductance vs. Axial Load for a 28-mm OD Dry Static Steel Bearing

3.3 Oil Lubricated Static Non-Run-In Bearing

Once oil was introduced into the bearing, thermal conductance increased significantly and became relatively insensitive to axial loads. Figure 3.3.1 compares the conductance of a dry static bearing with the same bearing after the addition of oil. The small bearing was used for this experiment, after oil was added according to the method described above. The bearing was not operated to run-in the lubricant.

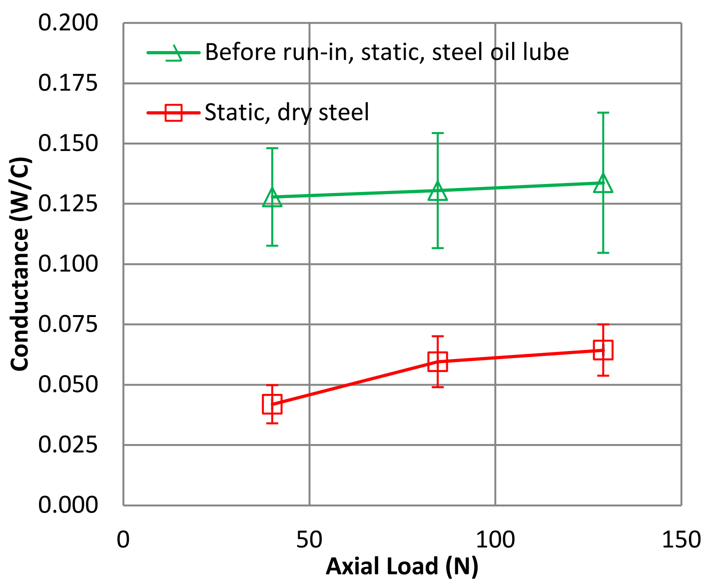

Figure 3.3.1 Experimental Thermal Conductance vs. Axial Load for a Static 28-mm OD Oil Lubricated Non-Run-In and Dry Steel Bearing

The green triangles in Figure 3.3.1 (oiled bearing) show a significant increase in conductance over the red squares (dry bearing); this is attributed to the increase in ball-to-race contact area due to the presence of the oil between the contacts. We surmise that increased conductance is due primarily to the oil meniscus surrounding the contact and secondarily due to the liquid mediated Hertzian contact (filling the microscopic pores between the rough contacting surfaces). In essence, the oil increases the thermal pathway, decreasing resistance and increasing conductance.

The decreased sensitivity to axial loading can also be attributed to the increased initial area due to the presence of the oil lubricant. With a much larger initial area, a subsequently greater increased area change is needed to affect bearing thermal conductance. Figure 3.3.2 shows the relation between total ball-to-race contact area and conductance. The red dots indicate the increased contact area in a dry bearing showing the change in conductance as a function of load. The yellow dots show the increased contact area in an oil-lubricated bearing due to the same increase in load, and its influence on conductance. The change in an oil-lubricated bearing is dramatically less pronounced because the relevant contact area is already an order of magnitude larger due to the oil.

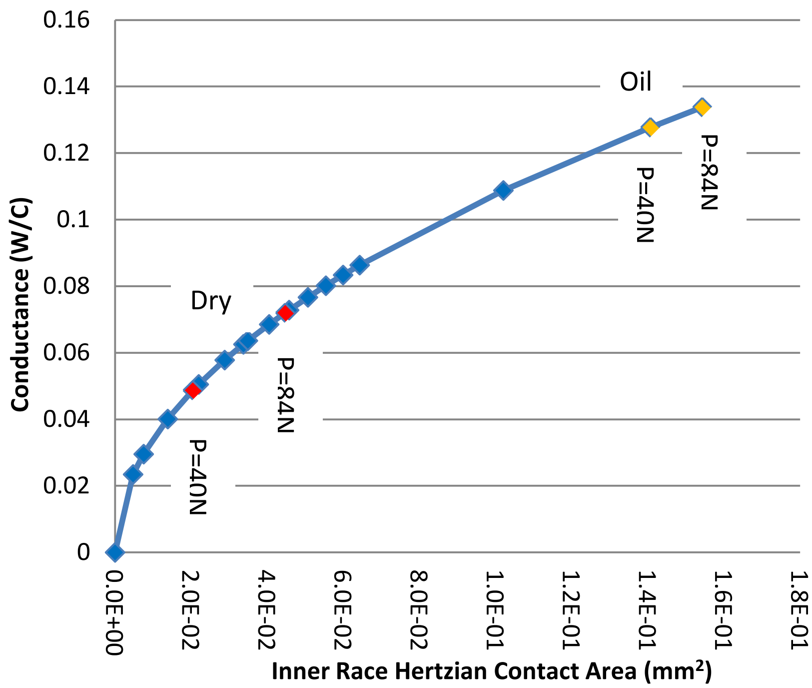

Figure 3.3.2 Theoretical Conductance vs. Total Inner Race Contact Area for a 28-mm OD Steel Bearing: Change in Area for a Dry and Oil Lubricated Case

The exact thermal conductance magnitude depends on the initial quantity and distribution of the lubricant. If we assume that the contact area of the lubricant, A_0, is constant, then, bearing thermal conductance is proportional to the square root of the total area of the lubricant and the ball to race contact:

$$G \propto \sqrt{\mathrm{A} + \mathrm{A}_0} \qquad (3.14)$$

The ball to race contact area, A, is still proportional to the axial load to the 2/3 power:

$$A \propto P^{2/3} \qquad (3.15)$$

Measuring conductance, G, the Yovanovich model can be used to estimate the total area $A+A_0$ [3], then, the ball to race contact area can be calculated based on knowledge of the load, P [3]. The assumption is that the thermal resistance due to the lubricant is negligible, despite the thermal conductivity of oil being only 0.3% of the 52100 steel. The thermal resistance is still insignificant because the thickness of the lubricant in the meniscus at rest is, on average, approximately 0.3 microns. This assumption introduces about a 3.5% error into the analysis, where the resulting contact area is under-estimated by that amount. Assuming that the thermal resistance across the metal and lubricant contact is small compared to the resistance generated due to the thermal constriction of ball to race geometry, the Yovanovich model can be used to estimate the lubricant area once the thermal conductance is measured across the bearing.

Thus, the total lubricant area, A_0, can be determined. Based on the analysis method, the contact areas due to the lubricant and the metal-to-metal Hertzian contact are shown in Table 3.3.1. The values depicted represent the lubricant and metal contact area between the inner race and one ball. Note that the initial lubricant area is dependent on the amount applied to the bearing system before testing began. In this case, the 62-mm OD bearing had a larger relative lubricant amount in the ball-to-race pathway than the smaller 28-mm OD bearing. In the next section, we'll explore how the run-in process impacts lubricant quantity in the ball-to-race contact.

Table 3.3.1 Static Bearing Ball-to-Race Hertzian and Lubricant Area Before Run-In

Bearing Details			Area – (Inner Race)			
Bearing Size	**OD (mm)**	**Load, P (N)**	**Hertzian Contact Area (mm^2)**	**Lubricant Area (mm^2)**	**Total Area (mm^2)**	**Lubricant/ Hertzian Area (%)**
Large bearing, steel	62	133	8.2E-02	45.E-02	53.E-02	550
Small bearing, steel	28	129	4.5E-02	11.E-02	15.E-02	245

3.4 Oil Lubricated Static Run-In Bearing

Oiled bearings (small and large, steel) were then run-in in the following manner: As the bearings turned continuously at 6000 rpm, the conductance, temperatures, and heat generation were observed to gradually decrease. Changes were most pronounced early in the test and gradually decreased in intensity over time. Each bearing was considered to be completely run-in when no additional changes in these variables could be detected with additional operational time. Measurements of thermal conductance resumed after bringing the bearing to rest (static conditions) and compared with the results before run-in conditions were established.

Observations on the run-in process indicated that the larger bearing runs-in significantly faster. While the small bearing took 5 days to run-in, the large bearing took significantly less time (less than 3 hours). It was not known exactly how long run-in conditions were reached for the largest 62-mm OD bearing tested because the thermal equilibrium in the test set up itself took 3 hours to occur and by then the bearing was run-in. A possible explanation for the much shorter run-in times for the larger bearing was that a much larger centrifugal force on the larger 62-mm OD bearing caused more lubricant to be thrown out initially.

Table 3.4.1 shows the impact of a 6000-rpm run-in on lubricant in the ball-to-race pathway. The data presented in the table represents the lubricant and metal contact area between the inner race and one ball, and provides a relative feel for the amount of lubricant present in the raceway.

Before run-in, the lubricant area in the race for the large bearing was significantly larger than the small bearing, but *after* run-in, the reverse occurred. While the lubricant contact area size was initially larger in the 62mm OD bearing, relatively, the lubricant was only 138% larger than its Herzian contact area compared with 183% for the smaller 28mm OD bearing.

This large decrease in lubricant was reflected in a large drop in thermal conductance measurements described next. Figure 3.4.1 shows the bearing thermal conductance as a function of axial loads for a 28-mm OD bearing, both before and after run-in and Figure 3.4.2 depicts the results for the large 62-mm OD bearing.

Table 3.4.1 Static Bearing Ball-to-Race Hertzian and Lubricant Area After 6 krpm Run-In

Bearing Details			Area – (Inner Race)			
Bearing Size	**OD (mm)**	**Load, P (N)**	**Hertzian Contact Area (mm^2)**	**Lubricant Area (mm^2)**	**Total Area (mm^2)**	**Lubricant/ Hertzian Area (%)**
Large bearing, steel	62	133	8.2E-02	11.4E-02	19.6E-02	138
Small bearing, steel	28	129	4.5E-02	8.2E-02	12.7E-02	183

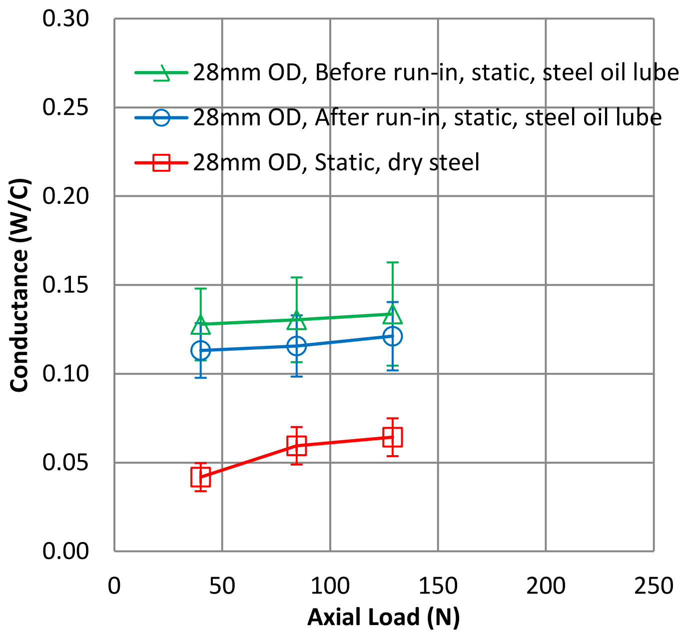

Figure 3.4.1 Comparison of Thermal Conductance vs. Axial Load for an Oil Lubricated Static Before Run-In, After Run-In, and Dry Conditions for a 28-mm OD Steel Bearing

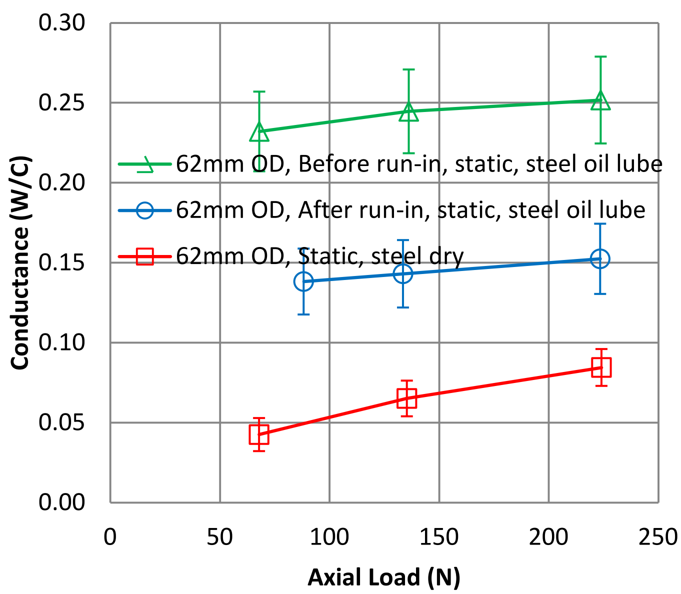

Figure 3.4.2 Comparison of Thermal Conductance vs. Axial Load for an Oil Lubricated Static Before Run-In, After Run-In, and Dry Conditions for a 62-mm OD Steel Bearing

We find changes in conductance that are caused by this run-in process. Bearing thermal conductance decreases in magnitude after run-in in both cases, as expected due to the smaller lubricant area. However, the thermal conductance in the larger bearing decreases significantly more than the smaller bearing, possibly due to a combined initial larger amount of lubricant and higher centrifugal force acting on the bearing during run-in, throwing out more lubricant. The final lubricant contact area is slightly greater in the larger bearing and likewise, the thermal conductance is slightly higher for the larger bearing.

In both cases, the final thermal conductance is still higher than the dry condition as would be expected. The dry case represents the lowest possible bearing thermal conductance value for a given bearing, geometry, materials, and load condition.

The total amount of lubricant in the bearing was measured before and after run-in for three different bearing sizes. The results are shown in Table 3.4.2, with additional initial weight data from a fourth bearing.

Table 3.4.2 Lubricant Loss as a Function of Run-In Speed

Bearing Details			Lubricant Weight		
Bearing Size	**OD (mm)**	**Dry Bearing Weight (g)**	**Initial Lubricant Weight (mg)**	**After 6krpm Run-in (mg)**	**After 10 krpm Run-in (mg)**
Large bearing, steel	62	228.54	130	100	not weighed
Large bearing, hybrid	62	188.05	130	90	70
Medium bearing, steel	42	63.4076	96.9	not operated	not weighed
Small bearing, steel	28	19.8058	49.9	39.2	not operated

In each case, we find that the conductance reached a steady state after prolonged operation at its run-in speed. The measured amount of lubricant loss can be attributed to the initial quantity and maximum run-in speed. In the case of the large hybrid bearing the weight fell further after additional testing at 10 krpm, showing that as run-in speeds increase, the amount of lubricant left in the bearing decrease. In section 4.4 below, we will show how this decrease affects conductance. Even though most of this lubricant weight is not distributed in the ball to race pathway, there is a corresponding drop in lubricant in the critical areas, resulting in lower bearing thermal conductance with increase run-in speeds.

3.5 Oil Lubricated Dynamic Run-In Bearing

Next, dynamic (constant speed motion at 6000 rpm) thermal conductance measurements were taken at various thrust loads for the small and large sized bearings. While the bearing is in motion at high speed, the heat transfer mechanism changes. For a static bearing, the mechanism of heat transfer was conduction through the contact pathway, through the ball, and finally through the outer contact pathway. However, once the bearing rotates at a significant speed (such as 6000 rpm), our data suggests that the mechanism of heat transfer changes to mass transport, as warm lubricant picks up heat at the hotter raceway and gets transported with the ball. It then deposits heat at the cooler raceway. Thus, the lubricant film thickness on the ball becomes important, and the Yovanovich analysis no longer holds. Comparisons of the conductance during both dynamic and static states as well as oiled and dry conditions were made for both bearing sizes in Figures 3.5.1 and 3.5.2.

For all of the dynamic test cases, bearing thermal conductance was stable for a given set of conditions once the bearing had fully run-in. To produce the test data shown below, the test fixture thermal conditions were adjusted and the data set was interpolated for a constant average bearing temperature of 20°C throughout. Despite the differences in bearing sizes, the behavioral *trends* were similar between the 28mm OD and 62mm OD bearing sizes. Conductance *magnitudes*, however, were very different, with the larger bearing having much higher conductance than the smaller bearing and rising more steeply with axial load.

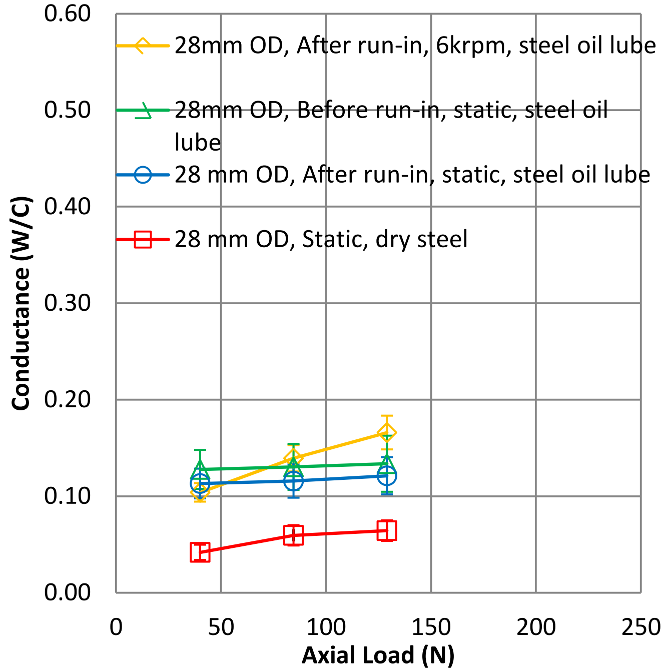

Figure 3.5.1 Comparison of Thermal Conductance vs. Axial Load for an Oil Lubricated Static And Dynamic States of Motion for a 28-mm OD Steel Bearing

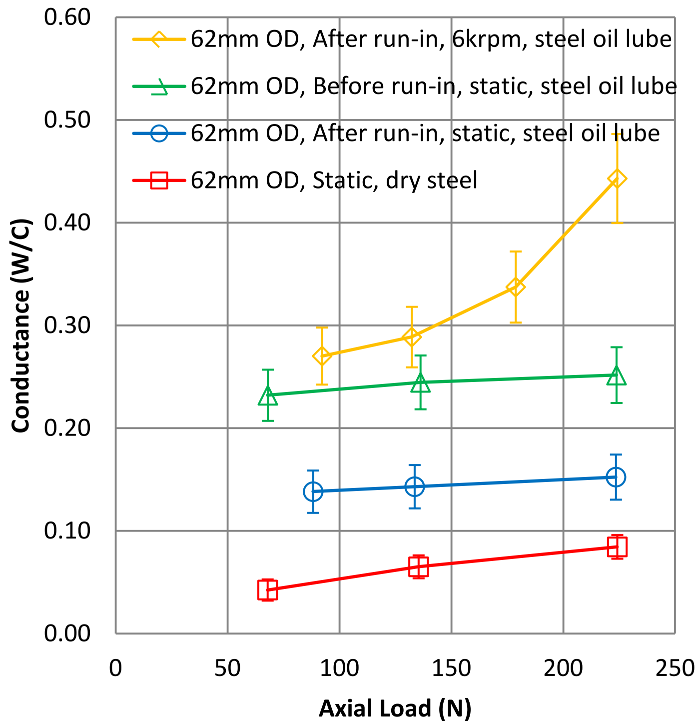

Figure 3.5.2 Comparison of Thermal Conductance vs. Axial Load for an Oil Lubricated Static And Dynamic States of Motion for a 62mm OD Steel Bearing

The reason for sensitivity to load is not yet fully understood, and is the subject of ongoing investigation. One possible explanation being explored is based on the mechanism of heat transfer. Once the bearing moves at a significant speed, the heat transfer process is very different than a static bearing. Significant motion would prevent conduction process through the contact points and ball. Instead, we believe that mass transport takes place, where the lubricant picks up heat at the hotter race, gets carried by the ball as it rotates, then deposit heat at the cooler race. As the axial load increases, this band of lubricant exposed to the mass transport process increases, resulting in greater heat transfer rates, thus higher bearing conductance.

4.0 Test Results: PART 2 –Bearing Lubricant Behavior Observations Based on Thermal Conductance Measurements

This section explores lubricant behavior observed through conductance changes during the run-in process. The impact of changing the maximum run-in speed was studied. The impact of bearing rest was also described below. From these measurements, we infer changes in lubricant quantity.

4.1 Test Method

The bearing inner ring raceway was rotated in vacuum at constant speed and environmental temperature. After thermal equilibrium was established, the conductance was measured while operating at the indicated speed. The speed was then changed, and the system was allowed to establish a new equilibrium before the next data point was measured. The resulting thermal conductance was collected for speeds ranging from a minimum of 0 rpm to a maximum of 10,000 rpm.

Because we were establishing the influence of run-in, a test matrix was *not* performed on each data point, making it impossible to separate out the influence of temperature and speed. Hence the temperature was allowed to vary with speed in the data set shown below.

4.2 Influence of Run-In on Bearing Lubricant Distribution

Two medium-sized bearings were tested as described above. The first bearing, labeled medium A in Figure 4.2.1, was freshly lubricated and had not been previously run-in. The second bearing, labeled medium B in Figure 4.2.2, had been run-in and tested previously but rested for months before the tests shown here. The curves on each of the plots show a set of runs that has the bearing tested at successively increasing or decreasing speeds.

The initial running cycles of the virgin bearing (blue diamonds, Figure 4.2.1) show erratic behavior caused by rapid changes in lubricant distribution. The high initial conductance at rest indicates a significant amount of oil in the menisci. As the first speed cycle returned to rest from 10,000 rpm, the conductance was less than initial, but still above the fully run-in value. With additional speed cycles, the conductance settled down to a continuous decrease with speed, showing the hypothetical effects of heat transport described above. After the multiple speed cycles, the static conductance leveled off at approximately 0.10 W/°C. At this point, we believe the bearing was fully run-in, as excess oil between the ball and race interfaces were displaced.

The used bearing, Figure 4.2.2, had been through a similar suite of testing, and then rested for months. We found that although the high-speed conductance changed little with additional run-in cycles, the conductance at rest (and, by inference, the quantity of oil in the static menisci) fell slowly with progressively greater accumulated operating time and as the bearing became run-in. The static conductance began at 0.13 W/°C, and fell with additional speed cycles to a stable value nearly equal to the static conductance of the virgin bearing after several speed cycles.

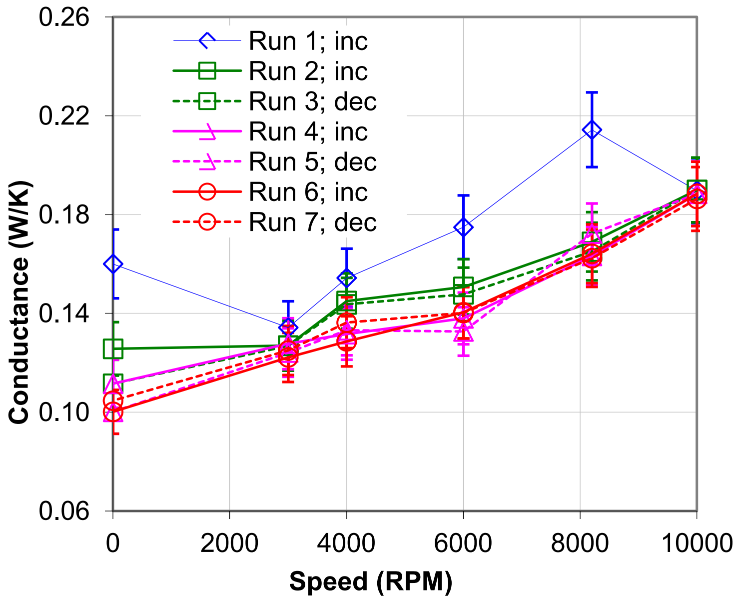

Figure 4.2.1 Run-In Effects on Thermal Conductance of a Virgin Bearing; Medium Bearing A

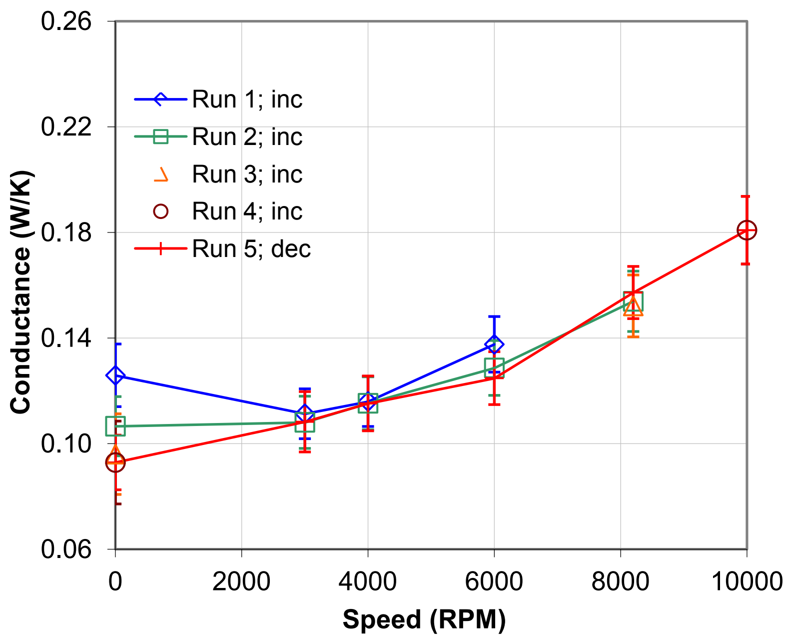

Figure 4.2.2 Run-In Effects on Thermal Conductance of a Previously Used Bearing; Medium Bearing B

We conclude form this that a bearing of this size, lubricated as these were, will have enough excess oil to enable a static conductance of 0.16 W/°C. After run-in, a portion of this oil is displaced from the rolling path, but remains close, while another portion is displaced to a location where it is far from the rolling path. This loss results in a smaller oil meniscus and conductance of 0.10 W/°C. With prolonged rest, the oil that remained close by is able to creep back to the ball-to-race interface, causing a recovery of the conductance to about 0.13 W/°C.

In both cases, additional run-in resulted in a decrease in static thermal conductance. However, the dynamic bearing thermal conductance behavior began quite differently between A and B. For the virgin bearing A, the thermal conductance began high, but with additional run-in the value decreased. We suggested earlier that the mechanism of heat transfer in a dynamic bearing was mass transport. The test data indicates that the lubricant film thickness decreases with time then stabilizes after the bearing is completely run-in. That film thickness is established by the centrifugal forces generated by the maximum run-in speed. The film thickness on the ball should *not* be confused with the Elastrohydrodynamic (EHD) film as the EHD film is the minimum oil thickness separating the ball and race during operation. Instead, the film thickness that we are referring to is the lubricant riding on the outer diameter of the ball.

The second bearing, B, had previously been run-in and clearly shows that the dynamic conductance values are repeatable. This suggests that the lubricant film on the ball does not recover after run-in.

The testing of both medium bearings varied in the number of start/stop cycles and the operational time, but both had seen maximum run-in speeds of 10,000 rpm and were operated at the same temperature. The conductance values after multiple speed cycles, shown in Figures 4.2.1 and 4.2.2, are very similar. Both the static and dynamic bearing thermal conductance values ended up comparable, suggesting final run-in lubricant quantity and distribution in the bearing is similar and is governed primarily by maximum run-in speeds.

<u>4.3 Influence of Maximum Run-In Speed on Lubricant Loss</u>

The bearing thermal conductance profile changes, as the bearing is run-in at even higher speeds (20,000 rpm). This effect is shown in Figure 4.3.1.

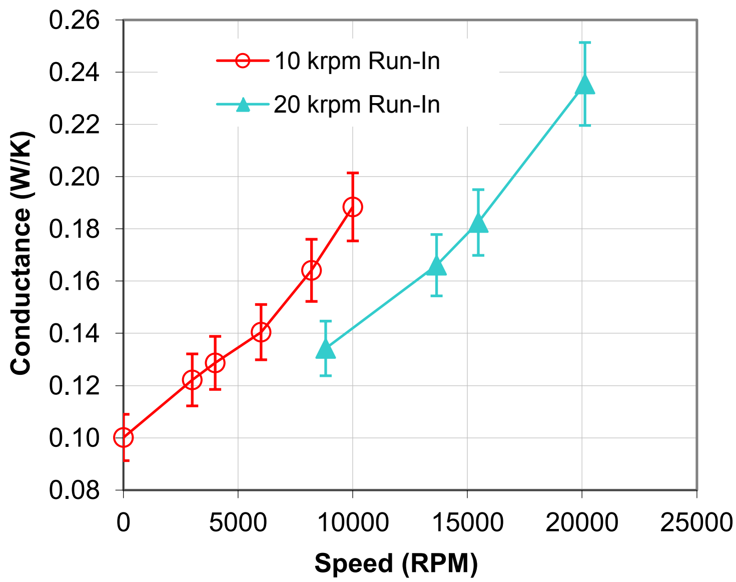

Figure 4.3.1 Effect of Preconditioning Bearing at 10 krpm and 20 krpm Run-in on Final Bearing Thermal Conductance for Medium Sized Steel Bearings

After operating at higher speeds, the thermal conductance decreases indicating that more oil has been displaced. We surmise that this oil has been driven away predominantly by higher centrifugal forces and possibly to a lesser extent by higher thermal gradients.

<u>4.4 Influence of Rest Time on Lubricant Redistribution</u>

For the final series of tests, one medium bearing (Medium Bearing B) was allowed to rest to observe the recovery of oil that had been displaced in earlier tests. After the suite of testing shown above, the test fixture remained dormant for three months at room temperature. During this time, the bearings did not spin. However, at three points in time during this interval, the static conductance was measured. Figure 4.4.1 indicates a slow growth in the static conductance. This growth is consistent with the recovery of oil at the ball/race menisci.

We find a slow equilibration process, resulting in a thermal conductance increase of about 45% after resting for four months. This corresponds to a lubricant area increase of over 160% in the ball-to-race pathway for a bearing at static rest. This suggests a possible strategy for remediation of mechanisms that are suffering from lubricant depletion.

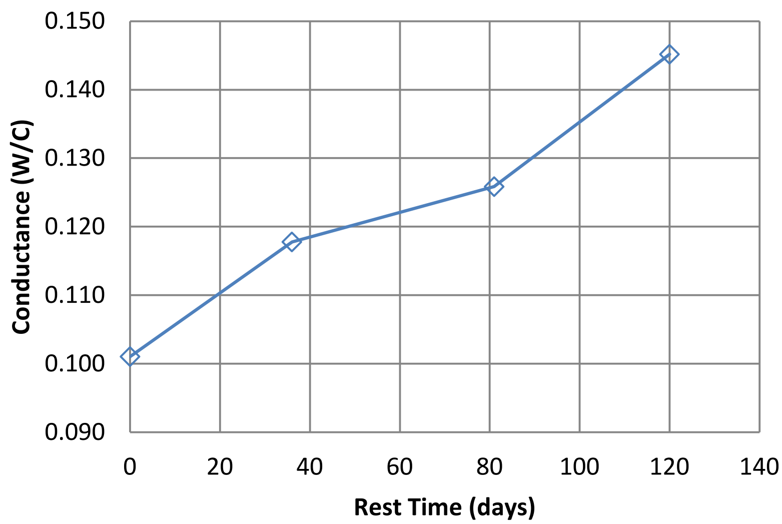

Figure 4.4.1 Static Bearing Thermal Conductance vs. Rest Time For Steel Medium Bearing A

5.0 Summary and Conclusions

The lifetime of precision bearings in space applications, where typically only a single charge of lubricant is expected to provide for the entire mission life, is closely tied to the lifetime of that lubricant. Mechanisms often function well until the lubricant is depleted, and then rapidly deteriorate. Some lubricant may be lost to thermo-chemical degradation during normal operation. However, in many situations much of it is simply displaced from the critical interfaces. Among this displaced lubricant some may be recovered by surface creep when the force balance is changed, while some is unrecoverable due to its having been thrown from the bearing or located with other impediments to migration. It would be helpful to understand the mechanisms of lubricant loss and recovery.

These experiments have shown that thermal conductance measurements are sensitive to lubricant quantity. For this reason, thermal conductance measurements can be used as a tool to make observations on lubricant behavior, such as run-in effects, and impact of rest times on lubricant redistribution. We show that conductance after run-in can be sensitively dependent on bearing geometry and operating speed, and less dependent on initial lubricant state, in the conditions studied here. These results imply that lubricant distribution is impacted by operational conditions, changing thermal conductance in logical ways. Some of the oil that is displaced by run-in can be recovered during rest, presumably by surface migration back to the contact, increasing conductance, but this process required months of time in the case studied here.

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