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Experimental Analysis of the Wear, Life and Behavior of PTFE Coated Thrust Washer Bearings Under Non-Axisymmetric Loading©

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In this work the behavior and life of a PTFE coating on a flat thrust washer bearing is investigated. The thrust washer bearing is located between a helical gear and its carrier, and is subjected to non-axisymmetric loading and wear. The volume of worn material is approximated by measuring the difference in height between the worn and unworn surfaces. It was also found that the surface roughness of tested washers increases with the severity of wear, in most cases. After a finite number of cycles the effective coefficient of friction between the surfaces increases, suggesting that the coating is wearing off and losing effectiveness. The rate at which the coating wears off also varies with load and speed, hence, there is a region of operation that minimizes the wear and friction.

KEY WORDS

Wear; Scuffing; Bearing; Surface Films/Coatings; Boundary Lubrication

INTRODUCTION

The wear and distress of thrust washer bearings is a significant engineering problem that can be approached in a number of ways. Coatings are often used to either harden the bearing surface in an attempt to reduce wear, or provide a solid lubricant to reduce wear. PTFE is primarily used as a sacrificial solid lubricant. From this point forward, bearing will refer to the stationary coated thrust washer bearing. The bearing has no contours, grooves, or purposefully applied textures. The coating's life is the number of cycles during which the coating still affects the interaction between the surfaces and is not yet worn away. The coating life is thus very difficult to quantify. However, the coating wear is measured by use of a profilometer to measure the depth of wear that has occurred in relation to a portion of the bearing surface which is not in contact during operation. The performance of the coated bearing is quantified by the effective coefficient of friction and the temperature near to the bearing surface.

It is the goal of this work to quantify and map the behavior of a PTFE thrust washer configuration under various conditions. The bearings are run at constant loads and speeds for prescribed amounts of time, while the temperature and frictional torque of the bearing are recorded. The test allows the bearing to be tested over an acceptable length of time, while still inducing severe wear and distress upon the bearing.

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General Thrust Washer Bearing Literature

Work concerning flat thrust washer bearings as described earlier is very scarce although a great deal of work does exist that relates to the thrust washer bearing case. Cameron and Wood *(3)* formulated a model of a grooved parallel surface thrust bearing, in which the hydrodynamic lift was created by a converging gap induced by thermal deformations. Similarly, Taniguchi and Ettles *(15)* modeled hydrodynamic and thermo-elastic behavior of radially grooved parallel surface thrust washers.

One of the few experimental investigations of thrust washers was the work of McClintock *(11)*. The washers tested contained a single groove across the diameter of the bearing face. McClintock investigated the wear of thrust washers using a variety of lubricant types. Thus, this investigation was less concerned with the bearing behavior and mechanisms governing their behavior than the effect of different lubricants on bearing behavior. The paper does suggest, though, the existence of full film hydrodynamic lubrication at certain loads because of the lack of wear at these loads.

Coatings, and in particular PTFE (polytetrafluoroethylene) coatings, have been in use in tribological applications such as bearings for many years. There are various methods by which the PTFE coating is applied and secured to the surface. Most of these methods were developed in attempts to reduce the wear of the coating while still maintaining its effectiveness. In the work by Bahadur *(2)*, a few of these methods are investigated.

Under dry conditions, which is not the case in this work, PTFE can reduce the coefficient of friction (COF) to values close to 0.04

NOMENCLATURE

H	= surface hardness
K	= Archard's wear coefficient
k	= dimensional wear coefficient
L	= normal load
L^*	= normalized load, L/L_{max}
N	= rotational speed
N^*	= normalized rotational speed, N/N_{max}
R_a	= average roughness
T^*	=normalized temperature, T/T_{max}
ω	= wear volume

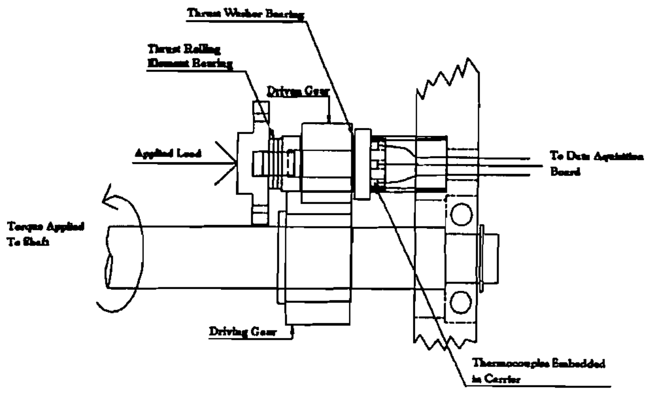

Fig. 1—Diagram of thrust washer bearing testing configuration within the test rig.

under high loads and low speeds (Rabinowicz, *(12)*). However, the COF can be much higher for lighter loads, higher speeds, and different coating compositions. Thus the COF of PTFE decreases with higher loads. Lubrication will also effect the COF in that it can provide hydrodynamic load support in the form of a boundary lubrication regime.

In the work by Freidrich, et al. *(7)*, the effect of reinforcements and coating combined with wear-resistant polymers is investigated. It was shown that pure PTFE will actually provide the lowest COF but also with the highest wear rate. The measured COF ranged from 0.2 to 0.4, which seems fairly high in comparison with other works.

In the works by Simmons, et al. *(14)* and Choudhary, et al. *(5)* the use of PTFE on hydrodynamic thrust bearing pads in hydrogenerators is investigated. The COF found during these tests were in the range of 0.15 for low loads and 0.055 for higher loads. It is also interesting that the COF sharply decreased from low load to high load conditions. The same phenomenon is also seen in the current work.

During the current work, the thrust washer bearing has often reached a point of distress under certain loads and speeds (Jackson and Green, *(8)*). This point of distress is marked by a sudden increase in the COF and the bearing temperature. While the bearing is in distress, material is often transferred between bearing surfaces and/or worn away. Under severe conditions the contacting surfaces can even weld together and cause the test rig to seize. Since this also occurs mostly at high speeds, it fits the definition of scuffing failure and wear as described in Williams *(16)*.

In the work by Alzoubi, et al. *(1)*, the resistance of surfaces to scuffing can be increased by a low-friction coating, specifically amorphous carbon. Hence, it seems likely that PTFE may also provide the same added scuffing resistance and improved bearing performance. This work addresses this possibility.

TEST EQUIPMENT

The test rig allows for controlled variation of the operational parameters governing the tribological behavior of the bearing. The parameters that most affect the life of the bearing and its tribological behavior are believed to be thrust or axial load, rotational speed, lubrication supply, lubrication properties, and the geometry of the bearing. The test rig also records pertinent real-time data from the bearing, namely, the frictional torque transferred through the bearing and the temperature near the bearing and of the fluid exiting the bearing. See Fig. 1 for a diagram of the rig. Thermocouples indicate the temperature near the bearing. Four thermocouples are embedded in the stationary carrier next to the bearing. Another thermocouple placed at the bottom of a plate positioned directly below the bearing gathers the exiting lubrication and reads its temperature. The thermocouples have an accuracy of approximately +/- 1.1% of the maximum value. The lubricant, automatic transmission fluid, is applied to the bearing through a small hole in a pressurized cylinder directly above the bearing. Automatic transmission fluid is used because the bearing's industrial application is within an automobile's automatic transmission.

The frictional torque is calculated from the power output of the electric motor. At a constant speed, the power will vary only with additional torque. The bearing is loaded through a lever and pulley system that is manually loaded with dead weights. Additional details of the test rig are given in Jackson and Green *(8)*. The only major modification to the rig since than has been the method by which the frictional torque is measured. The recorded frictional torque is accurate to approximately +/- 4.0% of the maximum value.

COATED WASHER SPECIFICATIONS

The stationary thrust washer bearing is made up of four main components. These components are pictured and labeled in Fig. 2. These components are used in a planetary gear-set in an automatic transmission, which is only partially simulated by the test rig. A helical pinion gear is loaded axially, such that the loading of the thrust washers is non-axisymmetric (Jackson and Green, *(8)*). A set of needle bearings upon which the gear rides governs the angle to which the gear can tilt perpendicularly to the axis of rotation. The axial face of the gear comes in direct contact with a single steel round washer. This steel round washer is free to rotate and

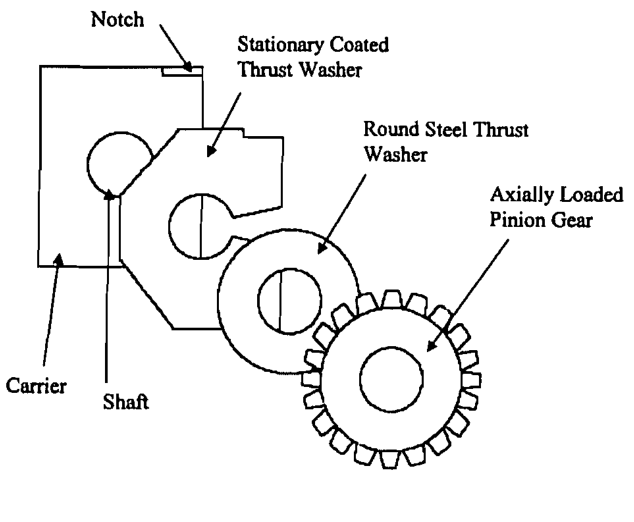

Fig. 2—Schematic of main components of the tested thrust washer bearing.

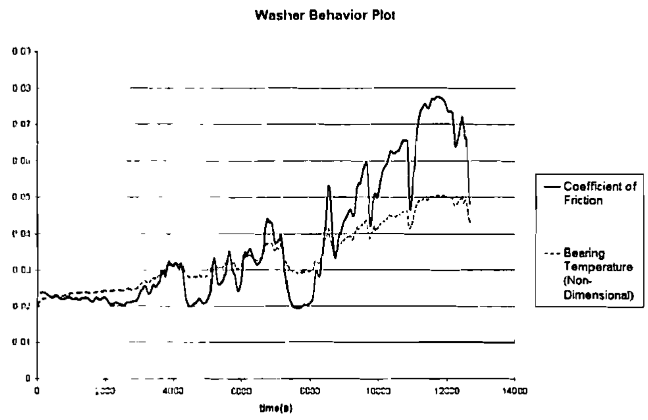

Fig. 3—Coefficient of friction and bearing temperature plotted *vs.* time for a loaded PTFE coated thrust washer bearing.

also tilt in relation to the gear. The steel washer is in direct contact with the stationary coated washer. The coated washer is kept stationary by a notch that fits onto a raised portion of the carrier surface. The stationary coated washer also has a radial gap to allow fluid into the bearing. This gap is not located in the region of contact due to the non-axisymmetric loading conditions. The effect of the radial groove on the behavior of the stationary washer is unknown.

The coating on the stationary washer has essentially the following composition: The washer itself is carbon steel. On top of the steel a bronze porous layer approximately 0.2 to 0.35 mm is sintered onto the washer. A mixture of PTFE and Pb is rolled into this porous layer of bronze. On top of the porous bronze layer a 10 to 30 μm layer of PTFE/Pb is added. The PTFE/Pb coating is composed of approximately 80% PTFE and 20% Pb. Thus, the coating layer is approximately a total of .21 to .38 mm thick, although only the outer surface is fully coated with PTFE/Pb.

TEST METHODOLOGY

All speeds and loads are normalized by each of the maximum values tested. Thus, the normalized load is given as, $L^* = L/L_{max}$, and the normalized rotational speed is given as, $N^* = N/N_{max}$. Each test is run for the same number of cycles unless the test interruption criterion is met. This allows for the differences in wear between the tests to be easily compared.

Test Interruption Criteria

At certain loads and speeds the effective coefficient of friction (COF) of the bearing rises suddenly from a value less than 0.04 to a value at or above 0.06. The COFs may for unknown reasons elevate for only a brief moment suggesting that the bearing is able to dynamically correct itself and return to a lower COF. The temperature at these elevated coefficients rises quickly and may cause damage to the test rig. During these moments of bearing distress the local temperature might rise above the boiling point of the lubricant in about 30 seconds. This boiling point causes lubricant in a gaseous or vapor form to exit the bearing and also the test rig. This occurrence is undesirable since it drastically decreases the effectiveness of the lubricant, and also signifies the occurrence of scuffing. Thus, an average friction coefficient of 0.06 over a time period of 30 seconds is considered the distress point or failure criteria for these tests. The tests, however, will not be interrupted if this criterion alone is met.

Under certain conditions, the friction coefficient may be high, but the bearing temperature may not rise significantly. In these cases, the test will not be interrupted until the temperature rises above a critical temperature, which may by past experience cause damage to the rig and its components. It should be noted, that this is the temperature behind the bearing in the carrier, and that the temperatures in the actual bearing may be significantly higher. These tests will help determine which of the two, temperature or frictional torque, is a better indicator of wear or distress of the bearing.

For a test run at 70% of the maximum speed, the minimum time for a test is set at 1 minute. Just as the maximum duration of tests was set to time periods that produce the same number of cycles, the minimum test period will also be set to produce the same number of cycles, according to the speed of the test.

A few different loading conditions are used to obtain a statistical representation of the bearing behavior and life. These tests are within a range of speeds and loads, which are comparable to what the bearing carries in the transmission. The 16 different load and speed combinations vary widely over the capable loads and speeds of the test rig.

A new set of washers is used for each test and the carrier and gears are exchanged for new parts at regular intervals to prevent wear from affecting the tests. This interval has been determined by measuring the surface parameters (roughness) after each test.

The resulting real-time data that is collected during a test appears as the plot presented in Fig. 3. Here, the frictional torque is used to calculate the coefficient of friction and the temperature is normalized. This plot also shows how the friction and temperature are coupled and follow the same general trends. The trend in this plot itself is interesting as it shows how the bearing behaves

Table I—Dry Effective COF Values for Various Axial Loads	
Normalized Load	Coefficient of Friction
0.151	0.0885
0.434	0.0723
0.717	0.0644
1	0.0539

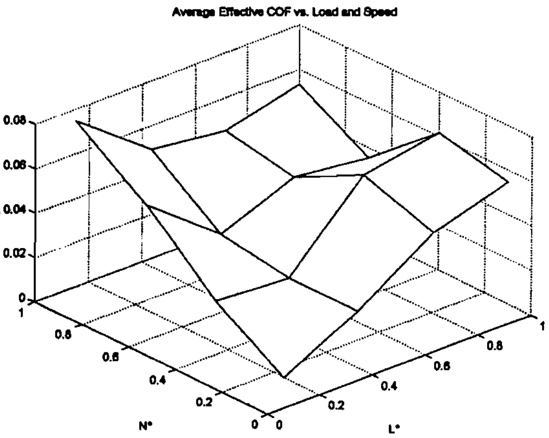

Fig. 4— The average effective coefficient of friction of the bearing *vs.* various loads and speeds.

over its life. After a finite number of cycles the effective coefficient of friction between the surfaces increases, suggesting that the coating is wearing away and losing effectiveness. Although not shown in the plot, eventually the coating will wear away and the bearing may start to scuff and reach a point of distress, at which the temperature will quickly rise above the cut-off criteria.

RESULTS AND DISCUSSION

Dry or unlubricated test runs on the PTFE coated bearings at various loads resulted in the coefficients of friction given in Table 1. These values were obtained by running the bearing without any lubrication and will be compared with the COF of the bearing run with lubrication. It is interesting to also note that the COF decreases in relation to load.

As reported earlier, other experimental studies such as the work by Rabinowicz, *(12)*, Simmons, et al. *(14)*, and Choudhary et al. *(5)* have also observed a decrease in the COF with the increasing normal load of contacting PTFE coated surfaces. In addition, there are various fundamentally derived models of friction that also suggest this same phenomena, such as the work by Chang, et al. *(4)*, Etsion and Amit *(6)*, Roy Chowdhury, et al. *(13)*, and Kogut and Etsion *(10)*. These works theorize that the plastic yielding of the contacting surface asperities can limit the COF between them.

The average effective coefficient of friction over time for each test is calculated and plotted with respect to the thrust load and rotational speed in Fig. 4. The COF results vary by an RMS value of approximately 23% between tests of the same speed and load conditions. These variances may occur because of slight differences which occur in the set up of each test and differences due to imperfect manufacturing tolerances in tested components. The effective COF is calculated using the following equation whose derivation is given in Jackson and Green *(8)*.

$$\mu = \frac{3}{2} \cdot \frac{T_c(r_o^2 - r_i^2)}{L(r_o^3 - r_i^3)} \qquad [1]$$

where

μ – effective coefficient of friction
L – axial load, [N]
T_c – frictional torque, [N•m]
r_i – inner diameter of washer, [m]
r_o – outer diameter of washer, [m]

It is evident from the plot that the COF varies greatly in relation to speed and load. At higher loads, the COF first increases with speed and then decreases, perhaps due to hydrodynamic or thermal effects. Otherwise, the COF increases with speed. At lower speeds, the COF increases with load, while at higher speeds the COF follows a trend that is similar to the Stribeck curve.

Due to the described mapping of the COF, there is a desired region of bearing operation where the COF is lowest. This region appears at a normalized load of approximately 0.4 and for a variety of speeds. In comparison to the dry tests, the COF also reaches values low enough to suggest that the lubrication further enhances the bearing performance at most loads and speeds. It is also evident from this comparison, that at high loads and low speeds the liquid lubricant has little effect on the COF.

The standard deviation of the COF during each testing condition is also calculated. This value represents the variation in the COF of the bearing during the duration of the test. For instance, if there is a large amount of stick-slip occurring during the test, this value will reflect that in comparison to other tests. From Fig. 5, it is shown that the standard deviation of the COF changes little with rotational speed and axial load except that at low loads there is a slight increase and at the highest load and speed the value increases drastically. At low loads there is less force to situate the bearing and then it is believed there may be some vibratory effects that will alter the effective axial load and the COF, which accounts for the increase in the standard deviation. At higher loads and speeds the increase in standard deviation is thought to be due to severe wear over a short period of time, which could lead to abrupt changes in surface properties and changes in the COF. The standard deviation of the COF also depicts a desired region of operation similar to the region previously suggested from the COF plot.

The average bearing temperature is calculated by averaging with each other and over time the output of the four thermocouples embedded in the carrier. The temperature in °C is normalized by the test cut-off temperaturein °C. This value for each test is plotted in Fig. 6. The bearing temperature results vary by an RMS value of approximately 8.3% between tests of the same conditions. Since temperature is used as a criterion for test cut-off, the average temperatures are not always indicative of the bearing steady-state values. Generally, the temperature seems to increase with axial load and rotational speed. There is also a ridge of high temperature that cuts diagonally from the high load, low speed

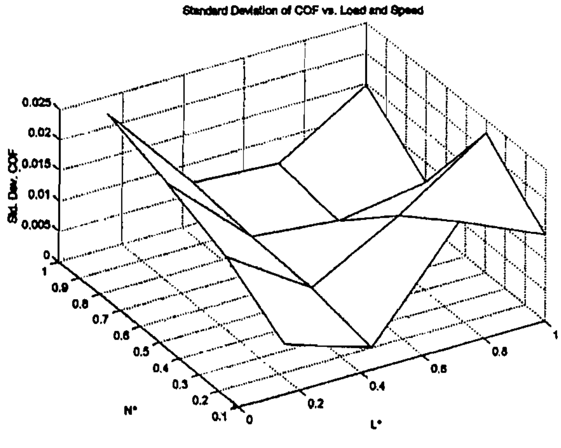

Fig. 5—The standard deviation of the effective coefficient of friction of the bearing *vs.* various loads and speeds.

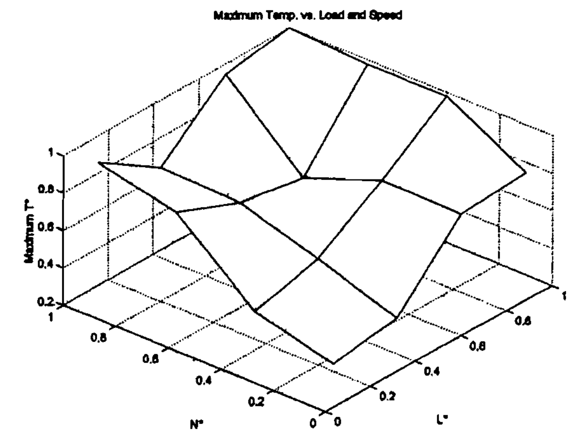

Fig. 7—The maximum temperature of the bearing *vs.* various loads and speeds.

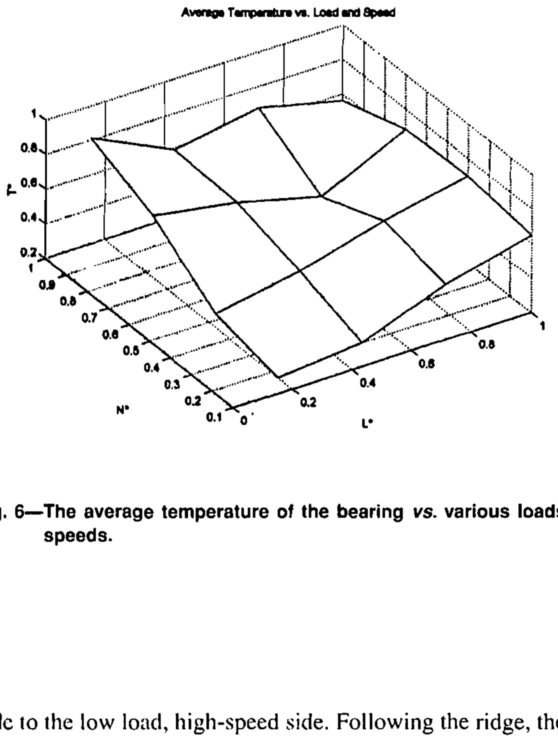

Fig. 6—The average temperature of the bearing *vs.* various loads and speeds.

side to the low load, high-speed side. Following the ridge, there is thus a region at which the bearing temperature actually decreases or stays near constant. This behavior might be caused by hydrodynamic pressures providing lift to alleviate the severity of contact between the surfaces. A similar trend is also seen in the COF plots (Figs. 4-5).

The maximum temperature the bearing reaches for each test is also found and is plotted in Fig. 7. This is not the local maximum temperature on the bearing surface, but rather the maximum temperature reached by the thermocouples. In reality, due to the thermal resistance of the carrier, and the adhesive between the thermocouple and the carrier, the temperature on the surface of the carrier can reach local temperatures higher than what the thermocouples read at any instance. On a comparative basis these temperatures can indicate differences in the efficiency between the bearings run at different loads and speeds. The maximum temperature plot maps trends that are similar to the average temperature, except that at higher loads the rise in temperature is more drastic. However, at the higher loads a maximum steady-state temperature is never reached since the test is stopped once the temperature reaches the cutoff criteria.

In Fig. 8, the calculated standard deviation of the temperature is plotted. Since the temperature generally increases from room temperature to the operating temperature at start-up, the standard deviation of the temperature at operating conditions that produce more heat is generally higher. General variations in the frictional heat generation and thus in the bearing temperature will also increase the standard deviation. A less profound diagonal ridge from high loads and low speeds to low loads and high speeds is also seen in this plot.

Generally, the various plots of bearing temperature also suggest preferred regions of bearing operation, although elevated temperatures may not always indicate an undesirable region of operation. Thus, an acceptable region of operation is dictated by a maximum temperature that is lower than the cutoff temperature. Based on this assumption, the region of desired operation seems to span all speeds at $L^*=0.43$ and span speeds up to 0.7 for loads between 0 and about 0.7.

The bearings are run until the test cut-off criteria (based on bearing temperature) or bearing failure criteria (based on friction) are met. Figure 9 plots the test duration in cycles, which is controlled by the temperature cut-off criterion. Figure 10 plots the bearing life, which is defined by the number of cycles until the coefficient of friction exceeds a value of 0.06. When examining these plots it should be noted that the grid is rotated 180 degrees from the previous plots so that the interesting behavior of the plot can be seen.

It is evident from these plots that the two failure criteria are closely related. The largest discrepancy comes at low loads and high speeds. For the plot of the actual test duration as decided by the critical bearing temperature the only region where the test is

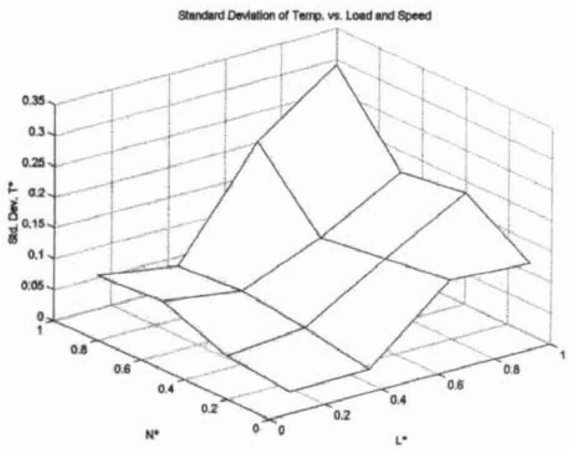

Fig. 8—The standard deviation of the bearing temperature *vs.* various loads and speeds.

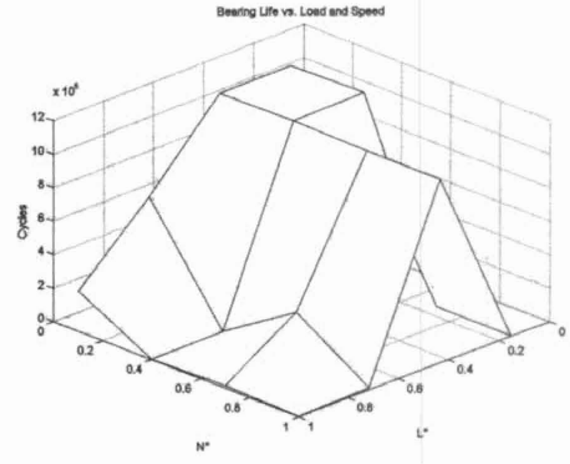

Fig. 10—Bearing life, which is defined by a maximum coefficient of friction value *vs.* various loads and speeds.

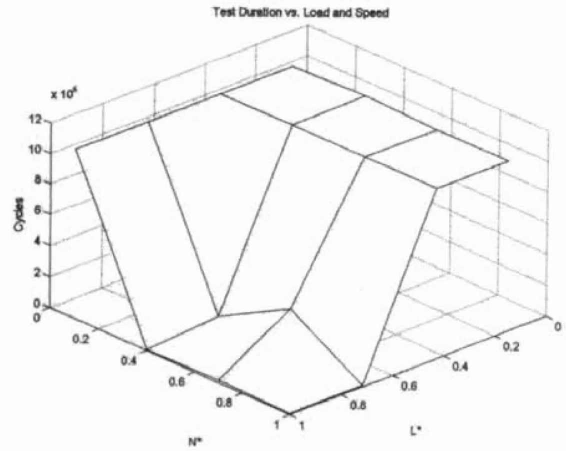

Fig. 9—Test duration, which is controlled by a temperature cut-off criterion *vs.* various loads and speeds.

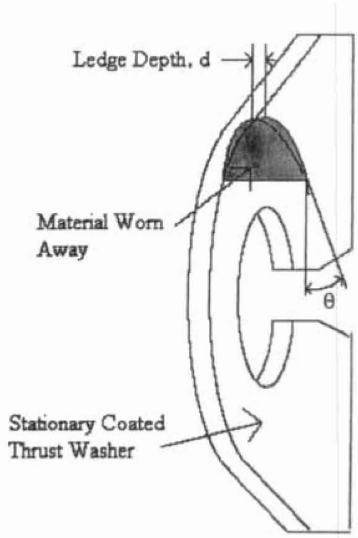

Fig. 11—Exaggerated view of washer wear and measurement.

stopped is at increasing loads and speeds, while the friction suggested bearing distress also for low loads at high speeds. The bearing assembly in the test rig is held together in the axial direction only by the axial load. Thus, one possibility for these higher COF at lower loads is because the bearing is less restrained under these conditions. From these plots the most desirable region of operation appears to be at low loads and low to medium speeds.

Wear Analysis

The amount of wear of the coated bearing surfaces is approximated by first using a profilometer to measure the depth of the deepest worn edge. Due to the non-axisymmetric loading, the wear is uneven as well, so that the volume of worn material is not simply the ledge depth, d, times the bearing contact area (see Fig. 11). By using the known geometry of the gear, and the maximum angle it can rotate, θ, the wear volume is estimated from the worn ledge depth, d. Based on the geometry before any deformation or wear occurs, the maximum angle the gear can rotate, θ, is calculated to be 0.118 degrees. In other words, θ is the angle between the unworn surface and the worn surface.

Next, the estimated volume worn from the coated stationary surface is plotted *vs.* load and speed (Fig. 12). Compared with the coating depth, it is deduced that the coating is never worn away completely, but under the most severe conditions sees severe wear and would be worn away with more cycles. It is interesting that the volume of material removed increased with load, except at the highest loads where the test duration was often much shorter. This behavior agrees, at least qualitatively, with Archard's wear law. Archard's wear law is given as:

$$\omega = K \cdot \frac{L}{H} \qquad [2]$$

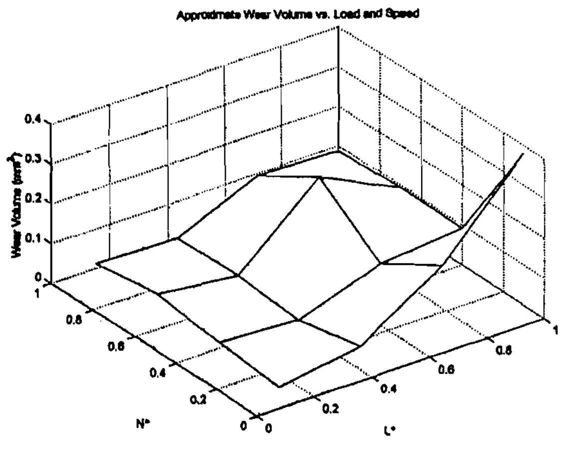

Fig. 12—Approximate wear volume *vs.* various loads and speeds.

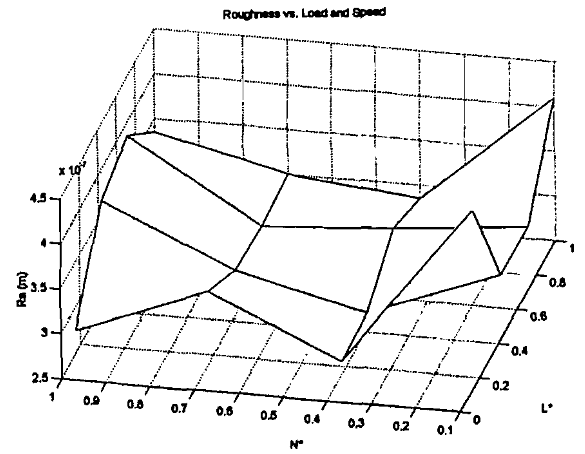

Fig. 14—Plot of the final average roughness value *vs.* various loads and speeds.

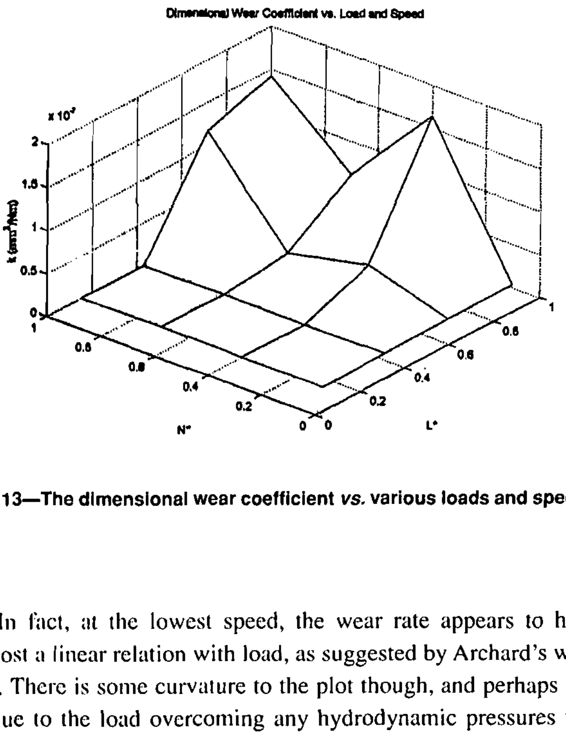

Fig. 13—The dimensional wear coefficient *vs.* various loads and speeds.

In fact, at the lowest speed, the wear rate appears to have almost a linear relation with load, as suggested by Archard's wear law. There is some curvature to the plot though, and perhaps this is due to the load overcoming any hydrodynamic pressures that may be present.

Little or no wear caused by these tests would indicate sufficient hydrodynamic lubrication. Thus, the length of bearing life during these tests could also indicate if bearing behavior is indeed hydrodynamic. It is apparent, that at low loads the wear is very small. This is also confirmed by visual inspection of the surfaces.

To account for the various number of tested cycles and loads, the wear volumes were used to calculate the dimensional wear coefficient (Williams, *(16)*) or wear rate, k, where

$$k = \frac{K}{H} \quad [3]$$

Plotting this value gives a representation of the severity of the wear of the coated bearing for different loads and speeds (Fig. 13).

Generally, for higher speeds the wear rate increases drastically for higher loads. For lower speeds the wear rate seems to stay nearly constant, which suggests that Archard's wear law may be valid for low speeds.

Scuffing Wear

Although not presented in this work, other thrust washer bearings have been tested without coatings (Jackson and Green, *(8)*). At times, these bearings, when under higher loads and speeds, have effectively welded together. This suggests the occurrence of adhesive wear mechanisms, which is the cause of scuffing (Williams *(16)*). At these loads and speeds a vapor mist is often seen exiting the test rig, which encloses the bearing. This suggests that the local surface temperatures on the bearing surfaces are very high. Severe failures or bearing distresses of this nature have not been encountered while testing the PTFE coated bearings under those loading conditions that caused welding of uncoated bearings and are within the load and speed combinations tested in this work. For this reason, it is believed that the PTFE coating is an effective means of increasing the bearing's resistance to scuffing and thus increasing its range of operation.

Surface Property and Wear Correlations

Due to the small amounts of material that are worn away during these tests, measuring wear by weighing requires precision instrumentation. Also, unlike the current coated stationary washer bearing, many of the washers being tested outside of this work have no extra surface area with which to compare with the worn surface. For these reasons another method of measuring wear is desired. Thus, various roughness parameters were taken before and after testing and then compared. Since the surface topography will change with wear, these parameters should also change with wear. Only the average roughness, R_a, is presented here. This is a common roughness parameter whose formulation can be found in many texts (e. g. Williams, *(16)*).

In Fig. 14, the resulting roughness values after testing are plotted. Even though there is no comparison here with values taken before testing, these plots still provide some interesting trends. It is believed that the operation of the bearing will help dictate the roughness of the final bearing surface. In general, the severity of wear, or larger wear rates, should lead to larger roughness values.

The largest increase in roughness occurs as the speed is increased and also to a lesser degree, as load is increased. This supports the theory of scuffing since at higher speeds, and thus usually higher bearing surface temperatures, the likelihood of micro-welds and adhesion occurring is greater. Also, at low speeds an interesting trend appears. The roughness starts from high roughness at low loads and low speeds and drops as load is increased but then increases again with higher loads. This may be due to the previously discussed lack of axial restraint at lower loads and the expected more severe wear at higher loads. It is also interesting to note that at high speeds and low loads there is a sudden drop in roughness, perhaps due to the removal of PTFE or a polishing effect. In fact, on average the roughness of the washers decreased from before to after testing. The trends for the average roughness, R_a, and the root mean squared roughness are generally the same.

CONCLUSIONS

The experimental data suggests a couple of points about the behavior of the PTFE coated thrust washer bearings. First, the coating has a finite life and will eventually wear off and thus lose effectiveness. This wear is greater for higher loads and also sees a slight increase for higher speeds. Second, the bearing has desirable regions of operation, based on what measure of bearing behavior is most important. Loads and speeds which are generally lower than the highest loads tested, but still slightly above the lowest appear to generally define a region of desirable operation. Depending on what bearing qualities are important to an application, the approximate desirable regions of operation can be found from these experimental plots.

The coating does appear to improve the bearing life and operational behavior, partially due to its ability to add resistance to bearing distress and scuffing. During testing, the coated washers never weld together or cause seizure like their uncoated counterparts have been known to do.

A correlation is made between bearing wear and changes in the roughness of the bearing surfaces. For the most part, it appears that scuffing or severe wear rates cause rougher surfaces to form.

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