Life Test Results for the Advanced Very High Resolution Radiometer Scanner

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Abstract

The following paper reports the results obtained during a 3.33-year life test on the TIROS Advanced Very High Resolution Radiometer/3 (AVHRR/3) Scanner. The bearing drag torque and lubricant loss over life will be compared to predicted values developed through modeling. The condition of the lubricant at the end of the test will be described and a theory presented to explain the results obtained. The differences (if any) in the predicted and measured values of drag torque and lubricant loss will be discussed and possible reasons for these examined. The life test was funded under NASA contract number NAS5-30384.

Introduction

The AVHRR/3 Scanner, designed and developed by ITT-A/CD of Fort Wayne, IN, is the meteorological imaging system on the TIROS polar orbiting satellites (Figure 1). The AVHRR/3 is a six-channel scanning radiometer providing three solar channels in the visible-near infrared region and three channels in the thermal infrared range. It scans the earth surface twice a day with 1.1-km resolution at a low earth orbit of 834 km (450 nautical miles) [1].
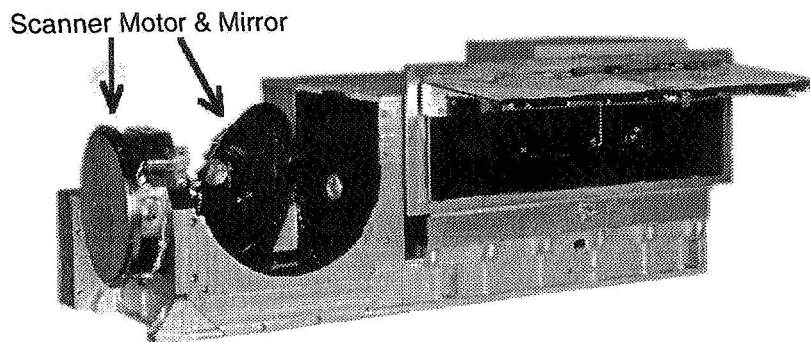

Scanner Motor & Mirror

Figure 1. AVHRR/3 Instrument

The earth scanning is achieved through the use of a continuously rotating scan mirror that produces scan lines normal to the spacecraft orbital track. The scan mirror rotates at 360 rpm, thus making the adjacent scan lines contiguous at the sub-satellite position. The scan mirror is driven by an 80-pole, two-phase hysteresis synchronous

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motor, which provides torque at a low level, thus making minimum drag torque a must. As a result of this particular motor, bearing torque and torque disturbances are a significant source of concern in the build up and operation of the instrument. Every effort is made to accurately predict the nominal torque and the torque jitter associated with its variation. In order to get a low running torque with almost no variation, a minimal amount of free lubricant must be used. The prediction of lubricant loss over the life of the bearing becomes an important analysis, since the loss of all the bearing lubricant can result in a catastrophic failure of the instrument. A life test unit, representative of a flight configuration, was built and tested. The lubricant loss and bearing evaluation after disassembly were the two main purposes of this test.

Discussion

Bearing System Description
The bearing system tested in the life test unit was identical in every way to the flight configuration. Angular-contact, back-to-back duplex pair bearings, lightly preloaded to 18-26 N (4-6 lb) with a contact angle of 23°, were installed onto the scanner shaft with a slight clearance fit. The tested bearing consisted of 440C races (double-honed), titanium-carbide-coated (TiC) 440C balls (24 per ball row), a porous polyimide retainer, fully impregnated with KRYTOX-143 AB oil (vacuum-baked in the oil for a minimum of 12 hours), and lubricated with 14 mg of KRYTOX 143-AB perfluorinated oil as the free lube. Figure 2 shows how the bearings are mounted in a scanner. The cover side bearing is mounted onto the shaft flush against the shoulder of the shaft. Two concentric cylindrical spacers are placed between the bearings with one acting on the set of inner rings and the other on the set of outer rings. The length of the spacers is honed to achieve the desired bearing preload. The mirror side bearing inner ring is secured by tightening a shaft nut to the desired torque (a spacer is used between the bearing and the nut). A retainer is bolted to the housing on both the mirror and cover sides to restrain the outer rings. All spacers, nuts, and retainers are coated with a barrier film to help prevent the migration of oil out of the bearing.
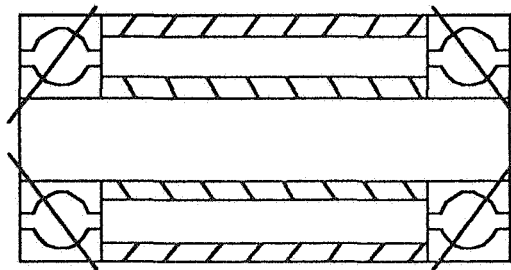

Figure 2. Bearing Configuration

Drag Torque Modeling

One of the two main concerns in this bearing system, as previously mentioned, is the drag torque. The hysteresis synchronous motor used to operate the life test scanner had a torque capability of 35.3 mN-m (5.0 in-oz) with a bearing torque of 7.1 mN-m (1.0 in-oz). This combination gives a small torque margin of 28.2 mN-m (3.0 in-oz) until end of life.

The model used to predict the drag torque was "BEARINGS9", developed by Al Leveille of the Aerospace Corporation [2]. The program analyzes the effects of the bearing configuration and external loads (including thermal loads) on the behavior of the ball bearing. The bearing, housing and shaft geometries, along with temperatures, surface finishes, lubricant and material properties are entered into the program. A lubricant quantity factor, which differs for each lubricant and the amount of lubricant used, needs to be determined prior to use of the program.

Several flight bearings (same type as life test) were tracked, as the actual drag torque during build up and amount of free lube used (same amount of KRYTOX-143AB oil as the life test) during processing were recorded. Several runs of the program were made by entering the bearing geometries and assuming a lubricant quantity factor (the lubricant quantity factor is a constant which helps define the operating condition based on the amount of lubricant used, i.e., for flooded systems the value could be ~3-4, for minimal systems it could be ~0.01).

The predicted drag torques were compared to the measured drag torques, and the lubricant quantity factor was adjusted for each bearing until an exact match was made. The lubricant quantity factors for each of the bearings examined were then averaged to arrive at a single value. Using this value, the program was re-run for each bearing and the predicted vs. measured values of drag torque are shown in Figure 3. The dashed line represents a perfect match based on the quantity factor of 0.343 (average of all other factors). The solid line is the best fit curve through the predicted values. The equation of the best fit line is

$$Actual\ Torque = 2*Predicted - 1.1$$

Using the lube quantity factor of 0.343 in the "BEARINGS9" model gives a predicted drag torque for the life test unit of 7.6 mN-m (1.07 in-oz). If the best fit curve were the 45° line, this would be our result. Since the data shows a different relationship, we must input this value into the equation of the solid line. Inputting this predicted value of 7.6 mN-m (1.07 in-oz) into the equation for the best fit curve yields a torque value of 7.3 mN-m (1.04 in-oz). The measured torque was 7.1 mN-m (1.00 in-oz), a difference of 4.0%.
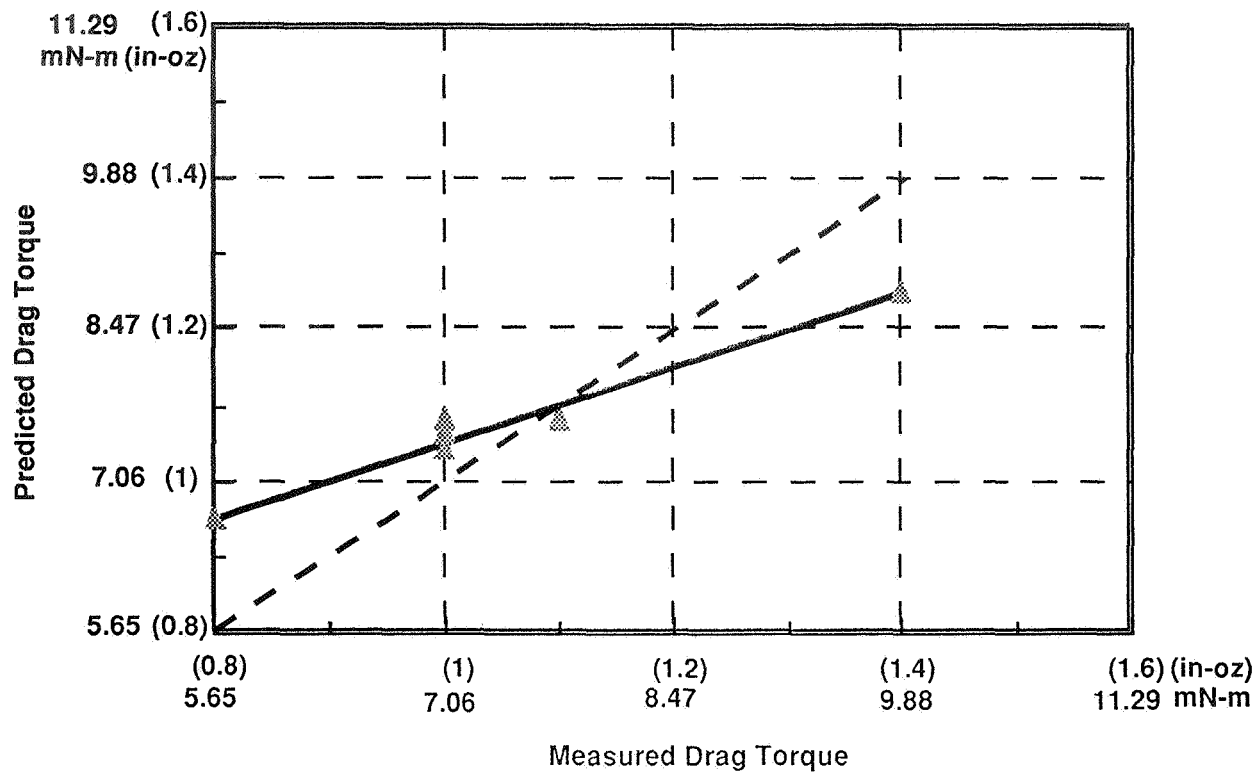
11.29 (1.6)
mN-m (in-oz)

9.88 (1.4)

Predicted Drag Torque

8.47 (1.2)

7.06 (1)

5.65 (0.8)

(0.8) (1) (1.2) (1.4) (1.6) (in-oz)
5.65 7.06 8.47 9.88 11.29 mN-m

Measured Drag Torque

Figure 3. Drag Torque Modeling Results

Lube Loss Modeling

In an effort to reduce the drag torque, a minimal amount of free lubricant needs to be used in the bearings. The more accurate the prediction of the lube loss, the less lubricant will be required by the bearings to make up for any uncertainty in the prediction. The lube loss for the life test configuration was modeled using the theory established in Scientific Foundation of Vacuum Techniques by S. Dushman [3].

Background
The following is a brief overview of the vacuum technique theory employed in this analysis.

The amount of lubricant lost in a bearing system is dependent on three major influences. The first influence is the physical characteristics of the lubricant used. The molecular weight, density and vapor pressure play major roles in determining what quantity of lubricant will vaporize and at what speed it will travel after vaporization. The second major influence is the environment in which the system operates. Temperature and pressure directly influence the speed of lubricant vapor molecules (in the same manner as the molecular weight) and the quantity in which it will vaporize (similar to the lubricant vapor pressure). The final influence that controls the amount of lubricant lost from the bearing system is the mechanism labyrinth system.

The labyrinths need to be designed small enough to restrict the flow of the lubricant vapor and still allow for free rotation of the parts. In the AVHRR design, there are two

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types of labyrinths: an axial labyrinth (volume between two concentric cylinders, Figure 4) and a radial labyrinth (volume between two flat discs with flow from the center outward, Figure 5). The ratio of the length of a labyrinth to its gap has a direct relationship to how well the labyrinth will work. For instance, if the length to gap ratio is very large (>100), the labyrinth will retard most of the vapor flow through it. Conversely, if the ratio is very small (<0.1), most of the vapor will pass without interference.
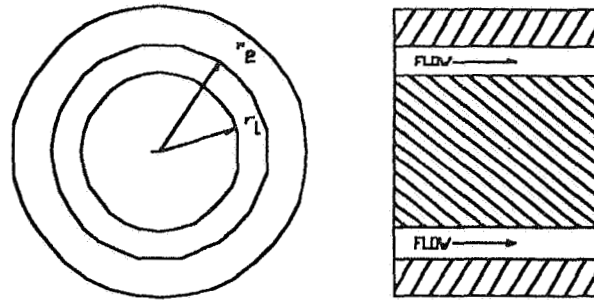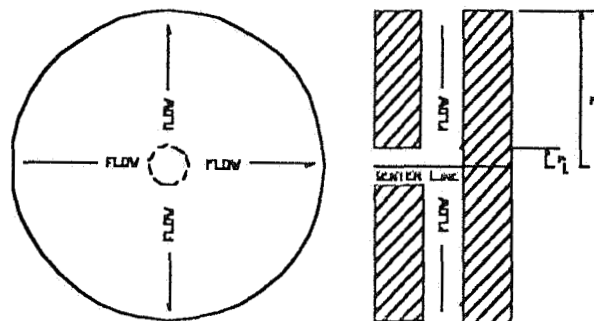
Figure 4. Axial Labyrinth

Figure 5. Radial Labyrinth

Derivation
A flow rate, Q, is used to define the amount of lubricant vapor per unit time that escapes from the labyrinth seal of the system. It is defined as:

$$Q = F(P_2 - P_1)$$ where: P_2 = Pressure at channel entrance
P_1 = Pressure at channel exit
F = Conductance

The quantity F (conductance) corresponds to the rate of flow per unit difference in pressure between the upstream and downstream ends of the channel. Each gap between the rotating and stationary parts in the system from the bearing through to the outside of the mechanism will be evaluated and a conductance calculated for it. All the conductances for a given bearing will then be added together to arrive at a total conductance for that bearing. The lubricant loss rate can then be predicted based on the conductance and the vapor density.

The conductance for an axial labyrinth (volume between concentric cylinders) is found by following the steps outlined in Dushman. First, the arithmetic mean velocity, V_a, is found. This is the average speed at which a molecule of a certain vapor will travel.

$$V_a = \left(\frac{8 \cdot R_o \cdot T}{\pi \cdot M}\right)^{\frac{1}{2}} \qquad V_a = 1.4551 \cdot 10^4 \left(\frac{T}{M}\right)^{\frac{1}{2}}$$

Where: R_o = universal gas constant (8.3146×10^7 cm^2/K$\cdot$S^2)
  T = lubricant temperature in Kelvin
  M = lubricant molecular weight

The conductance is defined as follows:

$$F = \frac{\frac{4}{3} V_a}{\int_0^L \frac{H_A}{(A_A)^2} dL}$$

Where: V_a = arithmetic mean velocity (cm/s)
  H_A = sum of inner and outer labyrinth
      perimeters (cm)
    $= 2\pi r_1 + 2\pi r_2 = 2\pi(r_1 + r_2)$
      Where: r_1 and r_2 are the inner and outer radii
  A_A = Axial labyrinth cross-sectional area (cm^2)
    $= \pi r_2^2 - \pi r_1^2 = \pi(r_2^2 - r_1^2)$

Which gives:

$$F = \frac{\frac{2}{3} \cdot V_a \cdot \pi \cdot \left[(r_2)^2 - (r_1)^2\right] \cdot (r_2 - r_1)}{L}$$

for an infinitely long labyrinth. Since our case is not infinitely long, a correction factor is required.

Clausing deduced a relationship between actual and theoretical conductances.

$$\frac{F_{Theo}}{F_{Act}} = \frac{8 \cdot a}{3 \cdot K \cdot L}$$

Where: F_{Theo} = theoretical conductance (cm3/s)
F_{Act} = actual measured conductance (cm3/s)
a = annular opening, gap (r_2-r_1) (cm)
L = length of labyrinth (cm)
K = Clausing's factor (based on L/a)

Substituting the above relationship into the derived conductance (for infinite length) gives an actual conductance based on the length to gap ratio.

$$F_{Act} = \frac{\pi}{4} \cdot v_a \cdot \left[\left(r_2 \right)^2 - \left(r_1 \right)^2 \right] \cdot K$$

Following the same process, a conductance for a radially expanding or contracting labyrinth can be found. The new Perimeter (H_R) and Area (A_R) values are:

$$H_R = 2\pi r \quad \text{and} \quad A_R = 2\pi r L$$
$$\text{as } r \text{ goes from } r_1 \text{ to } r_2$$

Since there is a change in the cross-sectional area and perimeter with respect to the radius and not to L, a new integral must be evaluated.

$$F = \frac{\frac{4}{3} \cdot v_a}{\int_{r_1}^{r_2} \frac{4 \cdot \pi \cdot \rho}{\left(2 \cdot \pi \cdot \rho \cdot L \right)^2} dr} = \frac{\frac{4}{3} \cdot v_a \cdot \pi \cdot L^2}{\int_{r_1}^{r_2} \frac{1}{r} dr} = \frac{\frac{4}{3} \cdot v_a \cdot \pi \cdot L^2}{\ln \frac{r_2}{r_1}}$$

The correction factor derived previously also requires modification for the radial change. If we assume that the correction factor will work for a radially changing labyrinth as it does for an axial labyrinth, changes to the gap and length definitions need to be made. The distance, a, is now equal to L and not r_2-r_1. The length of the labyrinth also changes from L to r_2-r_1.

This changes the Clausing relationship to:

$$F_{Act} = \frac{F_{Theo} \cdot 3 \cdot K \cdot \left(r_2 - r_1 \right)}{8 \cdot L}$$

Substituting back into the derived equation gives:

$$F_{Act} = \frac{\pi}{2} \cdot v_a \cdot \left(r_2 - r_1 \right) \cdot L \cdot K \cdot \left(\ln \frac{r_2}{r_1} \right)$$

Now that the bearing geometric considerations are determined, the properties of the lubricant need to be examined.

The temperature of the lubricant vapor, the lubricant molecular weight, and the vapor pressure of the lubricant at that temperature define the density (ρ) of the vapor:

$$\rho = 1.6035 \times 10^{-5} (M \cdot P)/T \quad g/cm^3$$

Where: M = molecular weight of lubricant
T = temperature of lubricant in Kelvin
P = pressure differential in Torr

The mass loss rate (m) is then defined as the density multiplied by the conductance as shown below:

$$m = F \rho \quad g/s$$

Knowing the initial amount of lubricant in the bearings, a time to complete depletion (t) is calculated:

$$t = (\text{initial lube mass})/m$$

or the amount of lubricant lost over a period of time can be calculated:

$$\text{mass loss} = (\text{time in operation})(m)$$

Empirical Lube Loss Data
The AVHRR life test concluded after 3.3 years of operation in a vacuum environment with no signs of performance degradation (drag torque was ~7.7 mN-m (1.1 in-oz)). The bearings were removed, weighed, and examined. The weights were compared to the original weights, and the lubricant loss was found to be 8 mg for the cover side bearing and 9 mg for the mirror side. The measured results are much different from the predicted values of 3.9 and 18.8 mg for the cover and mirror rows, respectively, using the previously derived equations. An examination into the differences was performed.

Data Comparison
The analytical prediction of oil loss for the life test unit considered each bearing as a separate entity and not as a single system. If the bearings were considered to be a single system, the total measured loss would be 17 mg over the 3.3-year life, compared to the predicted value of 23 mg. This predicted value is closer than the original prediction, but still not close enough to be useful.

The reasons for the difference between the predicted and measured lube loss values is most likely a function of one or more of the following:

    1. Vapor pressure of KRYTOX is not correct
    2. Sticking factor plays larger role than thought
    3. Bearing temperatures are not known
    4. Barrier film has a vapor pressure that adds to the system
    5. Conductances are not modeled correctly

The vapor pressure value published by the vendor of KRYTOX-143AB ($2 \cdot 10^{-10}$ kPa or $1.5 \cdot 10^{-9}$ Torr) [4] was questioned in an earlier analysis performed at ITT and was shown to be different in tests performed at NASA ($3.3 \cdot 10^{-11}$ kPa or $2.5 \cdot 10^{-10}$ Torr) [5]. The NASA value was the one used in the analysis, so it should be considered correct. The parts were inspected after disassembly, and no lubricant film was observed on any of the labyrinth parts, thus ruling out that the lube was adhering to the walls of the unit and increasing the local vapor pressure enough to inhibit new vapor from forming. The temperature of the bearings (not measured directly but through thermocouples on the cover and housing) should not increase enough from the thermal model to warrant a doubling of the evaporation. The barrier film is also an area that could help the lubricant remain in the bearings by raising the local pressure high enough to decrease the evaporation rate of the lubricant. The barrier film vendor was contacted about the vapor pressure of the film used and they stated that it was negligible. The conductances were modeled from the works of Dushman and Clausing for straight labyrinths. Our configuration has a straight axial labyrinth making a 90° turn into a radially contracting labyrinth. This is an area that requires another correction factor to account for the bend. The flow from the mirror side labyrinth is a good example of this configuration. Figure 6 shows the configuration and relative sizes of the labyrinths in question.
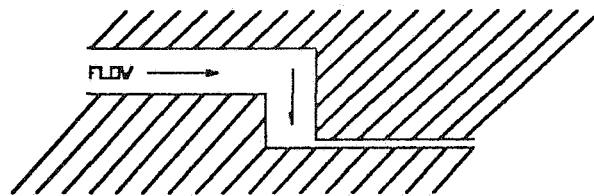

Figure 6.   Relative Size of Labyrinths

By taking 75% of the flow through the axial labyrinth and assuming that it makes the turn into the radial labyrinth at any given time, the mirror side conductance prediction gives a lube loss of 14 mg, compared to the 9-mg actual loss. The cover side would not be affected by this factor and has a predicted loss of 4 mg, compared to an actual loss of 8 mg. The sum of these predicted losses is 18 mg, which is close to the measured loss of 17 mg. This observation lends credence to the belief that the two bearings act as a single source. It can be theorized that the two concentric spacers

used between the bearings allow an unobstructed path for the oil vapor to travel from one bearing to the other. The straight path of the oil vapor allows the bearings to act as reservoirs for each other. When one bearing loses too much oil, a pressure drop occurs and pulls vapor from the other bearing of the pair to replenish the bearing that lost the oil (in this situation, it would flow from the cover side bearing to the mirror side).

Blackened Lubricant

An inspection of the AVHRR motor bearings under a microscope (30X magnification) was performed at the end of the life test. The bearing row near the cover end (away from the opening to vacuum) had a continuous band of blackened debris around the inner race and large clumps (~2 mm diameter) on the outer race and retainer. The debris was later identified as polymerized KRYTOX oil. When the polymerized KRYTOX was removed from the races, there was no sign of wear on either of the races or the balls.

The bearing row on the mirror side (near the labyrinth to vacuum) did not show blackened debris but had a thin film that resembled KRYTOX grease. The cloudy, thickened oil could possibly be the initial stage in the degradation of the oil. The inner race had a large wear mark all the way around , and the outer race had a smaller, variable wear mark. The difference in lubricant degradation between the two ball rows could be attributed to either the difference in radial loads as a result of the 17.8-N (4-lb) weight (used to simulate the mirror) or the easier flow path that the oil vapor has from the mirror side bearing to vacuum. The loads caused by the weight were not much different (19.6 and 37.4 N (4.4 and 8.4 lb) for the cover and mirror side rows, respectively), and a 17.8-N (4-lb) difference in a bearing rated for 543 N (122 lb) (3.3%) should not cause degradation to occur in one bearing and not the other.

The other difference, the conductance out to vacuum, has more merit as a possible explanation. The conductance out of the mirror side bearings is calculated to be 25.6 cm^3/s, while the cover row is 7.5 cm^3/s. This means that more oil vapor will travel out the mirror side row than the cover side row. If this is the case, and the bearings remain in a state of equilibrium, the loss from both sides must be the same. Meaning, that vapor from the cover side replenishes the vapor from the mirror side after it leaves the system. Since the loss rate is greater on the mirror side, the oil does not remain in the bearing as long as it would on the cover side, thus not accumulating the number of cycles required to become degraded. The cover side oil, however, does not easily leave the system (except to replenish the mirror side) and has much more time to become degraded.

Conclusions

The life test was terminated after 3.33 years of operation and showed no signs of performance degradation or impending failure. The jitter performance at end of life was identical to the beginning value (100% within 16 microseconds). Two measurements that were different from the initial testing were the coast down time (increasing from 32 sec to 76 sec) and the break-away torque (increasing from 2.65 mN-m to 7.94 mN-m (0.375 in-oz to 1.125 in-oz)). The 7.94-mN-m (1.125 in-oz) break-

away torque exceeds the test limit of 3.53 mN-m (0.50 in-oz) and can be attributed to the debris found in the bearing. The increase in coast down time is related to the decrease in free lube in the bearing.

The drag torque model used shows good correlation to empirical data once a satisfactory number of runs are accumulated to get the lubricant quantity factor required for the input. For an initial estimate on the drag torque, where precision is not required, the program is also very good. As with other tools, the more the program is used and the more familiar one becomes with it, the more useful it becomes.

The lubricant loss analysis correlates very well with the measured values once the correction factor for the 90° bend is used. The 75% factor should only be used for this labyrinth combination. Further testing and analyses would be required to get a relationship between labyrinth sizes (at a 90° bend) vs. conductance. A search of published results is being conducted to obtain further information. Based on the results obtained, a time to complete oil depletion will be 6 years, much longer than the 3.33 year requirement.

The blackened lubricant has been shown in many papers to be degraded KRYTOX. The interesting part is that only one of the two bearing rows showed signs of degraded lubricant. The conductance of the mirror side bearing row is much greater than the cover side row, yet they both lost about the same amount of lubricant. This shows that they act as a single source and that one bearing row will act as a reservoir for the other. The difference in the appearance of the lubricant from one row to the other also makes sense, knowing that the lubricant is more prone to leave the mirror side bearing and does not remain there long enough to accumulate the cycles required to degrade.

References

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