

# Wear Potential due to Low EHD Films during Elevated Temperatures

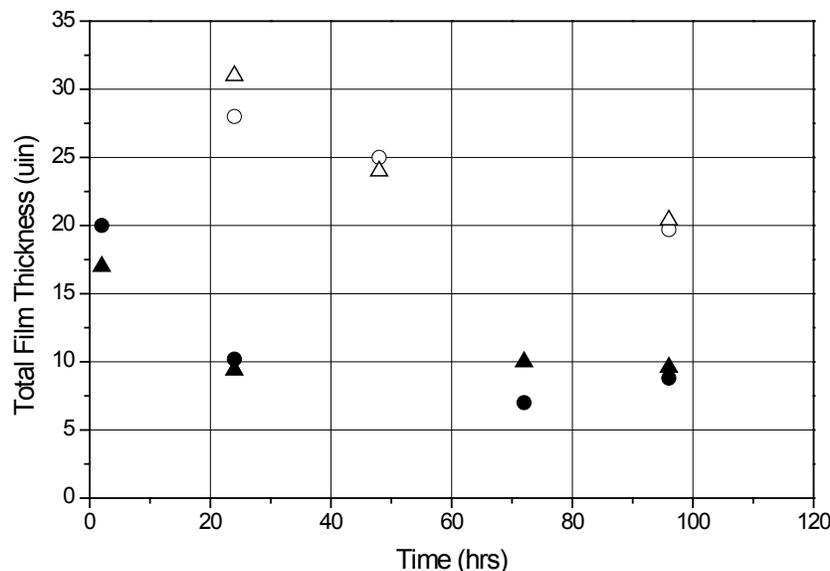
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## Abstract

An earlier study showed that EHD films could be accurately measured in a running bearing and that the EHD film eventually runs-in to a steady state value [1]. In the present paper, we report on additional tests conducted on bearings with more lubricants, wider speeds, and higher temperatures. The new results consistently show that all lubricants tested, including MAC-based lubricants have EHD film levels that are lower than model predictions in some situations. In addition, the MAC lubricants studied have lower film thickness than traditional hydrocarbons. Figure 1 is taken from [1] and shows room temperature data of MAC oil and Corey 100 oil, illustrating the smaller EHD film results when using this MAC oil.

Since higher temperatures produce lower films by changing the viscosity, the concern we have is that the EHD films may be too small to prevent ball/race metal contact and resulting wear at lower speeds. Best bearing practices would have the EHD film thickness be at least three (3) times the composite surface roughness. In this paper, we will present measured EHD thicknesses of lubricant films at speeds up to several thousand RPM for bearing bore sizes from as low as 6 mm (0.2 in) to as large as 35 mm (1.4 in) using MAC, Corey and KG-80. Ambient temperatures from room temperature to 52°C (125°F) are used. Testing was done with the base oils as well as formulated greases. Greases eventually ran in to the same EHD values as the base oil but took longer times to get there.

The results clearly indicate that wear is very possible in all steel bearings when using MAC lubricants and that this condition worsens with higher temperatures and smaller bearing size.



**Figure 1. Total Film thickness (in microinches) as a function of cumulative time operating at 6000 rpm with neat base oils formulated with TCP. Open symbols represent Corey and filled symbols represent MAC. Triangles and circles are the results from two independent measurement techniques**

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## Introduction

Coincident with the use of MAC-based lubricants in the 90's was the observation of wear magnitudes in all-steel components that struck most bearing engineers as larger than previously seen and had expected. In 2008, Ward, Frantz, and Leveille presented the first publication [1] of a new instrument that could measure EHD films in real bearings over hundreds of hours. These initial but limited tests showed that MAC-based lubricants produced EHD films about 50% smaller than previous hydrocarbon-based lubricants such as KG-80 and Corey 100. Additional testing done over the past few years has confirmed that initial observation. We also observed that all of the EHD film thicknesses were thinner than predicted by calculations using the method developed by Hamrock-Dawson [2] with the Coy-Zaretsky [3] starvation factors that we have been using in our bearing modeling tools.

The smaller EHD films raised the concern of how thin do the lubricant films become at elevated temperatures? Will the EHD films be too thin to prevent wear due to ball/race metal contact? We will present measured EHD films at speeds up to several thousand RPM for bearing bore sizes from as low as 6 mm (0.2 in) to as large as 35 mm (1.4 in) using MAC, Corey and KG-80. All the test data was at ambient of 24°C (75°F) except for the 6-mm (0.2-in) bearing size for which data was obtained up to 52°C (125°F). We picked the 6-mm (0.2-in) size for the higher temperature testing because a smaller bearing would inherently have a smaller film at any given speed. The results are that we see the potential for EHD films that may be bordering on marginal or a Lambda of 1 and therefore, may be a cause of wear that could affect longevity in the presence of wear acceleration factors.

### Results from Earlier Testing at Room Temperature

A review of data from our EHD test apparatus for actual bearings over the years of use for various problems was done. It showed a general trend that MAC lubricants were about 50% or lower in EHD film thickness than the old hydrocarbon lubricants such as KG80 and Corey 100. Table 1 summarizes some of this data over the years. Table 1 is a summary at 3000 or 6000 RPM at room temperature.

**Table 1. Summary of Results at Room Temperature**

LUBRICANT	BEARING SIZE (mm) [in]	RPM	TOTAL MEASURED FILM THICKNESS (µm) [µin]	MULTIPLIER TO MATCH H-D + C-Z
Corey 100	12 [0.47]	3,000	0.66 [26]	0.46
Corey 100	20 [0.79]	6,000	0.51 [20]	0.13
Corey 100	25 [0.98]	6,000	0.71 [28]	0.25
KG80	35 [1.4]	6,600	0.46 [18]	0.43
MAC oil	6 [0.24]	3,000	0.2 [7]	0.375
MAC oil	12 [0.47]	3,000	0.38 [15]	0.52
MAC oil	20 [0.79]	6,000	0.2 [9]	0.11
MAC oil	25 [0.98]	6,000	0.38 [15]	0.23
MAC grease	25 [0.98]	6,000	0.25 [10]	0.14
MAC grease	25 [0.98]	6,000	0.30 [12]	0.2

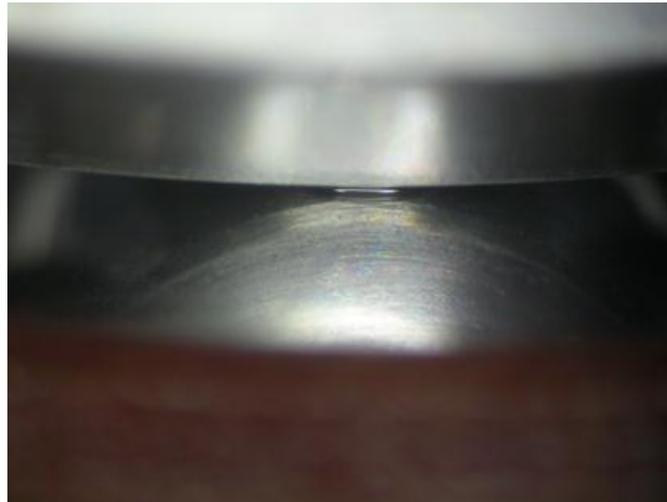
NOTE: The EHD values listed in Table 1 is the sum of the EHD film at the inner race plus the EHD film at the outer race. H-D+C-Z is the DYBA model calculation of film thickness using Hamrock-Dowson with Coy-Zaretsky reduction factors.

The data in Table 1 seem to illustrate around 50% less EHD measured values than modeling predictions for the older oils. Additionally, the EHD film thicknesses for MAC oil are lower than the older oils.

### Testing up to 52°C (125°F)

Further work was executed recently on a 6-mm (0.2-in) bore size bearing while using increased capability of the EHD test apparatus. Specifically, the test ambient temperature was varied using a heat lamp. The bearings and the entire test housing are allowed to reach a steady state ambient before starting the EHD test. Thermocouples measure the test bearing outer ring temperature rise above ambient to obtain accurate “actual average lubricant temperature” for modeling purposes.

EHD data was taken for various speeds from 3000 rpm to less than 1000 rpm to show the speed effect on EHD thickness. The actual EHD data was gathered, first on a bearing pair using just a small amount of MAC oil, which is Case 1. Each bearing had just a small meniscus of oil at the ball/race interface when viewed at rest (see Figure 2). Weight checks showed 30 mg of lubricant. The phenolic cage was properly impregnated with the oil prior to test. The intent was to mimic adequate but minimal amount of lubricant oil. The EHD measurement is taken within a few seconds of a spin-motor stop, allowing the endplay changes associated with EHD film collapse at 0 rpm to be captured isothermally.



**Figure 2. An impregnated phenolic cage with just an adequate amount of oil to show a small meniscus at start.**

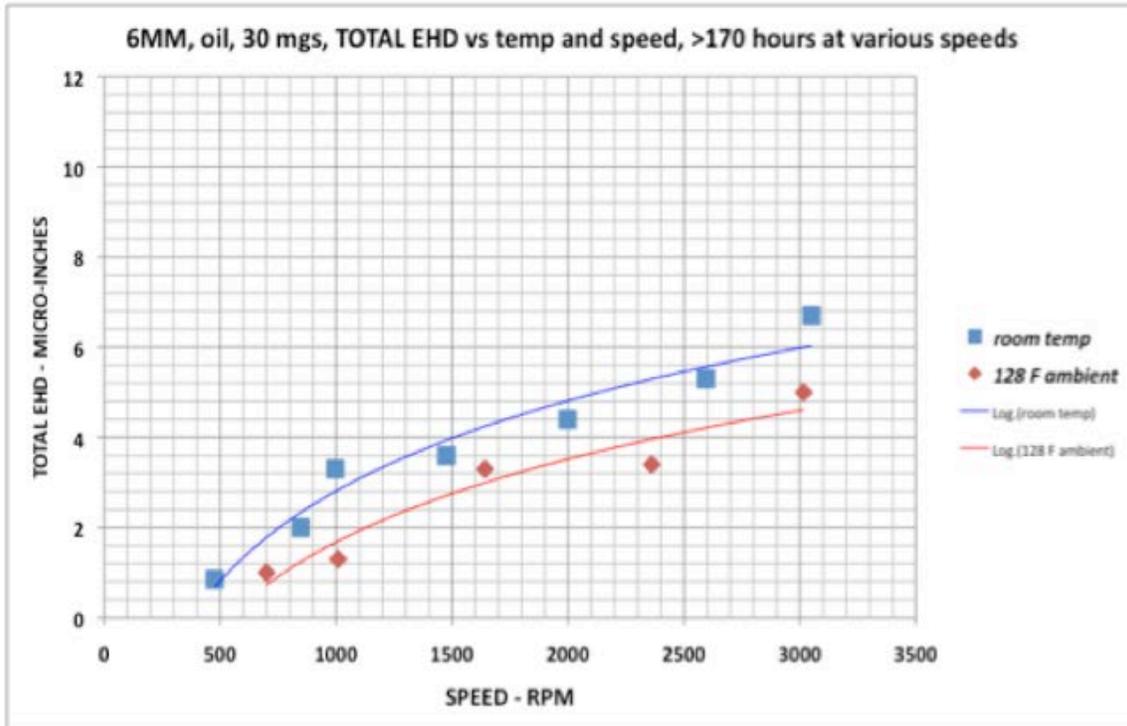
Next, another bearing pair was assembled with the same impregnated phenolic cage. Then typical cleaning sloshes with an oil/heptane mixture were done to assure bearing cleanliness followed by the addition of 25 milligrams of MAC grease. This Case 2 test shows a different lubricant procedure that represents methods known to be commonly used in industry, i.e., the use of grease lubricants.

The two lubricant cases were then subjected to a moderate amount of time at 52°C (125°F) running temperature while at 1000 rpm, approximating a beginning “run-in” process.

EHD data was then taken at various speeds from 3000 to less than 1000 rpm. Two running temperatures are shown in Figures 3 and 4, room temperature and 52°C (125°F). The EHD data is shown as a function of temperature and speed for the two lubricant cases. The overall intent is to show general operating lubricant amounts, coupled with running effects theoretically known to thin an operating lubricant film. We then correlate actual changes in EHD film thickness due to running conditions, with the modeling of EHD film thickness using the 50% reduction factor, over and above the standard Coy/Zaretsky reduction factor.

## Results and Discussion

The data presented in Figure 3 is on the 6-mm (0.2-in), all-steel bearing pair. It is the first lubricant case with a barely adequate amount of oil. As noted before, this data is collected after a brief “run-in” to distribute the lubricant and establish a steady-state condition. Then, the hot ambient tests are executed at different speeds followed by room temperature ambient tests after achieving the new steady state. This approach is used in an attempt to keep the lubricant condition in the bearing as identical as possible.



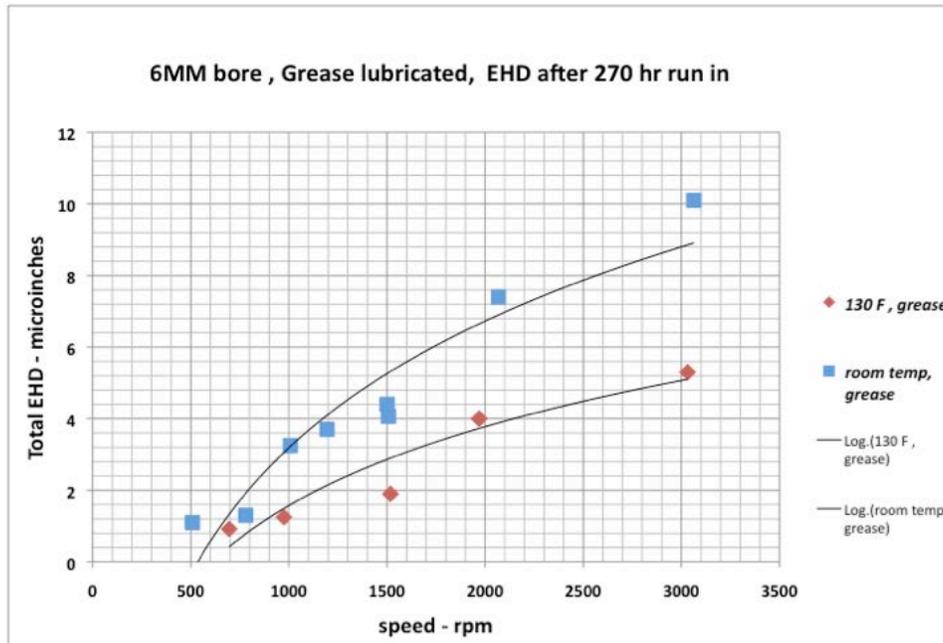
**Figure 3. Room Temperature and 52°C (125°F) EHD total values for the 6-mm (0.2-in) bore bearing lubricated with oil (Case 1).**

The data clearly shows the reduction of EHD film thickness due to the actual bearing running temperature and the accompanying viscosity change.

Next, Figure 4 shows data from the second lubricant case of tests with oil and grease. Again, the bearing is “Run-In” first at 1000 rpm in an elevated temperature condition, then data is taken.

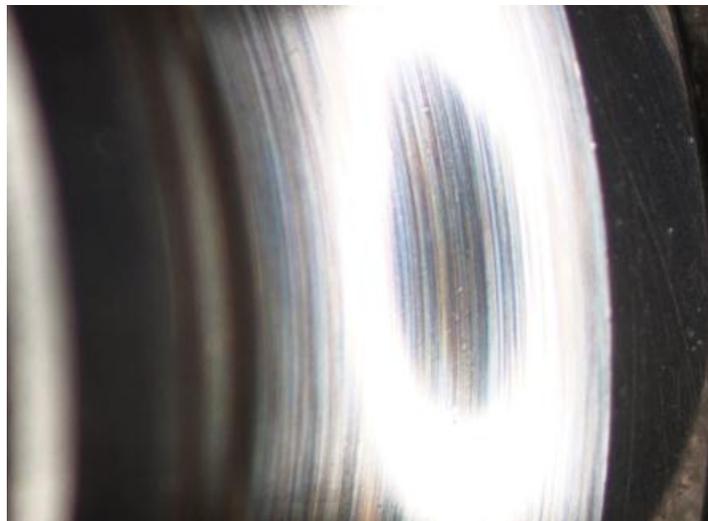
As before, the chamber temperature was elevated to 52°C (125°F) steady state before starting bearing rotation. The bearing temperature during rotation was approximately 54°C (130°F). Note at the high temperature and 1000 RPM, the total EHD film thickness has decreased to less than 1 micro-inch (0.03 micro-meters). This bearing is definitely starved.

The difference between this lubricant condition with oil and grease versus the previous test with only oil is rather minimal and close to the EHD measurement error. The results indicate that the extra grease did not provide better protection in the first few hundred hours of operation. However, the grease/oil mixture may act differently from pure oil as operational time continues.



**Figure 4. The “oil plus grease” bearing (Case 2) pair at room temperature and 52°C (125°F).**

In the next few figures, the bearing race surfaces from these tests are depicted. It is evident some non-EHD conditions were established as there is evidence of beginning wear present.

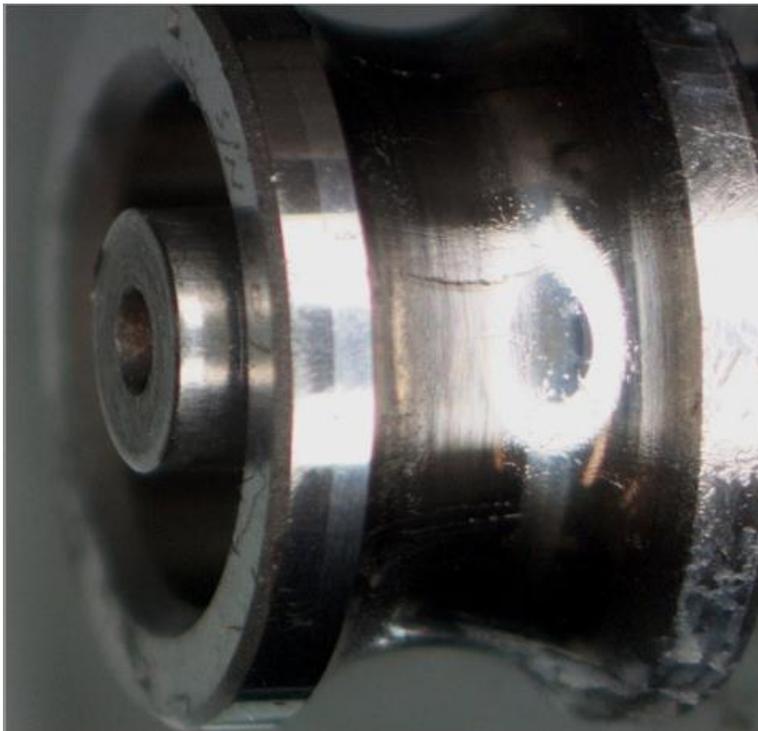


**Figure 5. 6-mm (0.2-in) inner ring race of an original test with a hybrid bearing of silicon nitride ceramic balls and steel rings after the “Run-In”.**

**Note there is no evidence of race wear with this dissimilar ball material on the steel ring.**



**Figure 6. 6-mm (0.2-in) race that generated the data shown in Figure 3, oil - only test. Note the worn ball path where hone lines have been eliminated when all-steel balls and rings/races are used.**



**Figure 7. Inner ring that generated the data in “Oil and Grease Test” shown in Figure 4. Again, there is wear in the ball path of the raceways that were tested against steel balls.**



**Figure 8. A higher magnification view of the race in Figure 7, clearly showing the worn off hone lines and darkened grease in upper portion of the photomicrograph.**

#### **Modelling Estimates of Inner Race (IR) EHD Values over a Range of Speed and Temperatures**

The measured EHD data was the total EHD at the inner ring plus at the outer ring.

That is: 
$$EHD_{ir} + EHD_{or} = EHD_{total} \quad (\text{Eq. 1})$$

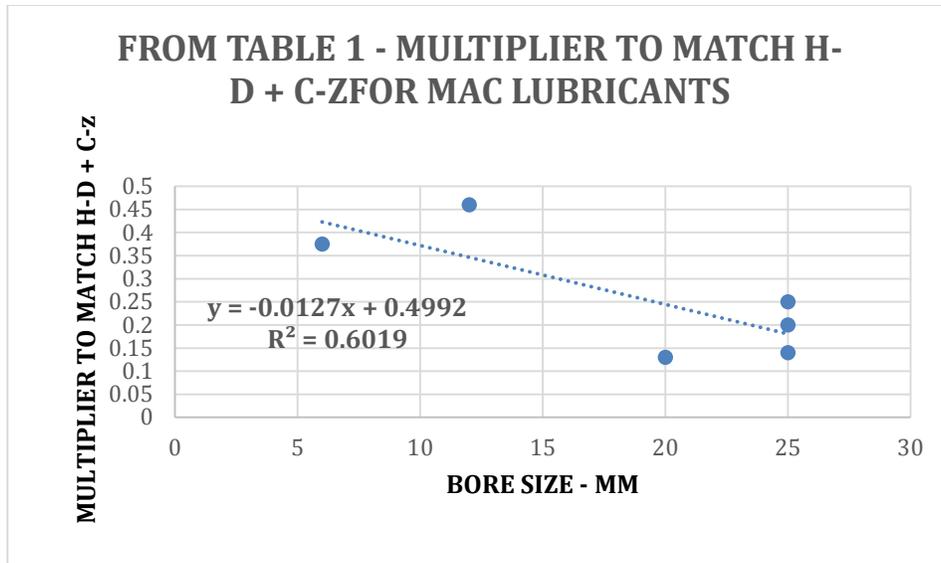
The Hamrock-Dowson equations estimate that the inner ring lubricant film thickness is about 40% of the total. While not exact, assuming that the inner ring EHD is 40% of the total EHD measured should be sufficient for illustrating the inner ring behavior over a range of speed and temperature versus bearing size.

All the modeling estimates of EHD films shown in this paper were based on the software “DYBA” [4]. The MAC lubricants “multiplier” factors needed to match the Hamrock-Dowson + Coy-Zaretsky starvation factors are plotted in Figure 9. Applying a linear fit, we see the reasonably expected reduction as bore size increases, i.e., the larger bearing would have higher starvation at a given speed. We candidly admit that the correlation coefficient is not good but we would need significantly more testing to assure that we have really good control on the test variables. Nevertheless, we will use the multipliers as deduced by Table 1 to make comparative estimates of  $EHD_{ir}$  vs. speed at 24°C (75°F) and 52°C (125°F) from 200 to 6000 RPM.

Conservative practice to prevent ball to race contact and wear is to have a value of  $\lambda \geq 3$  where:

$$\lambda = (\text{EHD film thickness}) / (\text{Composite ball and raceway surface roughness}) \quad [\text{Eq. 2}]$$

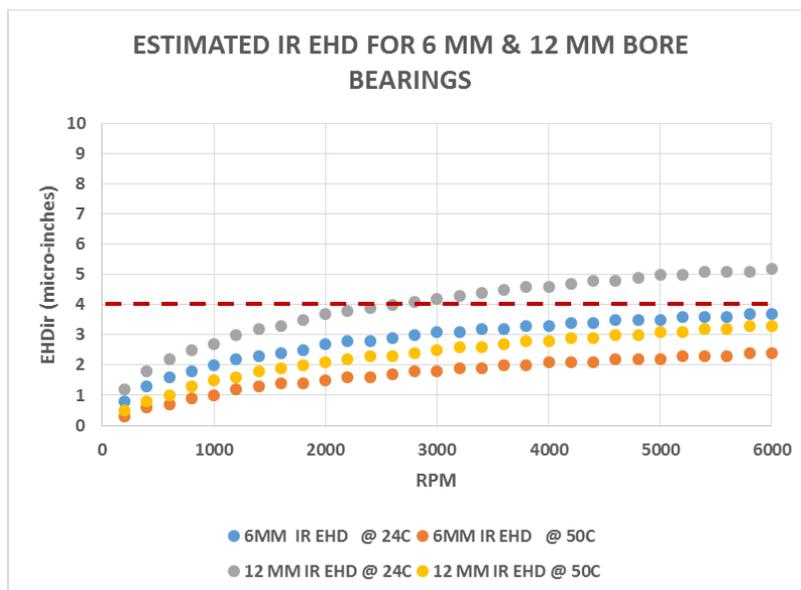
Using a composite bearing race/ball roughness of 2.0, a condition of  $3\lambda$  would require an EHD film of 6 micro-inches (0.2 micro-meters). The test bearings are at least this good.



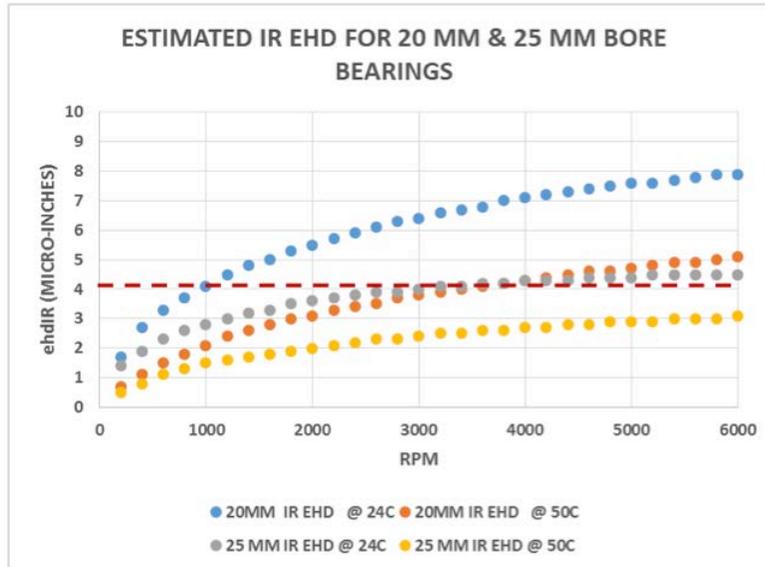
**Figure 9. Multiplier Factors from Table 1, to Match Hamrock-Dawson + Coy-Zaretsky EHD Predictions**

From Figure 9, the multiplier factors used in the analysis were: 6 mm (0.2 in) = 0.42, 12 mm (0.47 in) = 0.35, 20 mm (0.8 in) = 0.25 and 25 mm (0.98 in) = 0.18.

Figure 10 shows plots of the analytical estimates for the EHD film thickness at the inner race for the 6-mm (0.2-in) and 12-mm (0.47-in) bore bearings. Figure 11 shows similar plots for the 20-mm (0.8-in) and 25-mm (0.98-in) bore bearing estimates. To illustrate what might be an adequate lambda, we have placed an arbitrary line at 4 micro-inches (0.1 micro-meters) in Figures 10 & 11.



**Figure 10. Estimated IR EHD for 6-mm (0.2-in) and 12-mm (0.47-in) bore bearings at 24°C and 50°C**



**Figure 11. Estimated IR EHD for 20-mm (0.8-in) and 25-mm (0.98-in) bore bearings at 24°C and 50°C**

The results shown in Figures 10 and 11 illustrate that MAC-based lubricants do not have large margins to separate the balls and raceways and that elevated temperatures and/or smaller bearing sizes increase that danger.

### Conclusions and Observations for Future Work

MAC lubricants have thinner EHD films at comparable bearing running conditions than older natural hydrocarbon lubricants.

Both lubricant types, using EHD predictions with approximately 25% to 50% reduction, seem to agree with actual data on films present.

The thinner films react predictably to raising bearing operating temperatures and speeds, and the resulting film thicknesses at high temperatures and at low speeds in actual applications need to be approached with caution in terms of resulting ball to race contact and possible wear. The low film thicknesses do seem to allow ball metal to race metal wear to start. Both tests presented showed polished wear bands in the race and darkening lubricant with metallic fines in the lubricant. When silicon nitride balls are substituted, the wear is totally mitigated.

Varying the lubricant quantity and examining long term run-in effects on EHD films are obvious next directions for future experiments. In this preliminary study of two lubrication cases, oil and oil with grease added, the EHD thickness values appear to be equivalent. However, long term effects have not been studied.

## References

1. Ward, P. C., Leveille, A. R., "Measuring the EHD Film Thickness in a Rotating Ball Bearing" , *Proceedings of the 39<sup>th</sup> Aerospace Mechanisms Symposium, NASA Marshall Space Flight Center, May 7-9, 2008*
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4. Leveille, A. R> "DYBA – A New Dynamic Bearing Analysis Software Tool" International Symposium on Rolling Element Bearings, Baja Hotel, San Diego, CA May 8, 2007