A STUDY OF THE IN-VACUUM PERFORMANCE OF SPACE LUBRICANTS IN A COMPLIANTLY PRELOADED PAIR OF ANGULAR CONTACT BALL BEARINGS USING THE ADVANCED BEARING TEST RIG

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ABSTRACT

The long duration, cost and sometimes questionable validity of some mechanism development tests, especially life tests, are often debated due to the relative scarcity of definitive data on space lubricant behaviour at bearing level. Such data can verify the use of classical analysis in the definition of lubrication regimes for a test programme or substantiate a proposal for the adoption of an accelerated test campaign.

The recently developed Advanced Bearing Test Rig (ABTR) at ESTL has the capability of concurrent measurement of bearing torque, axial shaft displacement, preload, contact resistance, and temperature, for a pair of angular contact ball bearings operating in vacuum. This paper describes fluid film thickness estimation from axial shaft displacement and discusses the results of characterisation tests on four space (fluid) lubricants. The onset of lubrication regime transitions and associated vibrations are examined; lubrication transition diagrams are drafted.

INTRODUCTION

The operational lubrication regimes of fluid lubricated bearings running in vacuum are often debated in the context of mechanism testing, especially in programmes which require a compressed development cycle. Such regimes are defined and calculated on the basis of classical equations, but despite the many benefits of a greater understanding (lower programmatic risks, the possibility of accelerated testing which may facilitate lower cost or more rapid mechanism development, confirmation of lubricant performance and suitability under all operational scenarios), little experimental validation of such calculations under thermal vacuum conditions exists.

Recent tribological R&D activities have highlighted the need for detailed complementary measurements of lubrication regime changes within bearings through life or variations with speed. In compliant preloaded systems, lubricant film thickness and/or wear or other raceway changes can be inferred from shaft displacement measurements which can provide a highly sensitive means of monitoring sub-micron changes in the average lubricating film thickness for both solid and fluid lubricated bearings and so, for example the transitions between classical lubricating regimes.

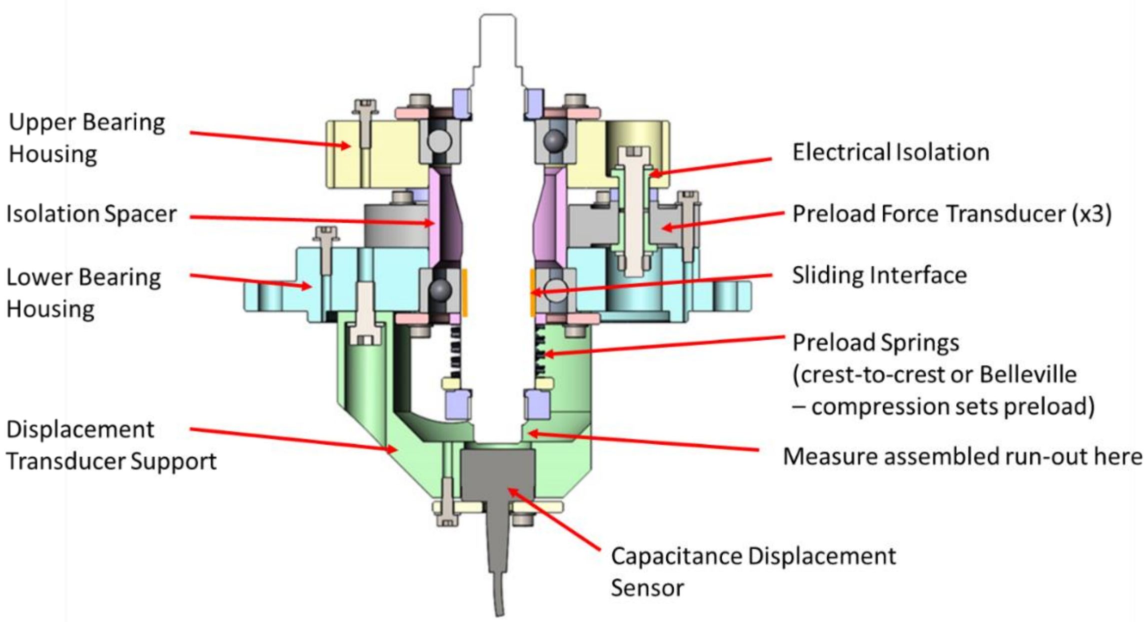

Figure 1. Detailed cross-section of the ABTR bearing assembly.

DESCRIPTION OF THE ABTR

Based on the above measurement principles and inspired by the concept previously reported by Ward et al. [1], ESTL recently developed its Advanced Bearing Test Rig (ABTR). The ABTR has the capability to control operational temperature and to measure and record simultaneously for a pair of angular contact ball bearings operating in vacuum the bearing torque, the axial shaft displacement, the preload, contact resistance, and temperature. Due to the sliding interface of the lower bearing with the shaft (Fig.1), the axial shaft position will be affected by only the film thickness on the ball/inner ring and ball/outer ring contacts of the upper bearing. This technique allows either solid or fluid lubricant film thickness and even raceway wear to be measured, however in this paper we will focus exclusively on fluid film lubrication. In this context therefore, by "film thickness", h, we will refer, hereafter, to the average of central film thickness of the inner and outer ring contacts.

MEASUREMENT PRINCIPLE

A very simple approach to relating the EHL film thickness, h, with the axial displacement of the shaft, δ_A (or C), would be to simply consider the bearing as sliding open on an incline sloped at its rated contact angle.

$$\delta_A = \frac{2h}{\sin\beta} \tag{1}$$

$$h = \delta_A \cdot \sin\beta / 2 \tag{2}$$

Despite its simplicity this is a quite appropriate approximation. A more precise formula, considering the bearing raceway conformity, is found following the approach of Tyler et.al. [3], who were the first to propose such a measurement principle. Assuming a fixed inner ring, when the film is developed, the outer ring must then move to a new position (shown by dashed lines on Fig.2). The axial distance that the outer ring moves is ΔS, which is also the axial displacement of the centre of the outer race curvature, i.e., from C_i to C_i'. The initial contact angle, β, then decreases slightly to β', while the initial contact points on the inner and outer races move from A and B to A' and B'. The measured axial shift for a duplex pair of bearings is:

$$\delta_A = 2\Delta S = BD(\sin\beta - \sin\beta') + \Delta \cdot \sin\beta' \tag{3}$$

where Δ is the increase in the effective ball diameter due to the film. It may be also shown that:

$$\cos\beta' = \frac{BD\cos\beta}{BD - \Delta} \tag{4}$$

Tyler et.al. assume different film thicknesses for the inner and outer rings and require an iterative procedure to solve the problem, but if we assume equal and uniform thicknesses, as above, then $\Delta = 2h$ and an expression for h can be produced directly.

$$\mathrm{h} = 1/2\left(BD - \sqrt{{\delta_A}^2 - 2BD\sin\beta\delta_A + B^2D^2}\right) \tag{5}$$

The above formula leads to results which are very close to the simple case. Tyler et.al. claimed that this sort of measurement of h, using δ_A, is very sensitive to small temperature differences between the inner and outer ring and the front and rear bearing.

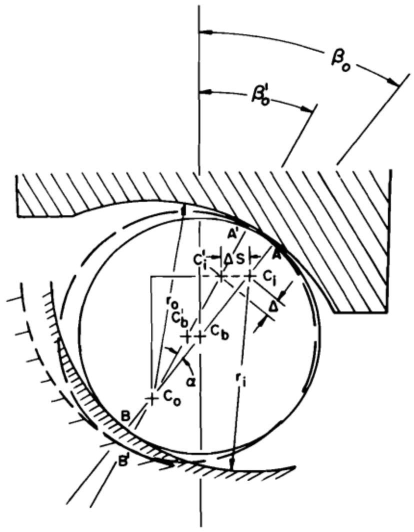

Figure 2. Schematic of an axially displaced bearing outer race according to Tyler et.al. (from [3])

PROCESSING OF ABTR DATA

In experimental terms, in order to translate an axial shaft position measurement into an estimation of fluid film thickness, some more steps are required before applying Eq. 5. Noise must be removed from all the signals with adequate filtering. Torque must be detrended for sensor drift. Although small, the effect of shaft speed on the contact angles at both the inner and outer ring and on the ball loads have to be determined (quasi-static bearing analysis with ESTL's bearing analysis software, CABARET, is applied to this effect). Finally, thermal effects must be addressed.

Thermal calibration

The axial displacement of the shaft (C) is influenced by thermal dilation. In short tests such as torque reversals and speed ramps, detrending C with the position of the shaft at rest at the beginning and the end of the test is all that is required to overcome this; however, for long tests, a model of the thermal displacement is required. Temperature is measured at 4 positions in the ABTR housing (T_U - upper, T_L - lower, T_H - heat exchanger, T_R - lower redundant). The ABTR, with pair of dry bearings, and the motor at rest, was submitted to a complex thermal cycling routine, between -25 and 60°C. Several models based on those 4 temperatures were examined; it was found that C correlates almost linearly with T_L and T_R.

Lubricant temperature and frictional heating

Since it is the closest temperature to the bearing whose film thickness we measure, the upper bearing temperature is used for the purposes of EHL calculation. The temperature of the lubricant inside the bearings is higher than the outside of the housing. The housing temperature reacts with some hysteresis to frictional heating at the races. That has to do with the heat capacity and thermal resistances of the entire bearing assembly.

Bearing conductance from the lubricant to the outside of the housing where the T_U – upper thermistor is located, has been calculated using a simple thermal network analysis. Then, the following expressions are applied:

a) In steady state:

$$T_{lub} = T_{ext} + P_{loss}/W_s \tag{6}$$

where $P_{loss} = \text{(torque)x(rpm)}/9550$ (7)

b) In a transient situation:

$$T_{lub} = T_{ext} + \bar{P}(t)/W_s \tag{8}$$

where $\bar{P}(t) = \int_{t_0}^{t_1} P(t)dt$ (9)

EHL modelling

The film thickness, h, estimated by the ABTR is the average film thickness for the inner and outer ring and for all the balls of the bearing. The film thickness is calculated separately for the inner and outer race of the bearing and averaged, since the entrainment speed is different. The equations of Dowson & Hamrock [5] and of Chittenden [6] are used to provide theoretical curves against which measurements are compared (the latter being a generalisation and recasting of the former). A correction factor for inlet shear heating of the lubricant is applied, using Gupta's approach [7].

The quantity which is of primary interest is ***central film thickness***, h_c, the average thickness at the centre of the EHL zone, since that is what characterises the separation between the ball and race which produces that axial displacement of the shaft. This quantity averaged over both inner and outer ring contacts is compared to the ABTR measurement and referenced in all figures in this text.

On the other hand, the contact resistance measurement and the examination of the lubrication regime are related to the ***minimum film thickness***, h_{min}, the thickness at the outlet where the EHL meniscus is forming a constriction, through the ***Lambda-ratio*** (Λ), the ratio of h_{min} to composite RMS roughness, s, of ball and raceway. Conventionally the limits between the main fluid lubrication regimes have usually been defined as follows:

- $\Lambda<1$: Boundary lubrication.
- $1< \Lambda <3$: Mixed lubrication
- $3< \Lambda <10$: Elastohydrodynamic lubrication (EHL).
- $\Lambda >10$: Hydrodynamic lubrication

A formula for estimating the minimum film thickness from the central film thickness over a broad range of operating conditions has been recently proposed by Sperka et.al. [8] This has been used for our calculations of the Lambda ratio.

TEST CAMPAIGNS

This paper describes the considerations involved in correctly estimating fluid film thickness through axial shaft displacement and discusses the results of characterisation tests on four space lubricants including a MAC, a mineral oil, a PFPE oil and a PFPE grease at three temperatures (-20, 22, and 60 °C) as a function of lubricant fill and preload (Tab 1). Detailed results of these campaigns are included in reports available at ESTL's site [3-4]. For each test a suite of 5 different characterisations is performed, involving speed ramps, speed steps, torque reversals, high frequency observations and 6x6h endurance tests.

Test	Temperature (°C)	Oil fill	Preload (N)
T01	22	Low	12
T02	-20		
T03	60		
T04	22	Nominal	
T05	-20		
T06	60		
T07	22	Low	48
T08	-20		
T09	60		
T10	22	Nominal	
T11	-20		
T12	60		

Table 1. Matrix of testing conditions, identical for all 4 lubricants.

RESULTS AND DISCUSSION

EHL film thickness: measured vs theoretical

When compared to the expected film thickness from theoretical EHL calculations, the induced measurements appear to be consistently thicker, by a factor larger than the observed measurement error. In fact, when the bearing is evidently in the EHL regime, as evidenced by the film thickness behaviour and the contact resistance measurements, a shift appears, amounting to ~2μm in film thickness terms (or 20μm in axial displacement terms). This shift marks the transition between the mixed and full EHL regime and covers the area where contact resistance measurements are between 0% and 100% (Fig.3). It has a characteristic shape when C or h are plotted against U or the theoretical h or Λ-ratio. It presents a slope that is a multiple of that expected by the calculated EHL behaviour. It happens generally between Λ-ratio (theoretical) values of 1 and 3, which is commensurate of it being a characteristic of the mixed regime. Also, it is independent of acceleration or deceleration (hence thermal history), both in direction, as well as magnitude, as evidenced by comparing results from speed ramps against reversals and long-term tests.

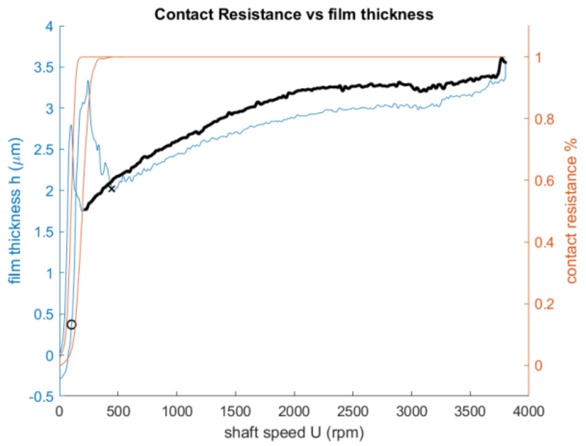

Figure 3. Raw film thickness as derived from axial shaft position and contact resistance. The EHL zone in acceleration is highlighted in black. (T10, Nye2001A)

Evidently, this shift is a characteristic of the dynamic behaviour of the bearing, when the contacts transition from asperity-to-asperity solid contacts to contacting via the fluid squeeze film. It is thus a manifestation of a change in the lateral position of the balls and the contact angle, a cause of axial shaft displacement quite different from changes to the film thickness. Therefore, in order to proceed to a quantitative measurement of the film thickness in the EHL zone, this "dynamic shift" must be measured and removed.

A few methods to do this have been investigated. The most practical one is to specify a conventional point within the EHL zone, at which the film thickness curve can be anchored; i.e. shift the curve so that the measured thickness at this point matches the theoretically predicted one. This of course means that the film thickness measurement will be relative. However, when this is done effectively, the measurements are found to quite closely match theory, including thermal effects like shear thinning (Fig. 4).

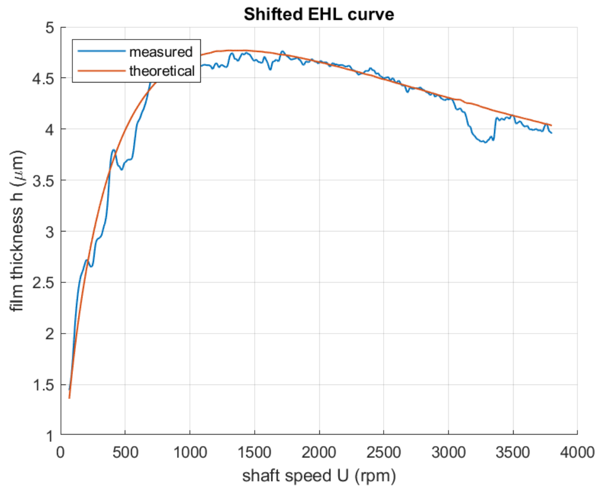

Figure 4. Match between measured film thickness after applying the shift and the prediction of the thermal EHL model (T11 SRG60).

The anchor point that is most suitable is the one where, according to theoretical calculations, the film thickness should have a specific Λ-ratio. Λ=5 is used, as it is compatible with all measurements. Indeed (Fig. 5), when plotting the Λ derived from measurement vs Λ derived from theoretical film thickness (Λ reference), by anchoring at Λ=5 all curves collapse into a master curve. For low Λ, the measured value drops quickly (in the mixed regime) to zero for Λ_{ref}=3. For high Λ, the measured value is higher than the theoretical prediction, which is an outcome well documented in the literature for theoretical/measured EHL curves.

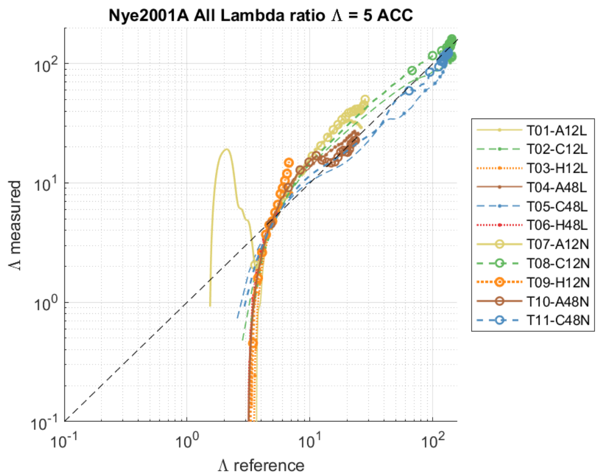

Figure 5. Measured vs theoretical Lambda-ratio after applying the shift. (Nye2001A, speed ramp acceleration)

Observable effects

The fluid lubrication effects observed with the ABTR are summarised in Fig. 6-7. In context of the illustrated numbers, these are:

1. **Boundary lubrication**: A zone where the film thickness is zero and does not change with speed appears, as expected. Of course, this is more pronounced in hot and ambient tests.
2. **Shear thinning**: The film thickness plateau decreases at high speeds and for deceleration, observed mainly in cold tests.
3. **Starvation (oil fill)**: The film thickness is lower bearings operating under a low fill of free oil when compared to bearings with a normal fill, observed mainly in cold tests.
4. **Transition to EHL**: it shifts the apparent film thickness during the transition through mixed lubrication.

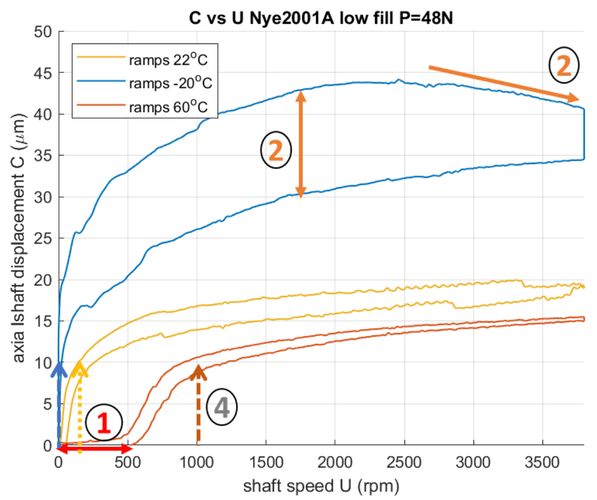

Figure 6. Observable effects: temperature comparison. (Nye2001A low fill, nominal preload)

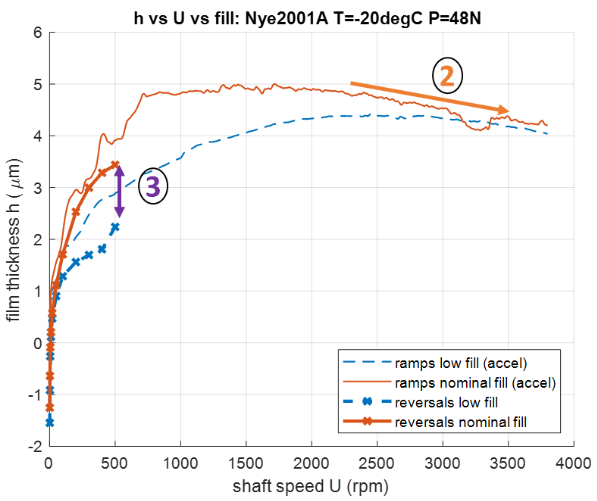

Figure 7. Observable effects: low vs nominal fill. (Nye2001A -20°C, nominal preload)

Effect of tested parameters

Temperature: The effect of temperature and its corresponding fluid viscosity on the film thickness can be quantified on the ABTR. The effect of inlet shear heating and starvation is evident. Also, in this case the contact was in the boundary lubrication regime during the hot test.

Preload: Film thickness decreases with preload. Preload has a more pronounced effect than predicted in literature, because of higher frictional heating which impacts the local fluid temperature within the bearing, but which is not captured in the housing temperature measurement due to thermal lag.

Oil fill: As expected, a lower oil than nominal produces lower film thicknesses to some degree, however at higher speeds, inlet shear heating (ISH) catches up and the film thicknesses for low and nominal fills converge. The thermal effect of lubricant churning can be observed.

High frequency observation

The FFT's of both the displacement and torque signal measured are obtained, during steps of constant speed running at increasing speeds (0.1-2000rpm). The FFT spectra of the signals are produced. The peaks of the FFT's are automatically detected. The results are presented in the form of a waterfall graph for each test, where each line represents an FFT spectrum at a specific shaft speed. The z-axis represents absolute intensity of the signal (Fig. 8).

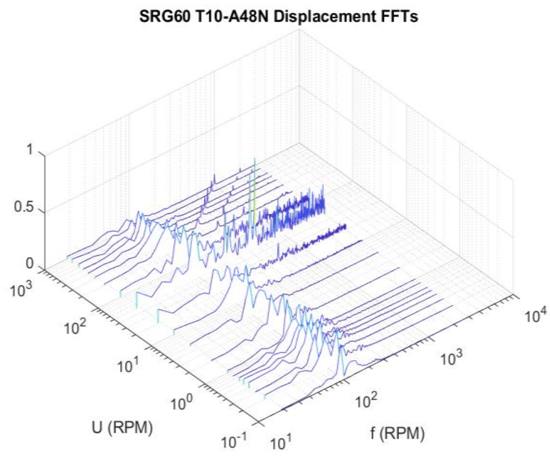

Figure 8. FFT spectra of axial displacement at various shaft speeds (SRG60 at ambient temperature, nominal fill and preload).

The most characteristic observation is the appearance of broadband noise at certain shaft speeds, which correspond to what the contact resistance readings suggest are the zones of mixed lubrication, where intermittent contact and the emergence and collapse of the EHL film and its related cavitation bubble would plausibly cause noise to appear for both the displacement and torque signals. These zones change location depending on the temperature (10-50 RPM for cold, 100-200 RPM for ambient, 200-1000 RPM for hot).

Transition diagrams

Transition diagrams are useful for mechanism design purposes, in the sense that the appropriate lubricant may be selected for the desired operational envelope, to avoid excessive vibration or wear of the bearings. Perhaps more interestingly, they can also be used to inform a development test campaign, by providing for the first time an experimentally justified confirmation of the extent of the boundary, mixed and EHL regimes, and hence permitting the identification of operational regimes where, acceleration of a test could be considered tribologically valid from the point of view of the likelihood of asperity contact. Transition diagrams are shown below (Fig. 9) for Nye 2001A and Fomblin Z25 oils respectively, however behaviours of all 4 oils tested were in line with expectations based on their rheological properties.

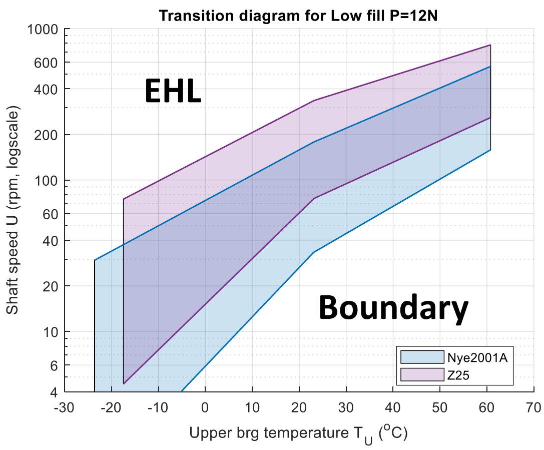

Figure 9. Comparison of transition diagrams for Nye2001A and Z25 for low fill, low preload. The shaded area delimits the mixed lubrication regime.

Starvation

Starvation ***induced by entrainment velocity*** has been observed in the ABTR. Its characteristic is an exponential decay with speed, which appears linear with a negative slope in log-log scale. However, it must be decoupled from Inlet Shear Heating-related (ISH-related) film thickness reduction which may induce an additional shift if it makes the contact revert to the mixed regime.

This has been observed on speed ramps both in grease, as expected, but also in oil; In Fig. 10, it can be seen how Nye2001A starves at higher speeds whereas SRG60 does not, despite having the same EHL film thickness up to 2000RPM. Likewise, for Braycote grease starvation occurs at 2000 rpm in acceleration and 1500rpm in deceleration, whereas for Z25 there is none.

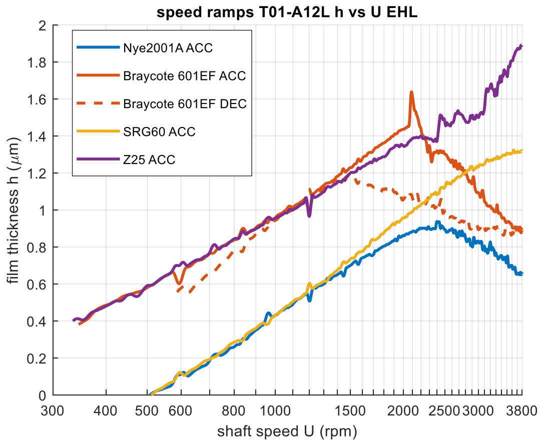

Figure 10. EHL film thickness as a function of speed (logscale); ambient temperature, low fill, low preload.

Starvation has also been observed in the guise of an instability, leading to transitions between mixed and EHL regimes over the course of hours of running. This has so far been observed on all 3 of the oils. A very characteristic example is presented in Fig. 11: shaft speed, U, upper bearing temperature, T_U, shaft axial displacement, C, and contact resistance, CR, are plotted in parallel during 6x6h tests at different speeds. At 1, 100 and 120 rpm the bearing is running in boundary, at 500 rpm in mixed and at 1000 and 2000 rpm in EHL, originally. However, after about 2h of running, an EHL-to-mixed-lubrication transition occurs, from which at 1000 rpm there was no recovery. A similar unpredicted film thickness reduction occurred also for operation at 2000rpm, but was followed by recovery, and even a second transition/recovery event later in the test.

These transitions act to highlight the importance of lubrication endurance testing. Running such a test on an instrument with the capability of the ABTR can provide insight into questions such as:

1. Is there any long-term time-dependent starvation occurring?
2. Does operation for a prolonged endurance of itself initiate lubrication regime change?
3. Are the regimes identified from ramps/steps or reversals stable or unstable over longer periods?
4. How does the film thickness equilibrium interact with thermal equilibrium?
5. Is there any short-term lubricant degradation?

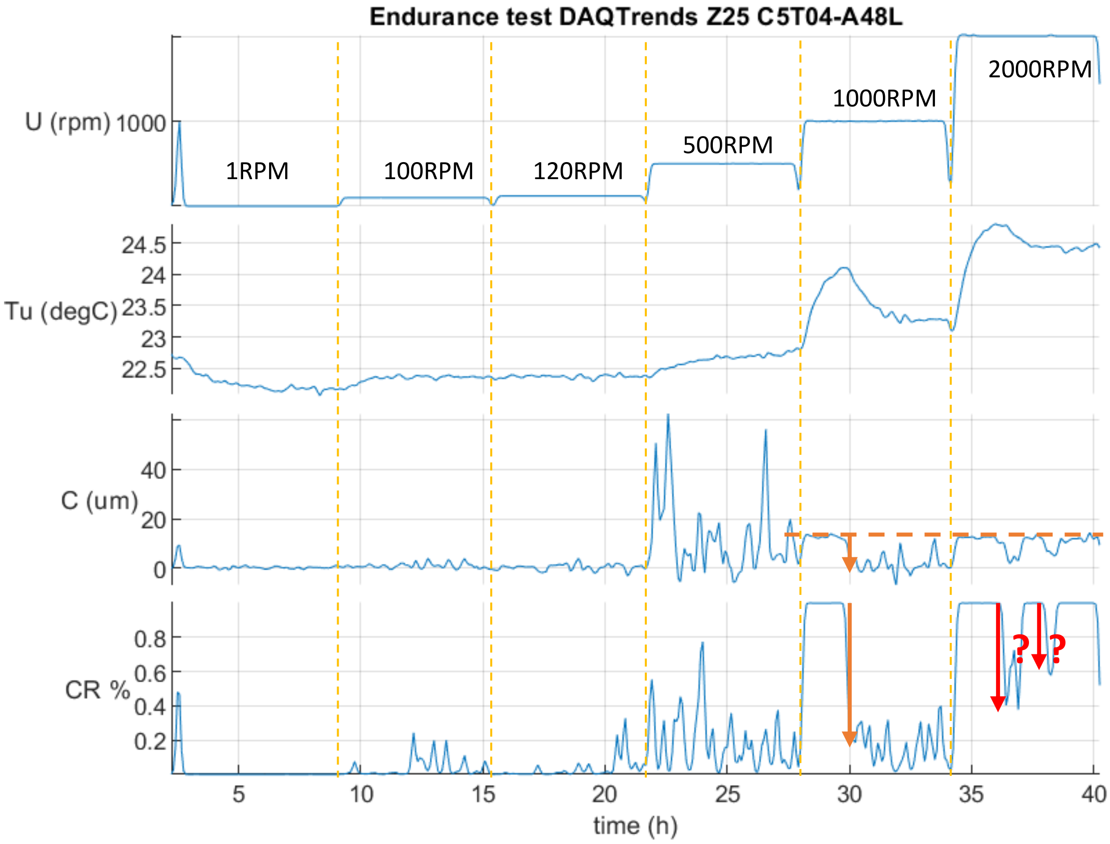

Figure 11. Starvation incidents occurring during 6h endurance tests with Z25.

In the case of solid lubricated bearings, extending axial shaft position monitoring with the ABTR to the entirety of a bearing lifetime test (to 60 million revolutions or more in a present test campaign) can help answer similar questions:

1. Is the running of the bearing really largely speed-independent; and can testing be accelerated and by what factor?
2. How long does running-in actually last?
3. Is the rate of material transfer between the cage, balls and raceway stable with time/speed?
4. In case of instability, does friction, material wear/transfer rate or temperature change?
5. Is there significant wear within the bearing sufficient to modify/lose preload in a rigidly preloaded system?

CONCLUSIONS

The paper shows the tested lubricants follow EHL theory and demonstrates how ABTR data may ultimately support a greater understanding of accelerated testing, the impact of operational conditions on lubricant performance, the onset of wear and the loss of preload both for fluid and dry-lubricated bearings.

In addition, it is shown that that low fill affects film thickness and torque only where significant starvation or churn heating are observed and despite the classical assumption that film thickness is not well correlated to preload, in fact the preload does have a moderate effect in vacuum, mainly via its effect on frictional heating.

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