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Direct Measurements of Grease Film Thickness in a Rolling Bearing Using a Novel Sapphire Bearing Rig with Optical Interferometry

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ABSTRACT

This paper reports direct measurements of lubricant film thickness in a radially-loaded grease-lubricated model rolling bearing. These measurements are performed on a recently developed set-up, termed the Sapphire Bearing Rig, which allows for direct observation and measurement of film thickness in a single ball-outer ring contact in a rotating bearing using optical interferometry. Measurements are made with a selection of greases covering different base oils and thickener types. The results show that starvation occurs as the bearing speed increases. This is confirmed by observations of the inlet meniscus. The bearing speed at which starvation occurs is determined by grease properties, particularly the base oil viscosity. When the bearing operates for a prolonged time at a constant speed, as is usually the case in practice, the initial film thickness is seen to decay in the first few hours of operation, to eventually reach a steady state value that is lower than that predicted by elastohydrodynamic lubrication (EHL) theory. For conditions tested here, this steady-state film thickness was largely independent of bearing speed, which suggests that it is mainly governed by the level of lubricant supply in the inlet. Such film behavior clearly cannot be predicted using the standard EHL theory. Stopping the bearing for a sufficiently long time leads to a temporary recovery of the film thickness upon restart. These measurements provide new insights into grease lubrication mechanisms of rolling bearings and can aid the development of new tools for improved prediction of grease film thickness in rolling bearings.

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KEYWORDS
Rolling bearings; elastohydrodynamic lubrication (EHL); grease

Introduction

The life of a rolling bearing is limited by its fatigue life if the bearing is well lubricated. The quality of lubrication is primarily determined by the thickness of the lubricant films separating the rolling elements from the rings relative to the roughness of the surfaces. If the films are thin relative to the roughness heights, then the bearing life will be reduced by increased stresses caused by roughness asperity interaction. (*1*,*2*) For oil lubricated bearings, the theory to calculate the film thickness is well established. Simple regression equations are available that can be used to estimate the oil film thickness under the condition that the contacts are fully flooded with oil. (*3*,*4*) In the case of grease lubrication, this is more complex. (*5*) A lubricating grease consists of not only oil but also of a thickener giving the grease its stiffness. The thickener may contribute to the film thickness (*6*) or act as a reservoir from which oil is slowly released called "oil bleed." The first effect will increase the film thickness versus that calculated using the base oil properties, but the second effect may cause starvation reducing this film. The exact mechanism depends on the grease type, operating conditions, and bearing type/dimensions. For an extensive description of the grease lubrication mechanisms, the reader is referred to Lugt. (*5*,*7*)

Thus far, most of the research performed on film thickness in grease lubricated bearings has been conducted on single contacts. Pioneering work on this was performed by Cann and coworkers (*8*–*13*) but also others. (*6*,*14*,*15*) The film thickness in a grease lubricated contact at very low speeds can be much higher than that calculated using oil EHL theory with the base oil properties (*8*,*16*–*20*) and can even be calculated. (*21*) However, under "normal" running conditions, starvation will occur. Starvation has been investigated extensively for single and even multiple contacts, starting in 1971 by Wedeven et al. (*22*). Models for starved contacts have been developed for point and elliptical contacts. (*23*–*25*) and there have been very recent efforts to predict grease film thickness in a bearing using an empirical approach. (*26*) However, it is still not possible to analytically calculate the lubricant film thickness in a grease lubricated rolling bearing. Input to these starvation models is the inlet combined (oil) layer thickness. The thickness of these layers is the result of the mass balance between feed and loss to

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the contacts. (27) Various aspects to this have been considered (described in Lugt (5)); however, the ability to apply such models to a real rolling bearing is severely limited due to the lack of understanding of the onset of starvation in a bearing and general mechanisms of grease lubrication. Film thickness measurements in a grease lubricated cylindrical thurst bearing were done by Muennich and Gloeckner (28) by measuring the displacement of one of the rings as a result of film formation. They found very thick films; however, this could have been caused by the fact that their measurements were done for short times. Measuring the film thickness in grease lubricated radial bearings could thus far only be done using the "capacitance method" where the bearing is assumed to be an electrical capacitor. The capacitance of a single contact in a bearing is approximately inversely proportional to the film thickness. (29) There are several problems with this technique. The first is the proportionality constant, which contains the dielectric properties of the lubricant. This is usually unknown and is difficult to measure, not least because it is a function of contact pressure. Second, the measured capacitance signal is the result of the capacitance of all contacts in the bearing. Therefore, the film thickness in a single ball-ring contact is not measured directly by this method but can only be inferred *via* an electrical model of the bearing (30) and only for a pure axial load where all balls are equally loaded. (31) Alternatively, all balls could be replaced by ceramic balls except for one in combination with a trigger. (32).

The first measurements with this method for grease lubricated bearings were done by Wilson. (33) He compared the measured grease films with those predicted for the base oil for a cylindrical and a spherical roller bearing and found grease films that were about 20–25% thicker than predictions for oil immediately after bearing relubrication, but became thinner than those predicted for oil after a short running time following the lubrication. For ultra-low speeds, Cen et al. (18) measured the film thickness in both single contacts and in a full bearing showing that the fully flooded single contact results agree well with that of a full bearing at very low speeds. Barz (31) found for radially loaded angular contact ball bearings that the ratio of measured film thickness to calculated film thickness using the base oil viscosity was 16–20%. Franke and Poll (34) using the same rig found a similar value of this ratio ("approximately 20%") for lithium and calcium-based thickeners. The work of Barz (31) and Baly (30) revealed some unexpected phenomena. The first surprising result was that the film thickness is a function of bearing speed at lower speeds but independent of speed at speeds higher than about ndm = 160,000 rev/min · mm. The second unexpected result was that the film thickness was independent of the number of balls. Similar results were found in Cen and Lugt (35) where it was shown that contact replenishment is a local phenomenon and hence it does not happen in-between the balls in a ball bearing.

The optical interferometry technique is more established as a method to accurately measure the film thickness with reported resolution of a few nm. The general method is well-recognized and described in detail elsewhere. (36,37) Optical interferometry has become a standard technique to measure oil film thickness in a single ball on ring contact. Provided that a bearing can be designed to afford optical access to a ball-ring contact, optical interferometry may in theory be used to measure the film thickness in a single specific ball-ring contact in a radially loaded bearing rather than only an "average film thickness" across all ball-ring contacts in an axially loaded bearing only as is the case with the general capacitance method described ealrier (although very recent work has shown that it is possible to separate the film thickness from the inner ring and the outer ring contacts using capacitance, at least in an axially loaded bearing (38)). Major advantages of this optical method are its accuracy and the fact that it makes the visualization of the flow around the contacting surfaces possible, which cannot be done with any other method. The main disadvantage is that one of the contacting surfaces needs to be made optically transparent so it needs to be made of glass or sapphire instead of steel and that it, thus far, requires one flat surface. Ohno et al. (39) used an optical interferometry method to measure the film thickness in an axial ball bearing where one of the washers was standard but the other was made of sapphire and was flat. The rings were counter rotating to fix the position of the balls, that is, the contacts themselves were stationary unlike in a real bearing. This makes the set-up less representative but avoids problems in acquiring images of a fast-moving contact. As the authors mention in their paper, ball spin, which is significant in this set-up, had a major impact on the results. They also showed that the grease film thickness depends on speed and operating time. The differences between the grease and base oil films were not as significant as those reported in the references previously mentioned because of the short testing times (30 min only) and low speeds. However, based on extrapolation to higher speeds and longer times, the authors concluded that the films will reduce to 30–60% depending on the grease type that was used.

Recently, a new, novel set-up for directly measuring lubricant film thickness in a ball-ring contact in a radially loaded model ball bearing was developed at Imperial College London. The set-up is fully described in Chennaoui et al. (39) where initial results with oil are also presented. It employs both optical interferometry and novel ratiometric laser induced fluorescence techniques to accurately measure the lubricant films in and around the ball-ring contact respectively. This offers obvious advantages over the previous approaches previously described as it makes it possible, for the first time, to directly measure film thickness in a radial bearing. It also allows for observation of the oil layers around the contact that aids in interpretation of the in-contact film behavior. The present work utilizes this set-up to provide improved understanding of the film thickness in a grease lubricated radial ball bearing.

Experimental methods and materials

Test greases

Four types of grease are used in this study. The greases were chosen to cover a range of thickener and base oil types as

Table 1. Properties of the test lubricants used in the present study.

Tested grease	Base oil type	Thickener type	Oil visc. [mm^2/s] 40 °C	100 °C
PU-E	Ester	Polyurea	70	9.4
Li-M-1	Mineral	Lithium	98	9.4
LiC-PAO	PAO	Lithium compl.	100	14.5
Li-M-2	Mineral	Lithium	32	5.0
Li-M-1 bled oil	Mineral	–	98	9.4

well as a range of base oil viscosities. Li-M-1, PU-E, and LiC-PAO are fully-formulated commercial greases that are designed for medium and high-speed bearing applications. Li-M-2 is a custom-made additive-free grease with the same type of base oil and thickener as the Li-M-1 grease but a lower base oil viscosity. In addition, the bled oil from the Li-M-1 grease was also tested. The bled oil was obtained by centrifugation. Table 1 lists the properties of the tested greases and the one corresponding bled oil. As can be seen, the greases cover three types of thickener, polyurea, lithium, and calcium sulfonate and two types of base oil, ester and mineral oil.

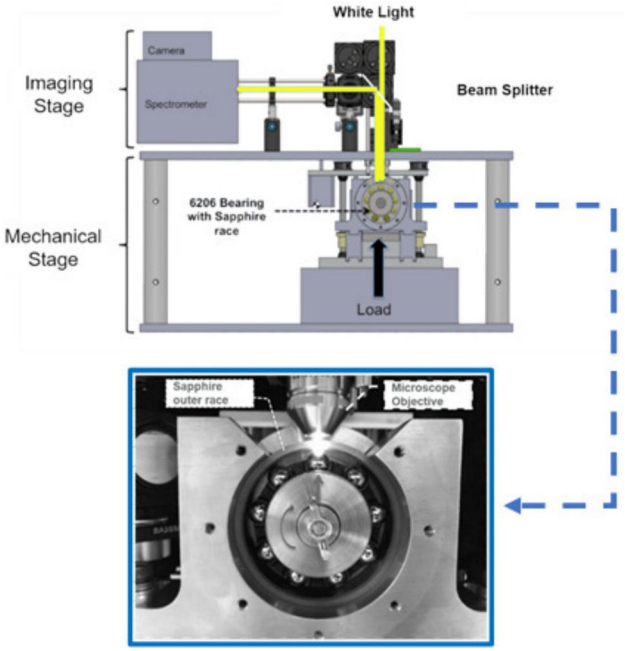


Figure 1. A schematic of the Sapphire Bearing Rig (SBR) with the optical interferometry set-up. (39)

Experimental equipment and procedures

All experiments in this study are conducted on a newly developed unique rig that enables direct measurements of the lubricant film thicknesses in and around a ball-ring contact in an operating model ball bearing. The set-up is referred to as the "Sapphire Bearing Rig" (SBR) and is described in detail in Chennaoui et al. (39) Only a brief description is provided here. The rig employs a 6206 deep groove ball bearing where the outer steel ring has been replaced with an equivalent sapphire (or glass) one. This allows optical access to the ball-outer ring contact and hence direct measurement of the lubricant film in this contact. The set-up utilizes both optical interferometry and a novel ratiometric laser induced fluorescence technique for film measurements. The former is used to measure the in-contact EHL film thickness in the ball-outer ring contact with accuracy of a few nm. The latter is used to measure the thickness of the lubricant films around the contact and, in particular, the inlet of the contact, so that the level of lubricant supply in the track between successive ball passages can be characterized quantitatively. Only the optical interferometry mode is utilized in the present study of grease lubrication. The inner ring is driven by an electric motor at speeds of up to 3,000 rev/min. The outer ring is stationary. The loading system utilizes a stepper motor connected to a loading arm. A custom triggering system and fast optics are employed so that required interferometry and fluorescence images can be obtained in the correct location and with sufficient clarity despite the ball-ring contact moving through the field of view. The method allows for direct measurement of lubricant films in a bearing operating at realistic speeds (up to 3,000 rev/min) and loads (outer ring contact pressures in the region of 1.6 GPa with a sapphire outer ring). In the present tests, the outer ring was made of glass, rather than sapphire, and its inner surface was coated with a thin chromium and silica layers to enable optical interferometry measurements. It must be noted that, unlike in a real deep groove ball bearing, the transparent outer ring in the SBR is flat in the direction transverse to rolling. This has obvious implications in terms of contact dimensions but may also affect lubricant supply to the contact and hence the exact conditions when starvation may occur, as discussed in depth in Chennaoui et al. (39) It is possible to avoid this complication by using a cylindrical roller bearing in this set-up, so that the flat outer ring would be representative of that in a real cylindrical roller bearing. However, owing to the line contact that would be produced in that case combined with the loading limits of the rig, the maximum pressure in the so formed roller-ring contacts would be much lower than in the current ball bearing arrangement, so much so that EHL conditions could not be ensured. Hence the roller bearing option is not used here. A schematic of the rig in the optical interferometry mode as used in this study is shown in Fig. 1.

In all tests in this study the applied radial bearing load was 30 N, which corresponds to a maximum Hertz pressure of 0.73 GPa in the most highly stressed ball-outer ring contact where the film is measured. This level of pressure ensures that the contact is operating in the EHL regime and is similar (or higher) to that typicaly used in the single contact optical interferometry experiments. At this load, the maximum semi-contact lengths for the elliptical contact between the ball and the outer race are 112 μ m in the rolling direction and 97 μ m in the transverse (axial) direction; the latter is smaller owing to the fact that the outer ring has no curvature in the transverse direction. Experiments were conducted at bearing speeds between 2 rpm and 3,000 rpm, which corresponds to entrainment speeds of 1.2 mm/s to 1.75 m/s or ndm of 138 to 138,000 rev/min · mm. All tests were done at room temperature which was kept constant at

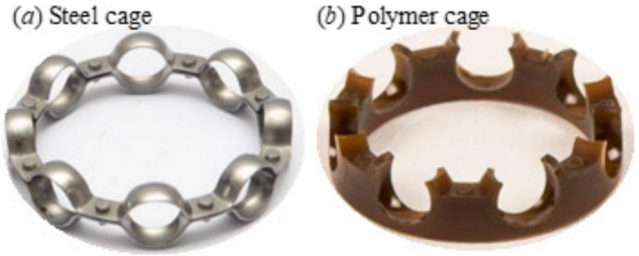


Figure 2. An example image of the two bearing cage types used in this study (a) riveted steel cage; (b) standard polymer cage. (Note: this is a generic image of the cage types, not the exact cages used in this study).

25 + 1 °C. The temperature of the outer ring close to the contact is measured by a thermo couple. The grease sample is spatulated both on the outer ring and the inner ring before the test and no additional grease was added during the test. The total amount of grease used was 2 g, other than in selected tests where 4 g was used for comparison (as stated in results section where relevant). The central film thickness in the ball-outer ring contact is measured using thin film optical interferometry as described earlier in Chennaoui et al. (*39*) and in parallel, a camera is used to capture an image of the contact and inlet oil reservoir using a monochromatic light-emitting diode (LED) as a light source. The latter allows for quantitative assessment of inlet supply conditions and a quantitative measurement of the inlet meniscus distance (distance between the center of the contact and the inlet meniscus) to be made. This, in turn, aids in interpreting the trends observed in the measured in-contact film thickness. The SBR rig does allow for a more detailed, quantitative evaluation of the inlet films using ratiometric fluorescence measurements but this was deemed unnecessary for the aims of the present study. Each film thickness value presented in this paper is an average obtained from 200 interferometry images at the same given operating condition. This accounts for inevitable variation in films with time commonly seen with grease lubrication, as well as any variation in film between different balls in the bearing of which there are nine. Tests were conducted with a standard polymer cage with all greases. One additional set of tests with a single grease was done with a steel cage to assess how/if the type of cage may affect bearing film thickness, since it is known that the type of cage can affect grease replenishment to the contact. The two types of cage are shown in the image in Fig. 2.

Results

Evolution of grease film thickness with increasing bearing speed

Figure 3 shows the measured central film thickness against bearing rotational speed for Li-M-1 grease and its bled oil. The theoretical film thickness, predicted using the Hamrock-Dowson EHL equation (*3*) with base oil viscosity, is also plotted for comparison. Grease Li-M-1 forms a thicker film than the predicted one when the rotating speed is lower than 1,000 rpm. At bearing speeds higher than about 1,500 rev/min the measured film thickness starts to

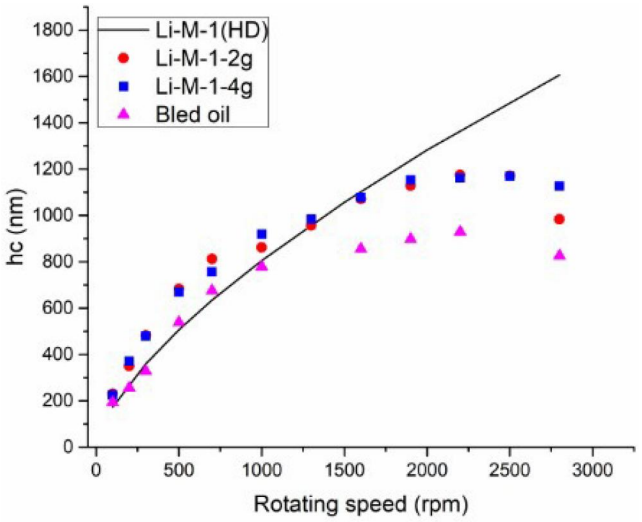


Figure 3. Measured central film thickness in the ball-outer ring contact versus bearing rotational speed for Li-M-1 and its bled oil compared with the theoretical film thickness for fully-flooded conditions calculated using the Hamrock-Dowson equation and base oil properties. Polymer cage, $p_H = 730$ MPa, $T_0 = 25$ °C, grease fill: 2 g, 4 g, oil fill: 2 mL.

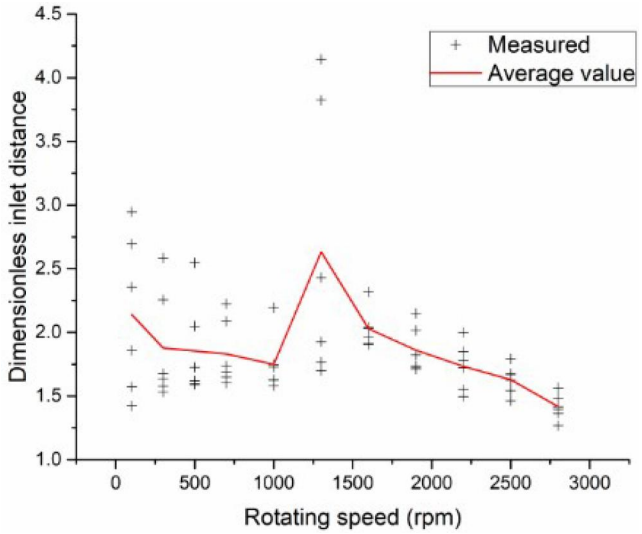


Figure 4. Measured inlet distance and its average value versus bearing rotational speed for Li-M-1. Polymer cage, $p_H = 730$ MPa, $T_0 = 25$ °C, grease supply: 2 g. Corresponding film thickness results are shown in Fig. 3.

gradually fall below the predicted value and at highest speed of 2,800 rev/min it is some 40% lower than the predicted value. This drop in the film thickness with speed is likely to be caused by starvation. Tests were repeated with a higher grease fill of 4 g, instead of previous 2 g. The film thicknesses were largely unchanged at all speeds suggesting that the drop in film at higher speeds was not due to insufficient grease fill. This also suggests that only a small amount of grease is entrained into the contact and participates in the film formation while most of it is pushed out away from the contact. Thus, increasing grease fill is unlikely to prevent or postpone the onset of starvation. Comparing the central film thickness of Li-M-1 and its bled oil, it can be clearly seen that the initial film formed by grease Li-M-1 is always thicker than the film formed by its bled oil for all speeds. At low bearing speeds, this film enhancement is in line with

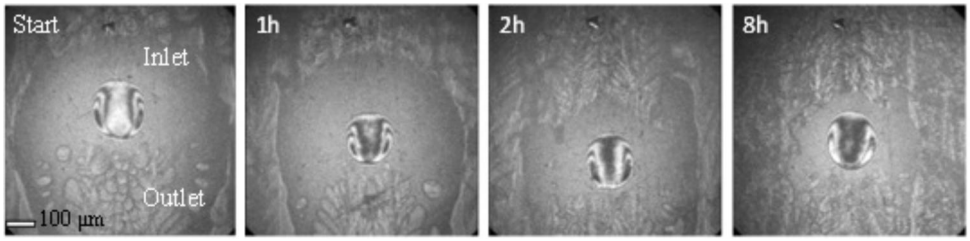


Figure 5. Images of oil reservoir around the ball-outer ring contact at the start of test and 1 h, 2 h, and 8 h of running time at 1,000 rpm. Li-M-1 grease, polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

the well-publicized single contact measurements and is most likely caused by the thickener contribution to film formation. (*18*,*20*) However, at thicker films, thickener contribution may be less significant such that any differences are probably also due to different starvation behavior of oil and grease.

Figure 4 shows the dimensionless inlet distance measured using the contact images as previously described, plotted against the same range of bearing speeds as used in Fig. 3. The inlet distance is defined here as the distance from the center of the contact to the edge of the oil meniscus. The dimensionless inlet distance is the ratio of the measured inlet distance and the Herzian half contact width in rolling direction. A number of measurements of inlet distance are made at each speed as shown in the figure. It can be seen that there is a significant variation in the values particularly at lower speeds. This indicates that the grease is not spread evenly in the bearing track during operation at low speeds, even though we made an effort to spatulate as even a layer of grease as possible on the ring before the start of the test. This variation reduces significantly at higher speeds, which suggests that a higher speed promotes a homogeneous distribution of grease on the track. Grease is a strong shear thinning material so that with increasing speed, the grease may flow more easily. It is evident that the average value of the inlet distance decreases with bearing rotating speed, leading to starvation at higher speeds. This is in line with the film thickness measurements shown in Fig. 3 where films at higher bearing speeds were lower than expected from fully flooded EHL theory and as such, confirms that the observed reduction in film thickness at high speeds is due to the onset of starvation. An interesting observation in Fig. 4 is the sudden increase of inlet distance at bearing speed of around 1,000 rpm. This is probably caused by the centrifugal-forces-induced random event where some grease situated away from the track is suddenly driven into the track and thus causes temporary enhancement in the contact replenishment.

Evolution of grease film thickness with bearing operating time

This section investigates the evolution of the grease film thickness with running time at a constant bearing speed. Li-M-1 grease is used in all tests in this section and separate tests are performed at bearing speeds of 500 rpm, 1000 rpm, 2,000 rpm, and 2,800 rpm. Figure 5 shows the interference images taken at the start of the test and after 1, 2, and 8 h

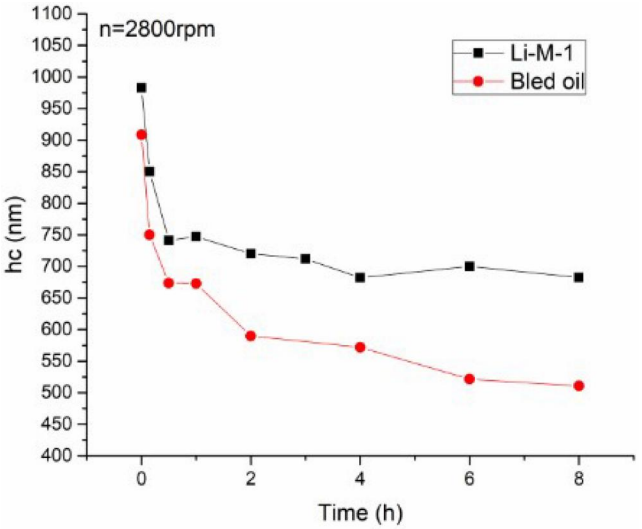


Figure 6. Measured central film thickness versus bearing running time for Li-M-1 and its bled oil at 2,800 rpm. Polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

of running at a constant speed of 1,000 rpm. The images show that the inlet meniscus distance decreases and the oil reservoir around the contact shrinks with increasing test time, which may be expected to lead to a decrease in central film thickness.

Figure 6 shows the average value of the measured central film thickness for Li-M-1 grease and its bled oil in a similar test conducted at 2,800 rpm. It is seen that the grease central film thickness decayed rapidly with running time in the first 30 mins of the test, from the initial value of about 1,000 nm at the start of the test to about 750 nm at 30 mins. The decay then continued but at a much slower rate until the film reached a steady state value of approximately 700 nm after around 4 h; it then remained relatively constant until the end of the 8h test. The bled oil film thickness was also seen to decay with running time; the decay was again most rapid in the first 30 mins, falling from about 900 nm initially to 650 nm at 30 mins; but unlike in the case of grease the film continued to decay throughout the test falling to about 500 nm after 8 h.

There are three potential reasons for the observed reduction of film thickness with running time: shear degradation, thermal effects and starvation. Since the set-up measures both the inner ring speed and the cage assembly speed, it is possible to deduce that the bearing is running close to pure rolling conditions and, therefore, thermal effects caused by friction in the contacts are likely to be relatively small.

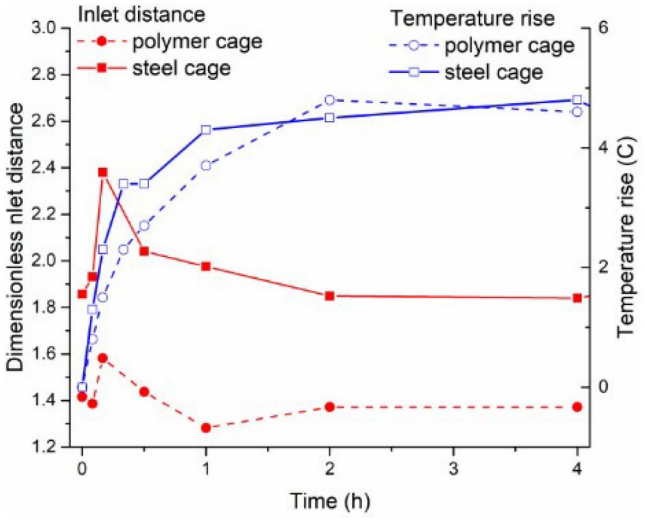


Figure 7. Measured central film thickness versus bearing running time with a standard polymer cage and a steel cage. Li-M-1 grease, $n = 2{,}800$ rpm, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

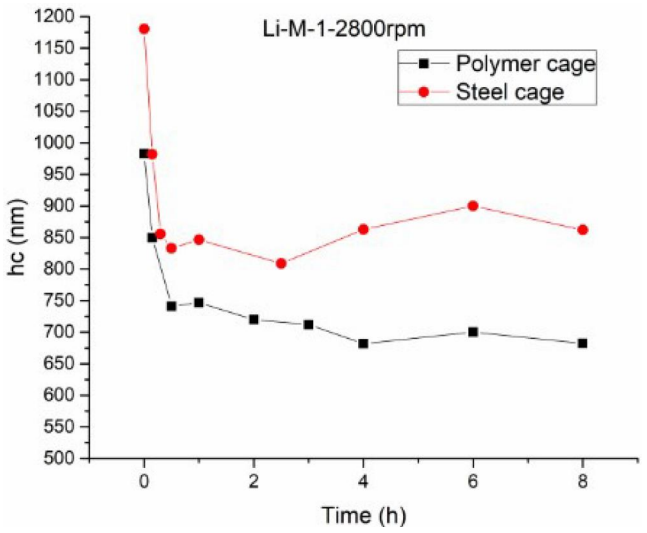


Figure 8. Measured inlet distance and temperature rise at the outer ring with polymer cage and steel cage. Li-M-1 grease, $n = 2{,}800$ rpm, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

Temperature measurements at the outer ring indicate a maximum temperature rise of around 4 °C above the initial bearing temperature (see Fig. 8). This temperature rise cannot account for the relatively large reduction in film thickness with time of around 300 nm observed in the tests. The film reduction is therefore likely to be mainly caused by starvation as suggested by the results of Fig. 5 where the oil reservoir around the contact is seen to shrink with running time. The reduction in film is less severe in the case of grease than bled oil probably because grease can form relatively stiff ridges next to the running track which help to keep the bled oil close to the contact.

Figure 7 compares the central film thickness measured when the bearing was equipped with a riveted steel cage to that measured when a standard polymer cage was used. Tests are at the same conditions as Fig. 6, that is, Li-M-1 grease at 2,800 rpm. In both cases, the decay in the grease film thickness with running time described above was seen. However, when a steel cage was used, the final steady state film thickness was somewhat higher (around 850 nm) than that with polymer cage (around 700 nm). The measured inlet distance and temperature rise for each cage is shown in Fig. 8. The temperature rise for both cage types was very similar, so the observed differences in film thickness cannot be attributed to any temperature differences. However, the measured inlet distance for the steel cage is larger than that for the polymer cage at all times during the 8-h test. This suggests that the higher film thickness achieved with the steel cage was due to better lubricant supply to the contact provided by the steel cage in the present set-up. This may be caused by the relatively large gap between the ball and the steel cage in the center of the bearing causing less grease" scraping" locally, which in turn improves lubricant supply to the ball-ring contacts, as they are also located in the center of the bearing in the present case. This may explain why this effect was not observed in the axially loaded bearing in Cen and Lugt, (35) since in that case the location of the largest cage clearance gap and the center of the contacts do not coincide.

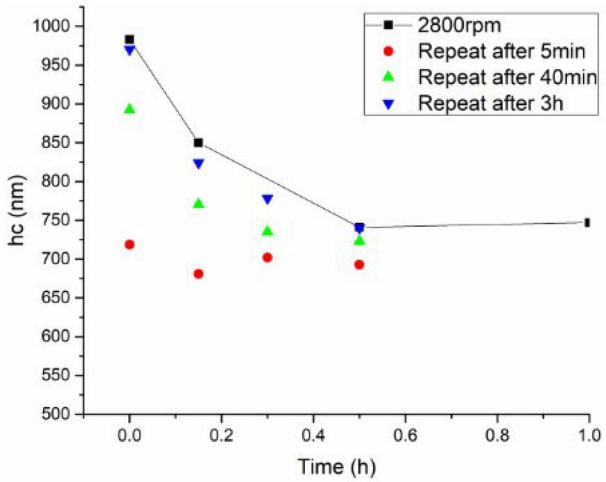


Figure 9. Measured film thickness versus running time during the start-stop experiment: Film during the initial 2 h test at 2,800 rpm (black squares), film after a 5 mins stop (red), film after a further 40 mins stop (green) and film after a further 3 h stop (blue triangles). Polymer cage, rotating speed $n = 2{,}800$ rpm, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

Effect of start-stop operation on grease film thickness

To investigate the effect of bearing start-stops on grease film thickness, the following sequence of tests was conducted: the bearing was first run at 2,800 rpm for 2 h; the rotation was then stopped for 5 mins, after which the bearing was again run at 2,800 rpm for 30 mins; the bearing was then stopped for 40 mins followed by another run at 2,800 rpm for 30 mins; finally, the bearing was stopped for 3 h and then run at 2,800 rpm for 30 mins. Film thickness was measured at regular intervals during each of the bearing runs. The film thickness results are shown in Fig. 9. During the initial 2 h run at 2,800 rpm the film decayed with time in a manner described earlier from around 1,000 nm to a steady state value of about 750 nm. After a short stop of 5 min, the initial film thickness measured at 2,800 rpm was close to 725 nm, that is, similar to that measured before the stop. A

similar trend was seen in tests described in Cen and Lugt (*40*) with a similar grease where no film recovery was seen with stop times shorter than 1 min. The fact that in the present tests no significant recovery was observed even after a somewhat longer time of 5 min may be partly due to the outer ring being flat leading to much lower osculation between ball and ring, which is unfavorable for contact replenishment. Notwithstanding this, the observed trends are very similar.

After a 40 min stop, the film thickness was seen to recover significantly to about 900 nm. This then decayed over 30 mins to about 750 nm, which is similar to the steady state value in the initial 2 h run. Finally, a long stop of three hours resulted in the full recovery of the film, with the measured values being very similar to those obtained in the initial run with the fresh grease fill and before any stops. The main reason for the observed film recovery behaviour is most likely due to the lubricant reservoir in the bearing track being replenished after the prolonged 40 mins and 3 h stops, whereas the short 5 mins stop is insufficient for such replenishment to take place. In addition, some film recovery is also likely due to the bearing temperature falling back closer to room temperature during the prolonged stop; although this is unlikely to be a major contribution as the bearing temperature rise during the initial 2-h run was only a few degrees.

Impact of bearing speed on film thickness

Figure 10 shows the measured central film thickness over an 8-h test at constant speeds of 1,000, 2,000, and 2,800 rpm with Li-M-1 grease. The initial central film thickness at the start of the test is different at different speeds as may be expected. However, the film thickness for all speeds eventually falls to the same steady state value of around 700 nm after a few hours of running time. This test was repeated with three additional grease types at 1,000 rpm and 2,800 rpm and the results are shown in Fig. 11. This range of tested greases covers different thickener types (Lithium, Lithium complex and Polyurea), synthetic and mineral oils and a range of oil viscosities. All greases show a similar behavior: after a few hours bearing running time the grease

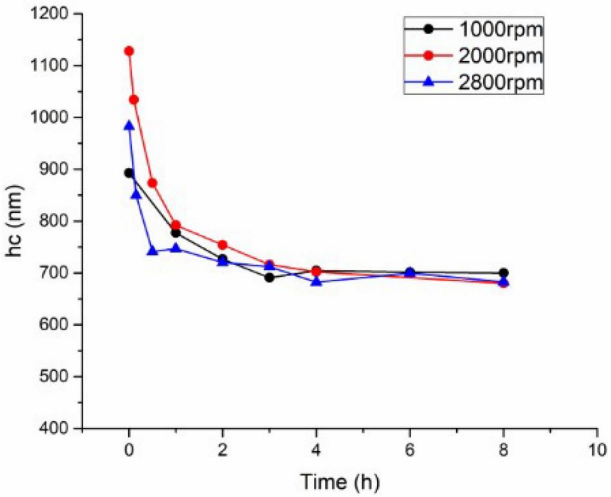


Figure 10. Measured central film thickness versus running time in an 8-h test at constant speed of 1,000, 2,000, and 2,800 rpm with Li-M-1 grease. Polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease supply: 2 g.

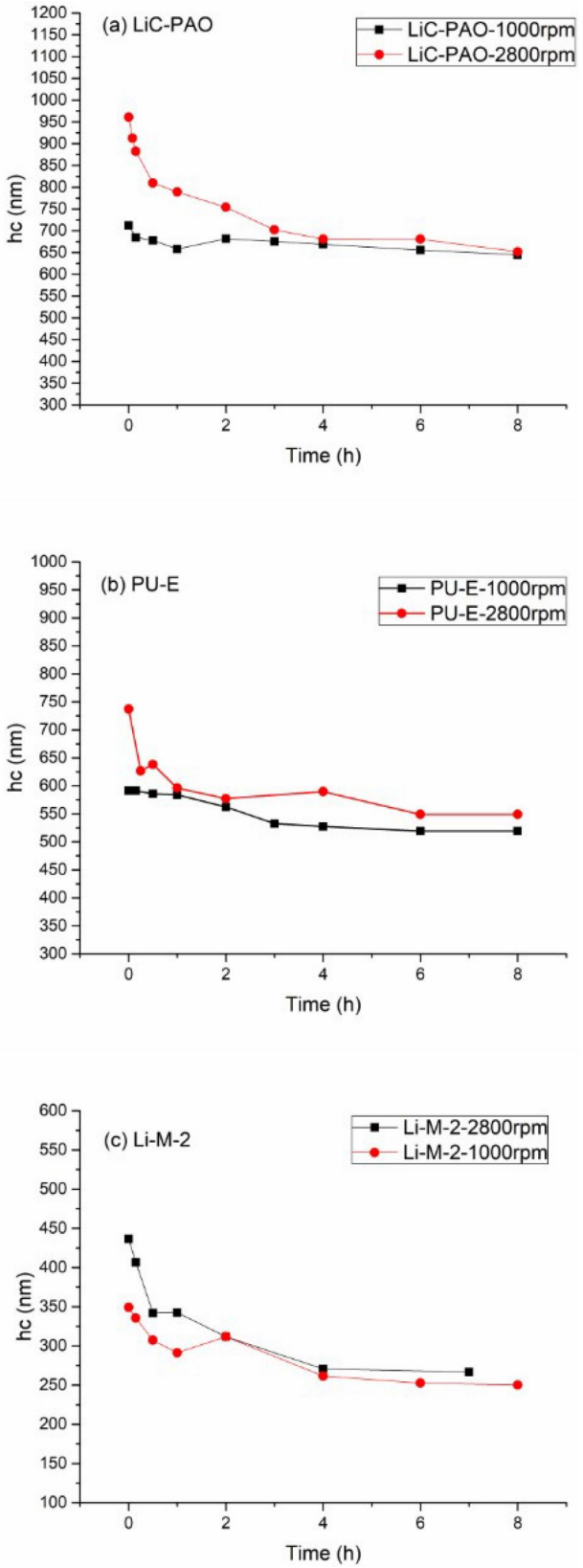


Figure 11. Measured central film thickness versus running time in an 8-h test at constant speeds of 1,000 and 2,800 rpm for (a) LiC-PAO; (b) PU-E; (c) Li-M-2 grease. Polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease fill: 2 g.

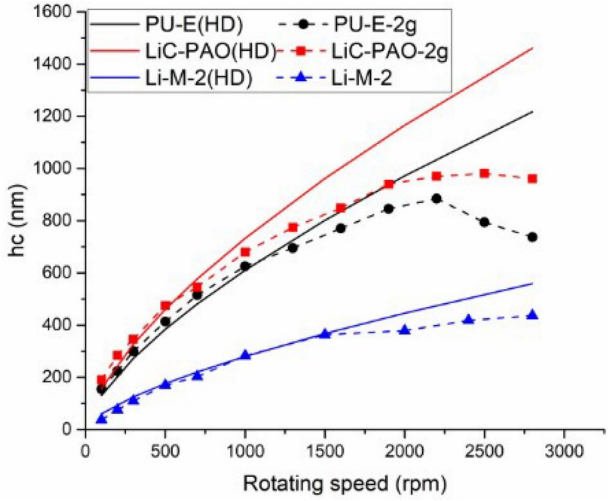


Figure 12. Measured central film thickness versus bearing speed for LiC-PAO, PU-E, and Li-M-2 grease compared to the theoretical film thickness predicted using Hamrock-Dowson equation. Polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease fill: 2 g.

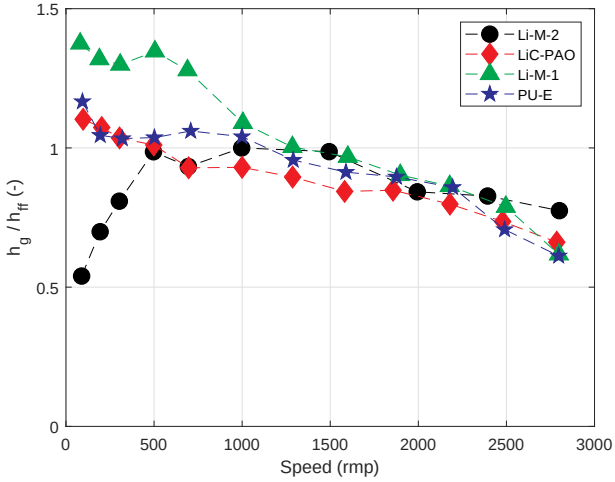


Figure 13. The ratio of measured grease central film thickness (h_g) to that predicted with EHL theory for fully flooded conditions (h_{ff}) plotted against the bearing rotational speed. Polymer cage, $p_H = 730$ MPa, $T_0 = 25\,^{\circ}$C, grease fill: 2 g.

film thickness for a given grease is independent of bearing speed. This steady state film thickness is different for different grease types and it seems to be strongly affected by the base oil viscosity, with lower viscosity resulting in lower steady state film thickness. For example, the final film thicknesses with Li-M-1 grease (98 cSt mineral oil and lithium thickener) and the LiC-PAO grease (100 cSt PAO oil and lithium complex thickener) are both in the region of 700 nm, whereas the final thickness with Li-M-2 grease (32 cSt mineral oil and lithium thickener) is as low as 250 nm. Figure 12 shows the film thickness measurements in tests with continuously increasing speed for the Li-M-2, LiC-PAO, and PU-E greases. The theoretical film thicknesses predicted using Hamrock and Dowson EHL equation for base oil properties are shown in each case for comparison. Similar results were already shown for the LiM-1 grease in Fig. 3. The lower viscosity base oil greases are shown to produce lower film thickness in these tests too. In addition, it is evident that starvation occurs earlier (at lower bearing speed) for the higher viscosity greases; for example, at 750 rpm for 100 cSt LiC-PAO grease versus 1,500 rpm for 32 cSt Li-M-2 grease.

To compare the starvation behavior of the greases, a dimensionless film thickness calculated as the ratio of measured film thickness to that predicted using Hamrock-Dowson equation using base oil properties alone (h_g/h_{ff}) is used. Figure 13 shows the relevant results. This shows that at low speeds, the grease film thickness is generally higher than that predicted with EHL theory for the base oils (this is the case for all tested greases other than the low visosity Li-M-2). Multiple previous studies have shown this to be the case in the single contact, and attributed the low-speed film enhancement to thickener effects (*18*,*20*); these results confirm that the same film enhancement at low speeds is seen in an actual bearing. As the speed increases, the dimensionless film thickness reduces and at a certain speed falls below the theoretically predicted one for all greases. It then continues to reduce as the speed increases further. The trend is indicative of increasing starvation with increasing bearing speed.

Discussion

This paper employs a newly developed optical bearing rig to explore the grease film behavior in a radially loaded rolling bearing. The method allows for direct observation of the ball-ring contact and direct measurements of the film thickness using optical interferometry. This direct film measurement approach is an improvement over previous studies, such as those based on capacitance, which largely employ pure axial load and rely on indirect film measurements averaged over all ball-ring contacts.

Onset of starvation in grease lubricated bearings

The results confirm that starvation occurs in grease lubricated bearings as speed increases, so that the film thickness is lower than that predicted using standard EHL theories for much of the bearing speed range. The exact operating conditions when starvation starts were dependent on grease properties; in particular, higher viscosity base oils led to the onset of starvation at lower bearing speeds. The onset of starvation was confirmed by the observation of the meniscus in the contact inlet. Occurrence of starvation with grease lubrication has long been recognized based on film measurements in single contact studies; the direct observations of the ball-ring contact presented here confirm that similar phenomena occur in real bearings.

The observed onset of starvation could not be prevented or postponed by increasing the amount of grease fill in the bearing. Assuming that the bulk grease is pushed sideways in the very early stages of operation, this suggests that, at least for the relatively short bearing running times (hours), grease far away from the running track does not contribute to the lubrication of the bearing. However, this grease may become relevant after much longer running times; the stored grease is subjected to constant stresses and creep due to

centrifugal forces that may eventually cause lumps of it to fall into the running track and thus temporarily reduce the starvation level. This is the most likely reason for the occasional increase in inlet distance, seen in Fig. 4, an effect that has also been observed by Zhou et al., (*41*) who measured occasional events in film thickness in combination with observed lumps falling off the cage.

Starvation was also observed in the equivalent tests with the bled oil obtained from one of the test greases. Once starvation occurs, the starved film thickness with grease was higher than that with its bled oil. This difference may be due to oil retention properties of the grease where the grease acts as a reservoir for oil and restricts oil side flow away from the track, thus restricting starvation.

It is difficult to compare the absolute values of the starved film thickness and the speeds at which starvation may occur reported in this study to previous studies, be it those employing the single contact set-ups or those employing capacitance measurements in real bearings under pure axial load. This paper reports film thickness in a ball-ring contact in an operating bearing under a radial load. This is obviously different from the single contact situation given the effects of bearing geometry and repetitive ball passage will have on grease distribution and supply to the contact inlet. The lubricant supply to the contact and the onset of starvation are also likely to be different from other bearing studies done under axial load, such as that in Cen and Lugt. (*40*) One of the important differences is that in the present case of a radially loaded bearing, the width of the ball-ring contact varies continuously during bearing operation as the ball load is changing, whereas, the ball-ring contact width is constant under pure axial load, which is likely to affect the replenishment of the grease in the contact since the contact width is known to play an important role in starved lubrication. (*35*,*42*)

It should also be noted that the use of the flat outer ring results in a much smaller width of the ball outer ring contact for a given bearing load than is the case for the real grooved outer ring. The inner ring contact is the same width in both cases; it is also larger than the corresponding outer ring contact in both cases. Assuming that grease present in and around the ball track once it leaves the inner ring contact partly contributes to the supply of the outer ring contact, in addition to the grease on the cage and the outer ring itself, the present set-up may potentially lead to an increased level of starvation in the outer ring contacts. This is because any grease present on the ball surface will have been pushed further away from the center of the ball track by the much wider inner ring contact before it reaches the outer ring contact so that it may be less able to contribute to the replenishment of the outer ring contact. It is difficult to say how significant this potential effect is in practice since this is only one potential grease supply mechanism and the exact kinematics of the balls are unknown; any spin for example may act to reduce or nullify any effects of this.

Steady-state film thickness after extended running time

The results show that when bearing is run at a constant speed for a prolonged period of time, a situation most often encountered in practice, the initially measured grease film thickness is higher than that expected with the base oil alone but then decreases rapidly as the bearing rotates. The decay is very fast in initial minutes of operation and continues at a somewhat slower rate for 1 to 2 h, after which a steady-state film thickness is reached. The bearing therefore spends most of its operation at this steady-state thickness, the value of which is lower than that predicted using the standard EHL theory with base oil properties. The relatively fast film decay, that is, the short churning phase, may be characteristic of the open type bearing used here, a sealed bearing may exhibit a somewhat longer churning phase.

Interestingly, the steady-state film thickness was observed to be largely independent of the bearing speed. This means that this steady-state film thickness does not follow the standard hydrodynamic theory. Its absolute value was different for different greases and was strongly influenced by the viscosity of the base oil and much less so by the type of the thickener. The reason for the apparent independence of the steady-state film thickness from speed is likely to be related to the complex kinematics of the lubricant supply to the contact in an operating bearing. Clearly, the fact that film is not increasing with speed shows that the lubricant supply is limited and the film is determined much more by the level of the lubricant supply in the inlet than any hydrodynamic effects. Conversely, the film thickness was lower with greases which had lower viscosity base oils, in line with hydrodynamic theory. This is despite the fact that lower base oil viscosities were seen to postpone the onset of starvation to higher speeds, suggesting that they improve the lubricant supply to the contact.

The independence of steady-state film thickness from bearing speed also implies that the lubricant supply to the ball-ring contact is not influenced by the bearing speed. This observation seemingly undermines some of the grease bleed theories of bearing lubrication, since grease bleed, and subsequently the oil supply to the contact, should increase with increasing speed due to higher grease shearing. One potential explanation for this apparent contradiction is that the oil is supplied to the contact primarily from a grease reservoir that is stationary and is therefore unaffected by bearing speed.

This apparent independence of long-term grease film thickness from bearing speed appears counterintuitive at first. However, the same independence of film from bearing speed was measured by Cen et al. (*16*) in an axially loaded deep-groove ball bearing using capacitance. A similar general decay of the initial film thickness and its subsequent independence from entrainment speed in a single contact was also described by Van Zoelen et al., (*25*) although they only considered oil lubrication and exclude the effects of contact replenishment. They provide an empirical equation that suggests that the oil film thickness is initially dependent on speed in line with the EHL theory, but with increasing operational time the film slowly decays to a value that is not affected by speed but is still dependent on base oil viscosity. The same general trend is seen here, perhaps indicating that the basic mechanisms of starvation are taking place in this

grease lubricated bearing although the exact details of the lubricant flows are much more complex here.

Transient effects due to start-stop operation

The results presented in this paper show that start-stop transient effects are important in the development of grease films in a bearing. Stopping the bearing for a sufficiently long time and then restarting leads to a recovery in the grease film thickness, from the long-term steady-state value described above, to one approaching the initial value that was recorded at the start of bearing operation. The effect was not seen for very short stop times of less than 5 mins, but was evident for stop times of 40 mins and 3 h. These results indicate that the supply of the lubricant to the contact is replenished during the pause in bearing rotation. With renewed bearing rotation, the same steady-state film is reached once again after some time. Therefore, unlike in oil lubricated bearings where start-stops are undesirable as they momentarily lead to the EHL film breakdown, start-stops may be beneficial in grease lubricated bearings by temporarily increasing the film thickness, even if the effect is relatively short-lived. Similar trends in film increase after a stop in bearing rotation were recently reported by Cen and Lugt (*40*) who used a capacitance method to measure grease films in an axially loaded ball bearing. In their study the stopping time needed for film recovery was much shorter than here; however, these discrepancies in detail are not surprising given the significant differences in the two test set-ups, particularly the continuously variable contact width under radial load and the relatively narrow outer ring contact employed here.

Effect of cage design on grease film thickness

The grease film thickness measured when a steel cage was installed in the bearing was higher than that when a standard polymer cage was used under the same conditions. This was the case for both the initial film thickness and the steady state-film thickness, that is, that established after the churning phase was over as previously described. These results are in apparent contrast to those of Damiens et al. (*43*) who used a single contact set-up with a segment of a cage mounted on the ball and found that the film thickness was the highest for the smallest cage clearance. They attributed this to scraping and redistribution of the grease under the cage which would reduce the level of starvation. Out of the two cages used here, it is the polymer cage that has a much smaller clearance so would be expected to produce a higher film based on Damiens et al. theory. However, the present results are based on a full bearing and therefore include many additional practical factors that were not accounted for in the previous single contact studies so some differences are not unexpected. In Cen and Lugt, (*35*) the grease film thickness was measured with the same two cage types in a ball bearing under axial load using a capacitance method and no significant difference was found between the two. The present study employs a radial load and a different bearing geometry so again some differences are to be expected. The exact reason for the improved film thickness with steel cage in the current set-up is not clear but may be related to potential improvement in replenishment of the outer ring contacts by the large cage clearance gap which converges toward the center of the ball, as is the case for the steel cage, thus pushing the grease into the outer ring contacts which are also located in the center of the bearing. The cage clearance gap is constant in the case of the polymer cage so this affect would not be present. The polymer cage also has advantages of being lighter and not producing any metallic debris. The apparent film thickness improvement seen with the steel cage may also disappear in the case of a combined load or pure axial load where the EHL contacts are no longer in the center of the bearing, that is, the center of the contact and the location where the gap between the cage and the ball is the largest no longer coincide. In any case, the present results show that the cage design has an important role to play in bearing grease lubrication and consequently, that it may be possible to improve bearing lubrication via optimization of the cage.

Implications for prediction of grease film thickness in rolling bearings

In practice, film thickness in grease lubricated bearings is calculated using the standard EHL equations for fully flooded conditions and assuming the lubricant properties to be those of the base oil alone. Present results confirm that this approach is not appropriate. Based on single contact studies, it is already well established that at low speeds many greases build thicker films than their base oils due to the film enhancement by the thickener. The same effect is seen here in a full bearing; this is clearly not accounted for by the standard EHL theory. Another striking effect observed here that is not in any way described with the standard EHL theory is the decay in film with time to eventually reach a steady value which appears largely independent of bearing speed. Finally, the film thickness measurements under start-stop operation also show that transient effects are important in grease lubricated bearings. None of these phenomena are accounted for by the EHL theory, so clearly using the standard EHL equations to predict grease film thickness in a bearing is likely to lead to erroneous results. It is hoped that the insights presented here can form the basis for the development of better models for prediction of film thickness in grease lubricated bearings.

Conclusions

In this study, we used a recently developed model bearing set-up, referred to as the Sapphire Bearing Rig (SBR) (as fully described in Chennaoui et al. (*39*)) to directly observe and measure the grease film thickness in a ball-ring contact in a model radially-loaded ball bearing using optical interferometry. To the best of our knowledge, this is the first time the grease films in a radially loaded bearing were directly measured. The approach offers obvious advantages over single contact studies and other methods of measuring grease

films in real bearings, such as capacitance, which measure average film thickness indirectly over multiple contacts and are generally limited to axial bearing load only. The optical method employed here is particularly suitable for grease studies as it allows for the flow of lubricant around the contact to be observed, which then makes it possible to correlate the parameters such as the inlet meniscus distance to the level of starvation and the measured film thickness inside the ball-ring contact. The results reveal the following:

- Lubricant starvation occurs in grease-lubricated bearings as the bearing speed is increased.
- The bearing speed at which the starvation first occurs is governed by the grease properties; in particular, the speed is lower for higher base oil viscosities. Starvation could not be prevented or postponed by increasing the amount of grease fill in the bearing.
- When the bearing is run for a prolonged time at a constant speed, the initial grease film thickness is generally higher than that predicted by the EHL theory using base oil properties; This initial film decays with running time and eventually reaches a steady-state film thickness that is significantly lower than that predicted by the standard EHL theory.
- This steady-state film thickness was seen to be largely independent of bearing speed. This observation was surprising but is in line with previously reported capacitance-based film measurements in real axially loaded ball bearings. (*16*)
- The long-term steady-state film thickness was higher with greases with higher base oil viscosities, regardless of the thickener type. This is despite the fact that higher base oil viscosity led to an earlier onset of starvation.
- Start-stop operation of a grease lubricated bearing appears beneficial in terms of film recovery: For a sufficiently long stopping time, the film thickness upon restart recovers from the steady-state value described above, back to the higher value found at the very start of the bearing operation. No recovery was seen for very short stopping time of 5 mins, partial recovery after 40 mins and full recovery after 3 h.
- Film thicknesses measured with a riveted steel cage and a standard polymer cage were different indicating that cage design plays an important role in bearing lubrication.
- These observations show that the standard EHL theory cannot reliably predict the film thickness in grease lubricated bearings. The insights presented here can help in the development of new tools for better prediction of grease film thickness in rolling bearings.

Disclosure statement

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