EHL/mixed transition of fully formulated environmentally acceptable gear oils

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ABSTRACT

Understanding the ability of fully formulated industrial oils to protect the contacting surfaces requires techniques revealing the effect of tribofilm growth on the EHL film. This was achieved by employing sliding/rolling smooth contact in a ball-on-disc test for two industrial EAL gear oils and two industrial mineral oils. Friction and electrical contact resistance (ECR) of three stribeck stages at different running-in periods were studied besides measuring the tribofilm thickness evolutions by Spacer Layer Interferometry Method (SLIM). In the end, a wear test was performed and wear scars were measured. Results showed that lower pressure-viscosity coefficient of EALs does not necessarily lead to high metal-metal contact, and their thin tribofilm serves to keep the friction low in mixed and boundary regimes.

1. Introduction

Lubricant formulation is currently changing due to environmental considerations. This change also includes additives formulation and concentration [1–4]. Since the early 1970s, usage of Environmentally Acceptable Lubricants (EALs) in industry increased and different sectors started investing in the development of EALs, e.g. development of EAL hydraulic fluids for forestry operations in Germany in 1980s [4]. Reduction of oil reserves, increasing oil prices and environmental considerations to avoid the negative effect of oil spillage have attracted attention to new and less environmentally harmful lubricants [5].

Having high biodegradability, low toxicity and good thermal and oxidative stability ester oils are the most common EAL used in industrial lubricants [6–9]. The lower friction coefficient of ester-based lubricants in full film and EHL regime is observed in several studies [10–13]. Martins et al. compared two EAL oils with a mineral oil in terms of power loss, oil temperature rise and wear prevention in FZG test rig [14]. The results show that the tested ester oils give lower friction, temperature and wear [14].

José A. Brandão et al. studied the stribeck curves of some mineral and ester-based lubricants and illustrate that while the friction coefficient of ester lubricants is lower at full film regime, it is higher under mixed and boundary lubrication [11]. The same result is found by Riegler and Kassfeldt testing the environmentally acceptable hydraulic fluids [15]. José A. Brandão et al. suggest that poorer boundary performance is due to the undesirable additives used in the lubricant formulation. EALs should have limitations for toxicity and biodegradability, so it is not possible to use some additives [11]. This assumption can be true; however, it requires further investigation to find out whether it is a result of the base oil, the additives, or both.

Compared to a mineral oil with a certain viscosity, ester oils have lower pressure-viscosity coefficient which, according to EHL theory, leads to lower film thickness [16]. However, ester-based lubricants show higher thermal conductivity and heat capacity. The film thickness in the EHL contact is very small, so the heat capacity would not be very effective in the performance. Yet, the thermal conductivity can play an important role in the friction [17] and film thickness by faster heat transfer to the contacting bodies.

In addition, the higher viscosity index of esters excludes the usage of VI improvers. The ester linkage is stronger than the C–C bond in mineral oils resulting in better viscosity-temperature behavior and oxidation stability [18]. Also, the high polarity of ester molecules makes them sticky to the surface which improves the boundary lubrication [6] though it can compete with the additives and negatively influence their function. These parameters are also beneficial in film formation especially under high slip and high load conditions [3].

In the worm and hypoid gears, the lubricant usually fails to prevent the asperity contacts, so the lubrication is performed under the mixed
lubrication. The same condition happens for the cylindrical and bevel gears operating at low speed and high loads \[19\]. Concerning wear, pitting, and scuffing, the EHL film formation ability hinders the transition from EHL to mixed lubrication and consequently less asperity contact and lower abrasive and adhesive wear. AW and EP additives play an important role in wear, scuffing and pitting, but they are functioning in the boundary and near boundary lubrication \[20\].

Being fast, compact, and safe, ball-on-disc test is considered to be an important tool for the development of new lubricants. It provides good repeatability and control over a wide range of parameters like speed, temperature, load \[21\]. There are some problems with the simulation of gear contact with a twin-disc machine which are also present in ball-on-disc machine. These problems can be listed as below \[22, 23\]:

- The different surface topography and orientation between test specimens compared to the modern gears
- Different film formation process which is continues in ball-on-disc, while a new film is formed in every tooth engagement in the gears
- Lower dynamic effects in MTM compared to gears

Despite the mentioned limitations, ball-on-disc test is successfully used in fundamental researches on lubricant testing. Björling et al. used friction maps developed earlier in Ref. \[24\] to make a correlation between the friction measured by ball-on-disc test and FZG test rig \[25\]. The results showed that ball-on-disc device is able to well simulate the working condition of FZG gears. It is observed that using FZG and ball-on-disc devices leads to the same results in ranking the friction performance of different oils. So, it can be said that using ball-on disc method is a reasonable way to understand the performance of gear oils before testing with full gear test equipment. This will help to select a proper lubricant for a known condition.

With the development of new traction machines, the experimental studies on the lubrication regime \[11, 26\] and EHL film formation of the oils \[27\] became very popular. Optical interferometry has been vastly used to measure the film EHL film thickness under different lubrication conditions \[28–30\]. In this method, the requirement for a transparent disc is the main disadvantage causing differences with the contacting metallic surfaces that are common in machine elements \[31\]. The electrical contact resistance (ECR) technique, though unsuccessful to measure the film thickness \[32\], has been used for qualitative analysis of contact in real components. In gear teeth contacts, the running-in leads to changes in the lubrication regime, surface roughness alteration and tribofilm formation between two metallic surfaces. These changes are influential in the performance of lubricant regarding efficiency \[33\], or failure \[34\]. The ECR technique is successfully used to evaluate the tribofilm formation \[35–38\], the mapping of lubrication regime and the influence of running-in on the lubrication regimes \[26\], influence of the texture on the transition between lubrication regimes \[39\] and checking the contact lubrication conditions along the line of action of a simulated gear pair \[22, 40\].

Rizvi discusses the effect of the chemical structure of different base oils including esters and shows the correlations between tribological characteristics and the structures. Depending on the type of alcohol and fatty acid used in the production of the ester, the final properties can be optimized for a special application. Influential parameters on EHL film formation ability like density, viscosity index, pressure-viscosity coefficient, heat conductivity and capacity, polarity are enhanced by manipulating the chemical composition and molecular structure of esters. The final EHL performance of the lubricant is the superposition of all these parameters \[41\].

In order to examine the performance of a lubricant, it should be tested by a practical machine under real conditions. These tests are costly and time-intense. Therefore, it is critically important to have laboratory tests with scientifically analyzed results which are applicable to the real components. Using a fully formulated lubricant in these prior laboratory tests is necessary to correlate the results to the real
component tests. There are scientific papers using fully formulated lubricants to investigate the wear and pitting of engine oils [42], friction [11,14] and tribofilm generation of gear oils [43]. However, there is a literature gap in the interaction of EHL and tribofilm formation for the fully formulated lubricants. Also, the mechanism by which EALs can compete with mineral oils is not examined clearly. This paper focuses on the EHL-to-mixed transition in three stages of running-in, the effect of tribofilm thickness on the EHL film and mild wear protection of the industrial EALs formulated for the gears. These objectives are achieved by studying stribeck curves in a rolling/sliding contact of a ball-on-disc machine, together with ECR technique and Space Layer Imaging (SLIM) technique. The experimental strategy used in this paper provides a scientific experimental means to evaluate the fully formulated lubricants. Investigating the tribological performance of the EALs would help to understand the advantage and disadvantages of the available oils in the market.

2. Experiment detail

2.1. Experimental rig

All the ball-on-disc tests were performed using a mini-traction machine (MTM) developed by PCS instruments. As it can be seen in Fig. 1, a ball and a disc are in rolling/sliding contact. The tilted ball shaft minimizes the spin, and a load cell attached between the ball shaft and the instrument body measures the friction force. A series of tests can be performed by fully automated control overload, velocity, temperature and the test duration. The ball and disc are driven independently in order to achieve a wide range of lubricant entrainment speed and slide to roll ratio. Lubricant entrainment speed, sliding speed and slide-to-roll ratio (SRR) are expressed by the Eqs. (1)–(3):

\[ U_r = \frac{U_d + U_s}{2} \]  
\[ U_s = U_d - U_s \]
\[ SRR = \frac{U_s}{U_r} \]  

where \( U_d \) and \( U_s \) are respectively the disc and ball circumferential velocities in the contact point, \( U_r \) is the entrainment velocity and \( U_s \) is the sliding velocity.

Lubricant and pot temperatures are measured by two thermometers placed respectively in the lubricant, and the pot wall. These temperatures are automatically adjusted by a heater, and a circulating fluid provided by an external heater/cooler equipment.

The electrical contact resistance between the ball and disc can be measured by coupling an electrical bridge between them. As it is shown in Fig. 1, a balance resistor of 10 kΩ is mounted in series with the ball and disc, and an electrical potential of 15 mV is applied. If the contacting bodies are fully separated, the ECR corresponding to the electrical resistance of the contact is 100%, and if they are in contact, the ECR is 0. The amounts between 0 and 100% indicate the partial separation by EHL film or tribofilm.

The tribofilm formed on the ball surface can be observed using a technique called Spacer Layer Imaging (SLIM). In this technique, the ball is loaded against a spacer layer of transparent silicon dioxide coated with a thin, semireflective layer of chromium. When white light is shone into the contact, a colored interference image of the contact is formed and recorded by the camera. The evolution of this interferometry images gives the tribofilm thickness changes during each step [44]. The tribofilm thickness was calculated according to the technique shown in Appendix A used in Ref. [45].

2.2. Test specimen

The ball (19.05 mm diameter) and disc specimens were both AISI 52100 steel with hardness 750–770 VPN polished to the surface roughness indicated in Table 1, where \( R_a \) is average roughness of profile, and \( R_z \) is mean peak to valley height of roughness profile. Fresh ball and test were used for testing each oil, and they were cleaned by immersion in toluene and isopropanol in an ultrasonic bath for 10 min.

2.3. Tested lubricants

In order to get an overview of the environmental acceptable lubricants, two different EALs with the same viscosity class from different companies were selected. Both oils meet the US EPA requirements for “Environmentally Acceptable Lubricants”. Additionally, two mineral oils were selected: one gear oil with the same viscosity class, and an engine oil with the similar 40 °C kinematic viscosity that is practically used in ships for gear lubrication. All the oils except M1 that is an engine oil, comply with the DIN 51517 part 3 (CLP) standard. The oils specifications can be found in Table 2:

2.4. Experimental procedure

Four types of test were performed for each oil: stribeck test, traction test, tribofilm formation test, and high-sliding wear test.

Stribeck test represents the variation of coefficient of friction as a function of stribeck parameter (viscosity*speed/load). In this experiment, load and viscosity are constant, so the stribeck results are presented as a coefficient of friction versus entrainment speed while keeping SRR constant. Traction curve shows the variation of friction coefficient by increasing sliding speed while having a constant entrainment speed. As the sliding speed is equal to SRR multiplied by entrainment speed, so the curve is usually shown as friction versus SRR. The traction curve can give an explanation about the shear properties of the lubricant [46], and gives an estimation of the frictional behavior of the oils along the line of action of a gear pair [25].

As it is illustrated in Fig. 2, the set of stribeck and traction tests were repeated three times: the first set on the fresh samples, the second set after 1 h of rolling/sliding running, and the third set after an additional 2 h of running. During each set of stribeck and traction test, friction and ECR were recorded, and tribofilm thickness was checked by SLIM technique before and after each test. The evolution of friction, ECR, and tribofilm thickness indicates the ability of each oil to separate the contacting surfaces by either EHL film or boundary tribofilm. The main idea of this test plan is to evaluate and compare the EHL/mixed transition of oils and study the effect of tribofilm thickness on this transition.

The parameters for each of the tests are presented in Table 3:

After finishing three sets of stribeck and traction tests, a wear test inspired by Ref. [47] was performed. The advantage of this wear test is minimization of the contact geometry changes, and providing a slide-roll ratio greater than two that leads to high sliding speed and low entrainment speed. The surface roughness of worn and new discs was checked by an Alicona InfiniteFocus G5 optical profilometry system from Optimax.

3. Results and discussion

3.1. Evaluation of the fresh samples

Fig. 3 (a) and (b) respectively show the stribeck and traction curves of the lubricants performed on the fresh samples. As it is performed on
the new samples, and the tribofilm thickness after the test was negligible, it can be supposed that the friction and ECR are not yet affected by the tribofilm.

Fig. 3 (a) shows Strubeck curves performed at 5% SRR. Decreasing the entrainment speed from 3900 mm/s, at first the friction increases. Then at an entrainment speed between 1200 and 1600 mm/s (depending on the oil), it reaches the maximum, and then it decreases till a minimum appears at an entrainment speed around 50 mm/s, and then it rises. This curve is different from an ideal Strubeck curve in which the transition from (elasto)hydrodynamic to mixed lubrication goes with a sharp increase of friction coefficient till it reaches the maximum at the transition to the boundary lubrication regime. Lafountain et al. observed the same trend as presented in Fig. 3 (a), and described that it is due to the heating effect. The ideal Strubeck is considered to be isothermal, while a non-isothermal Strubeck curve can be affected by the shear heating. By calculating the viscosity changes due to temperature rise in the contact, the isothermal model can be corrected. The thermal correction results in a matched theoretical and experimental Strubeck curve [48].

The thermal effects can vary the Strubeck curve to the extent that the friction coefficient does not represent the EHL/mixed transition. Hansen et al. illustrate that the friction coefficient does not signify the initial asperity contacts, and consequently the transition to the mixed lubrication [26]. From ECR changes presented in Fig. 3 (a), it can be seen that the transition to the mixed lubrication takes place at an entrainment

| Table 2 |
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| Measured lubricants properties. |
| | Kin. Vis. @40 °C (mm²/s) | Kin. Vis. @100 °C (mm²/s) | ρ @15 °C (kg/m³) | VI | Comment |
| Method | ASTM D445 | ASTM D445 | EN ISO 12185 | ASTM D 2270 |
| M1 | 127.60 | 13.83 | 908 | 105 | Mineral engine oil |
| M2 | 142.50 | 14.24 | 888 | 97 | Mineral gear oil |
| EAL1 | 147.50 | 18.84 | 972 | 145 | Synthetic oil, EAL |
| EAL2 | 150.37 | 18.44 | 929 | 137 | Synthetic oil, EAL |

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| Table 3 |
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| Conditions of each test. |
| | Traction test | Strubeck test | Tribofilm formation test | Wear test |
| SRR | 0–1.9 | 0.05 | 0.50 | 100 |
| Entrainment Speed (mm/s) | 700 | 10–3900 | 150 | 70 |
| Maximum Hertzian pressure (GPa) | 1.25 | 1.25 | 1.11 | 1.25 |
| Temperature °C | 40 | 40 | 100 | 40 |
| Duration | – | – | 1 h before 2nd stribeck, and 2 h before 3rd stribeck | – |

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speed around 500 mm/s for all the oils, and a minimum in the friction is not visible, whereas interestingly the friction coefficient is decreasing in the mixed lubrication regime. However, for the case of transition to the boundary lubrication, the friction stays constant or increases that signifies the onset of severe asperity contacts.

Johnson and Tevaarwerk proposes three friction regions in the traction curves of a lubricant: 1) a linear region in which the fluid behavior is Newtonian, 2) a non-linear (isothermal) region in which the shear-thinning happens and the maximum friction is bound to the limiting shear stress, and 3) a thermal region in which thermal softening behavior is dominant in the friction [46]. Björling discussed friction mapping in Ref. [49], and based on works in Refs. [46, 50] extends the friction regions into a 3D friction map. Fig. 4 illustrates a similar friction map for the oil M2 as an example. The border between non-linear and thermal region is set at the point in which the friction drops with either increase or decrease of entrainment speed. With increasing the entrainment speed in a constant SRR, depending on the SRR, the lubricant experiences different transitions. In low SRR, by increasing the $U_b$, the lubrication is first affected by the shear-thinning in the non-linear region, then by the limiting shear stress, and at the end by the thermal region. However, for the high SRR or near zero SRR, the Striebeck curve will remain respectively in the thermal region or linear region [50]. If a specific lubrication condition is considered near to the border of the non-linear and thermal region, by keeping the SRR and increasing the entrainment speed, it will fall into the thermal region and consequently lower friction coefficient. This is what happens in the friction of tested oils when the speed increases from 1200 to 3900 mm/s.

In Fig. 3 (a), the friction coefficient is decreasing by reducing the entrainment speed even in the mixed lubrication regime of all the oils. Therefore, with the smooth contacts specified in Table 1, the effect of asperity contact on the friction coefficient is considerably less than the effect of fluid shear stress drop.

In Fig. 3 (a), the ECR changes show the EHL/mixed and mixed/ boundary transition of the oils. Although it is difficult to exactly find the EHL/mixed transition point, it is clear that the ECR of EAL2 is the highest. The better protection provided by EAL2 is approved by the 3D image of wear scar in section 3.3, Fig. 9. Finding an explanation for the higher ECR of EAL2 calls for comparing effective parameters on EHL film thickness. Jackson and Rowe defined a parameter named Lubricant Parameter (LP) which is the product of pressure-viscosity coefficient ($\alpha$) and dynamic viscosity ($\eta$) at a specific temperature [51]. This parameter is used as an indication of EHL film formation ability of the oils in line contact and is employed in Ref. [43]. Coseausu et al. used Hamrock and Dowson’s formula [52] and redefined the lubricant parameter as $LP = \alpha^{0.53}\eta^{0.67}$ for elliptical contacts [53]. In order to find the LP parameter for the oils, pressure-viscosity coefficient and dynamic viscosity were estimated using a method presented in Ref. [11]. This method is based on an experimental equation developed in Ref. [16], and it is used for modeling the traction curves of gear oils [12] and approximating the EHL film thickness [43, 54]. Calculated values of pressure-viscosity coefficient ($\alpha$) and dynamic viscosity ($\eta$) and LP are given in Table 4.

In order to approve the estimated values of the pressure-viscosity coefficient, the specific EHL film thickness was calculated using Hamrock and Dowson’s formula [52]. The specific film thickness is plotted in Fig. B.12 in Appendix B. It is clear that the transition to mixed lubrication happens at the lambda ratio equal to 3, and the transition to boundary lubrication is visible at the lambda ratio equal to 1. It is noticeable to say that the calculated EHL film thickness does not consider the thermal effect and inlet shear heating, so that is why the Lambda value (and consequently EHL film thickness) of EAL2 is lower than the others.

From Table 4, EAL2 has the lowest LP value which is not consistent with its higher ECR and EHL film thickness. The temperature rise within the EHL film was not considered in LP value, so the higher ECR of EAL2 can be attributed to the lower generated heat by friction, the higher thermal conductivity, higher heat capacity, higher viscosity index of ester-based oils compared to mineral oils. The oil entrapped in the contact is so small, so heat capacity should not be significant compared to the thermal conductivity. In addition, the higher polarity of ester-based oils can be another parameter enhancing EHL film thickness. EAL1 with a similar oil type as EAL2 shows lower ECR than EAL2 and M2. This indicates that the chemical composition and molecular design play an important role. Although the LP value did not signify the EHL film of EAL2 and EAL1, it well confirmed the lower ECR of M1 oil in comparison with M2.

The traction curves in Fig. 3 (b) clearly show lower friction of EALs compared to the mineral oils. Björling et al. show that the friction performance of oils can be ranked by their traction curves, and it gives the same result as FZG device [25]. Based on this, the power loss by using EAL1 and EAL2 is lower than the mineral ones.

Brandão et al. show that the lubricants can be classified according to the similarity of their EHL traction curves [12]. This interesting feature is visible in the present tested lubricants, and the lubricants can be classified into two groups: M1, M2 with high friction, and EAL1, EAL2 with low friction. In each group, the friction, VI and pressure-viscosity coefficient are almost similar, while the kinematic viscosity and lubricant parameters (LP) are different. This indicates that friction in the EHL regime is more dependent on the pressure-viscosity coefficient than the dynamic, kinematic viscosity or the lubricant parameter.

In the traction curves (Fig. 3 (b)), except for the case of M1 oil, the ECR of the other oils is not sensitive to the SRR and remains around 100%, providing EHL lubrication. So, just for the oil M1 the viscosity drop with SRR is significant and causes EHL/mixed lubrication transition. Hansen et al. observe that the ECR is sensitive to SRR for the fresh rough samples (Ra of ball = 208 mm and disc = 7 mm), but it becomes insensitive to SRR (in the range of 0–0.5) after running-in period [26]. Therefore, the insensitivity of ECR to SRR in Fig. 3 (b) is due to the smooth surface of the samples compared to the EHL film thickness at 700 mm/s entrainment speed.

The data points of traction test were acquired by increasing from

![Fig. 4. 3D map of friction for the oil M2.](image-url)
SRR = 0, and each data point is the average of data for 20 s. By increasing the SRR, if the asperity contact is very small, it can be polished during 20 s. So, a very short running-in happens during each data point measurement. However, if the oil film thickness is small enough, the asperity contacts are more severe and the change of the lubrication regime can be detected by ECR technique. This is the case of M1 in Fig. 3 (b) that the transition to mixed lubrication is evident in the high SRRs.

Kumar and Khonsar show that the lower thermal conductivity leads to appearance of limiting shear stress at lower SRR [17]. From Fig. 3 (b), it is noticeable that the limiting shear stress of the EALs happens at higher SRRs compared to the case of mineral oils. Also, the drop of their traction is less (0.016 for EALs vs 0.010 for mineral oils). Based on the literature [6], it is also expected that EALs, mostly consisted of ester molecules, show better shear stability. Based on what was discussed on the Stribeck curve variations and the lubrication regions, it can be said that the latest the limiting shear stress is reached, the less the oil is affected by the temperature rise. The mean oil film temperature rise is equal to Ref. [55]:

$$\Delta T_{oil} = \frac{h}{8K_{oil}} \tau U_s$$  \hspace{1cm} (4)

where $K_{oil}$ is the oil thermal conductivity, $h$ is the film thickness, and the generated heat is equal to mean shear stress ($\tau$) multiplied by the sliding speed ($U_s$). From traction curves of the oils in Fig. 3 (b), for the case of EAL2, it has the lowest friction coefficient, so it generates the lowest amount of heat. Based on the Eq. (4), the EALs provide less temperature rise by providing firstly lower friction, and secondly higher thermal conductivity. This lower temperature rise results in less viscosity drop in

Fig. 5. Stribeck curves of the oils. The “1st” is for the fresh samples, the “2nd” after 1 h rubbing, and “3rd” after 2 h more additional rubbing.

Fig. 6. Growth of tribofilm and its effect on the solid-solid contact.
high loads and high shear contacts. High viscosity index of ester-based oils makes them free from viscosity index improver leading to better shear stability and film formation ability [56]. In conclusion, higher thermal conductivity, higher viscosity index and lower heat generation help them to keep their viscosity less affected by the heat generation in the contact.

3.2. Running-in effect

After the 1st stribeck and traction tests, a 1-h rubbing at high temperature (100 °C) was carried out to accelerate the tribofilm formation and increase asperity contacts. Then the 2nd stribeck and traction tests were performed. After that, another additional 2 h of rubbing at high temperature was performed, followed by the 3rd stribeck and traction tests (Fig. 2). For the case of traction curves, a small amount of decrease in friction was observed and the ECR of all oils was around 100% showing an EHL lubrication.

Regarding the stribeck curves shown in Fig. 5, the friction in the EHL region decreased for all the oils. However, in the mixed region, the reactions were different from one oil to another. In all the cases, the ECR was increased compared to the 1st stribeck. The ECR can increase by flattening of high asperity summits, or by the formation of tribofilm that both reduce metal-metal contact [38]. As the ECR variations were different from one oil to another, it is good to first discuss each case.

For the case of M1, the following points can be remarked based on Figs. 5 and 8:

a) Friction: In the mixed and boundary regime, friction was increased. Also, in the 2nd and 3rd stribeck curves, the friction rise is started in higher \( U_e \) which indicates weakened EHL film due to the shift of boundary lubrication to higher \( U_e \). It is noticeable to say that in the EHL regime, the friction is a bit decreased due to the flattening of the high summits.

b) ECR: The ECR has the lowest amount for the 1st stribeck. In the 2nd stribeck, the ECR is increased and then decreased in the 3rd stribeck.
c) Tribofilm: From Fig. 8, the Tribofilm is growing in every stage, and the thickest is found before the 3rd stribeck. Considering the point a) and c), it can be said that the growth of triofilm in oil M1 leads to higher friction in mixed and boundary regime. It seems that the available AW additives in this oil enhanced the protection against wear by scarifying the effect of FM additives. Dawczyn et al. investigated the tribofilm formation of ZDDP and observed the same friction rise, and correlated it to the increased roughness [57]. This behavior can be seen in the other triofilms, and it is not just the characteristics of ZDDP films [58].

From point b) the increased ECR in the 2nd stribeck seems logical as signifies the formation of the triofilm [38] in Fig. 8. However, the decrease of ECR in the 3rd stribeck was not expected, because the thickest triofilm is found before this stage (point c), and it had to lead to the lowest metal-metal contact. A thick triofilm can also change the friction, so the higher friction in mixed and boundary lubrication is affected by growing triofilm.

Taylor and Spikes observed the effect of triofilm formation on the friction in the mixed regime, and argue that the increased friction in the mixed regime and its extension to higher $U_e$ is not just due to an increase in the roughness. They proposed the following possible mechanisms by which this change happens [59]:

- Starvation due to
  - Lower wettability of tribofilm
  - Inlet blocking by the triofilm
- Localized oils films around the triofilm by lower EHL film due to:
  - Lower viscosity
  - Lower pressure-viscosity coefficient
- Modification of the inlet geometry by the triofilm
- Slip at the boundary of triofilm/oil

In addition, if we consider the starvation due to the accumulation of the long-chain molecule hydrocarbons in the inlet, the weakened EHL film seems reasonable. Now, the ECR decrease in the 3rd stribeck curve of M1 can be explained. In the mixed and boundary lubrication, the load is partly supported by the EHL film and solid-solid contact. If a triofilm is formed between the surfaces, part of the solid contact is supported by the triofilm, but still, there is a degree of asperity contacts. By adding the triofilm load carrying capacity to the equation presented in Ref. [60], the load carrying shares can be formulated as below:

$$F_T = F_{EHL} + F_C + F_H$$

where $F_T$ is the total load, $F_{EHL}$ is the load carried by the fluid film, $F_C$ is the load carried by the asperities which are not covered by the triofilm, and $F_H$ is the load carried by the triofilm. For the case of M1 oil, the inhibiting effect of grown triofilm weakens the primary EHL film, so the separating EHL film is not generated very well. When the $F_{EHL}$ is decreased, the asperity contacts are increased to compensate for the EHL load share. Furthermore, when there is no EHL film, the triofilm can be penetrated easier and the probability of metallic contact will increase. Fig. 6 shows a schematic of the effect of triofilm growth on the metal-metal contact. In Fig. 6 (a), there is no triofilm available which is the case of 1st stribeck curve. By a small growth of the triofilm (Fig. 6 (b)), part of asperity contacts are covered, so the ECR is increased (lower metal-metal contact). For the 3rd stribeck, the thick triofilm damages the EHL film, so consequently the asperities that were separated by the EHL film come to the contact (Fig. 6 (c)).

The blocking character of triofilm formed in oil M1 can be amplified by its combination with the long-chain molecule hydrocarbons found as impurities in the oil. Though Dawczyn et al. show that the roughness is the most important parameter influencing the friction increase [57], it seems that other parameters are also important, as there are several pieces of research indicating the inhibiting effect of patch like structure in ZDDP triofilms [58,59,61,62]. From this case, it can be concluded that there is an optimum triofilm thickness being able to separate the surfaces properly, higher or lower thickness leads to higher metal-metal contact.

Discussing the other oils in Fig. 5, friction in mixed lubrication was increased for EAL1 showing an extension of mixed and boundary regimes to higher entrainment speed, same as the feature observed for the case of M1, though it is with smaller amplitude. This is due to the triofilm growth in EAL1; however, it is not too large to significantly block the EHL film, so it did not result in high metallic contact (decrease in the ECR). For the case of M2 and EAL2, ECR was raised and the friction was slightly decreased in the EHL regime. These changes are mainly due to high summits flattening, and partial surface coverage by the triofilm, and maybe activation of the friction modifiers. Nevertheless, no meaningful change of mixed and boundary friction is visible in any one of these two oils. Although some small amount of triofilm is visible in the SLIM images of M2, the friction curve does not show the extension of mixed lubrication to higher $U_e$, so the triofilm in these oils are not damaging the EHL film.

### 3.3. Ranking the oils regarding surface protection

As was mentioned in section 3.2, the ECR is a result of the separation
between ball and disc by EHL or tribofilm. For a specific oil, if the ECR of a stribeck curve is increased after a running-in, it means that high asperity summits are flattened resulting in a stronger EHL film, or the tribofilm thickness is grown [38]. For the case of 1st stribeck test, the ECR and EHL film of oils was discussed in section 3.1. Regarding the 2nd and 3rd stribeck, it is necessary to check Figs. 3 (b), 5, 7 and 8 together in order to separate the effect of tribofilm thickness and EHL on the ECR.

In the 2nd and 3rd stribeck test, performed after a 1 h and 2 h of running-in, the tribofilm thickness in M1 and EAL1 is noticeably high (Fig. 8). So for these two oils, the increase of ECR in 2nd and 3rd stribeck (Fig. 5) is originated from the tribofilm growth. For the EAL2, the tribofilm is negligible in all stribeck stages (Fig. 8), so the high ECR of EAL2 in all the stribeck tests is mainly attributed to its high ability to form and maintain EHL film. For this oil, the ECR changes in Fig. 5 are mostly from the high summits flattening. For the oil M2, the tribofilm thickness is not as thin as EAL2, and not as thick as M1 and EAL1, so the tribofilm and asperity flattening both are effective in ECR changes of this oil.

Comparing EAL2 and M2 in Fig. 7(a) using Eq. (5), $F_{EHL}$ of M2 is lower to the extent that it is not able to be compensated by the tribofilm, so its $F_C$ is bigger and consequently, it has lower ECR. Compared to EAL2, EAL1 had lower ECR in 1st stribeck (Fig. 3(a)), but its tribofilm is

|       | $R_a$ (nm) | $R_z$ (nm) |
|-------|------------|------------|
| M1    | 64         | 415        |
| M2    | 62         | 482        |
| EAL1  | 49         | 393        |
| EAL2  | 30         | 198        |

Table 5
Roughness parameters of the disc after the wear test.

Fig. 10. Roughness profile of the disc across the wear scar before and after finishing all the test stages.
able to compensate for the difference of $F_{\text{EHL}}$ that it had with EAL1. For the 3rd stribeck, all the oils except the M1 have almost similar ECR but the share of loads that are carried by each term of Eq. (5) is different for them.

After the 3rd stribeck curves, a high sliding wear test was performed to accelerate the wear. The 3D images of the worn surface of discs are shown in Fig. 9, and the surface roughness profiles before and after the tests can be observed in Fig. 10. The EAL2 has a very smooth disc surface with few scratches similar to the new sample. The surface height is not grown showing that there is negligible tribofilm on its surface which is in agreement with the slim images of Fig. 8. So it approves that for the oils EAL2, the surfaces are protected mostly by the EHL film. In Fig. 9, M2 has more scratches compared to the EAL2, and from Table 5, its higher roughness clarifies that the surface lubricated with M2 has more damages than the surfaces lubricated with EAL2. For the case of M1, the thick tribofilm does not allow to see a clear image of the damages to the surface, nevertheless, deep scratches comparable to those of M2 are visible. These scratches and a thin tribofilm are present also in the case of EAL1 making it comparable to the case of M2.

4. Conclusion

The objective of this work was to examine the EHL film formation ability of fully formulated EALs compared to mineral ones. Besides, the interaction between EHL film and tribofilm was investigated. A ball and disc test equipment provided sliding/rolling contact between two smooth surfaces loaded with 1.25 GPa maximum Hertzian pressure. Friction and ECR variations in the Stribeck and traction curves were analyzed together with an in-situ measurement of tribofilm thickness. Two industrial EALs formulated for gears and two fully formulated mineral oils were evaluated regarding their EHL/mixed transition. The overall conclusions are as follow:

- The excessive growth of tribofilm damages the EHL film and additionally can result in higher metal-metal contact. So, a thin tribofilm is an advantage for the components working in the EHL regime.
- The damage of EHL film by a thick tribofilm is not just due to the roughened surface. It is mostly related to the inlet blocking by the tribofilm.
- Comparing two tested EALs, the one with lower LP value showed less metal-metal contact. This indicates the importance of chemical composition and molecular design of base oil and additive in EHL film formation.
- Compared to the tested mineral oils, EALs showed better or similar EHL film formation ability though having lower LP value. The lower LP value of EALs can be compensated by:
  - Less generated heat
  - Their higher VI makes them free from VI improver
  - Their better thermal conductivity
  - Higher polarity
- In the EHL regime, friction is more dependent on the pressure-viscosity coefficient than the dynamic, kinematic viscosity or the lubricant parameter. Regarding this point, the lower pressure-viscosity coefficient of EALs is an advantage that causes lower friction.
- Friction does not signify the transition between EHL to mixed lubrication regime.
- ECR technique together with the friction and SLIM technique provide a good understanding of the interaction of tribofilm and EHL film.

A fully formulated industrial oil is composed of several chemical compositions for the base oil and additive. Studying EHL formation ability of these oils calls for considering the interaction of different compositions. Physical properties like viscosity and pressure-viscosity coefficient, though useful for rough approximations, are not enough to estimate the EHL/mixed transition especially when the tribofilm is formed between the surfaces. The results and approach in this study provides an insight into the interaction of tribofilm with EHL film and approves that industrial EALs can be substituted for mineral oils, especially for the components working in the EHL regime.

Author contribution section

Reza Bayat: Conceptualization, Methodology, Software, Validation, Formal analysis, Investigation, Resources, Data Curation, Writing - Original Draft, Writing - Review & Editing, Visualization.

Arto Lehtovaara: Conceptualization, Writing - Review & Editing, Supervision, Project administration.

Declaration of competing interest

The authors declare no potential conflicts of interest with respect to the research, authorship and/or publication of this article, and we have fully respected the research ethics principles.

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Appendix A. Tribofilm thickness measurement
Appendix B. Specific film thickness

The specific film thickness is calculated using Eqs. (B.6-B.8):

\[ \Lambda = \frac{h_{\text{min}}}{\sigma_{\text{rms}}} \]  

(B.6)

\[ h_{\text{min}} = 3.63 R_{x} \left( \frac{U_{e} \eta_{0}}{E^{*}} \right)^{0.68} \left( a E^{*} \right)^{0.40} \left( F \sqrt{E_{r}^{*} R_{x}} \right)^{-0.073} (1 - e^{-0.68}) \]  

(B.7)

\[ \sigma_{\text{rms}} = \sqrt{R_{q1}^2 + R_{q2}^2} \]  

(B.8)

where \( R_{x} \) is the radius of curvature in the x-direction (m), \( U_{e} \) is the entrainment speed (m/s), \( \eta_{0} \) is the dynamic viscosity of the lubricant (Pa.s), \( E^{*} \) is the reduced Young's Modulus (Pa), \( a \) is the pressure-viscosity coefficient (Pa \(^{-1}\)), \( F \) is the applied load (N), \( k = 1 \). \( R_{q1} \) and \( R_{q2} \) are the Root-Mean-Square roughness of disc and ball respectively amounting 34.08 (nm) and 50.95 (nm).

The specific film thickness is shown in Fig. B.12:

Fig. A.11. Thickness measurement of the tribofilm for oil M1 before the 2nd stribeck.

Fig. B.12. Specific film thickness and ECR vs entrainment speed for the 1st stribeck test.

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