Influence of the Pressure-Viscosity Behavior of Different Base Oils on the Formation of Lubricating Films in Tribological Contacts

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Abstract.

Hydraulic fluids ensure power transmission and lubrication in hydraulic systems. The performance of hydraulic fluids primarily depends on their physical properties. The development of environmentally friendly hydraulic fluids has become more important. Depending on their chemical structure, these fluids have different physical properties, which influence the formation of load-bearing lubricating films in tribological contacts. In this paper, the influence of pressure-viscosity behavior on the formation of lubricating films in tribological contacts will be discussed by means of tribometric tests. For this purpose, hydraulic oils with different base oils are subjected to two test procedures. The tests are done for mineral oil based hydraulic oils (HLP) and biohydraulic oils based on synthetic esters (HEES). To determine the hydraulic fluids friction behavior, they are tested using a disc-on-disc tribometer. This test allows investigating the formation of load-bearing lubricating films of the oils. A comparison of the lubricant film formation at increasing rotational speeds is made. This allows evaluating the friction conditions and the influence of the oil's pressure-viscosity behavior based on Stribeck curves. To obtain a detailed statement on the formation of the lubricating film, tests are also carried out on a high frequency reciprocating rig (HFRR). The HFRR is an oscillating friction wear test, which provides information about the lubricity of fuels and lubricants.

Keywords. Biohydraulic oil, friction coefficient, load-lubricating film, pressure-viscosity behavior, synthetic ester, tribometer test, high-frequency reciprocating rig.

1. INTRODUCTION

The use of bio-based hydraulic oils is becoming more and more important due to increasing regulations regarding environmental protection. In some areas of application in which hydraulic systems are used, the utilization of biohydraulic oils is already regulated by law. For example, in forestry [1] or in shipping applications [2]. Other areas are expected to be affected by similar requirements in the future.

In addition to environmental benefits by using biohydraulic oils, the compatibility of the fluids used in the respective hydraulic system is of paramount importance for a safe and efficient operation of the respective machine. Furthermore, the physical properties of the hydraulic fluids used influence the operating behavior of the hydraulic components.

The properties and chemical structure of the base oil are decisive for the physical properties of hydraulic oils. The chemical composition of biohydraulic base oils differs from that of mineral oil based hydraulic fluids. Most hydraulic components, such as displacement pumps, have been developed with regard to the physical properties of mineral oil based fluids. A different behavior of biohydraulic oils can cause an insufficient formation of load-bearing lubricating films in tribological contacts for example in displacement pumps. In the worst case, this results in damage due to disproportionate wear. Experimental investigations in previous research works [3], detected higher leakage temperatures at the hydraulic pump when using biohydraulic oil instead of mineral oil, which indicates a less developed lubricating film in the tribological contacts.

In this paper, the influence of the pressure-viscosity behavior of different hydraulic oils on the formation of lubricating films in tribological contacts is analyzed. For this purpose, the performance of a synthetic ester (HEES) and a mineral oil based hydraulic oil (HLP) regarding the pressure-viscosity behavior is discussed by carrying out two test methods. The hydraulic oils are firstly subjected to a tribometer test in a disc-on-disc tribometer in which the conditions of real tribological contacts of hydraulic displacement units can be reproduced. Secondly, the wear and the lubricant film thickness is analyzed in a high frequency reciprocating rig (HFRR) at high contact pressures.

In previous research work dealing with the behavior of the tribological properties of biodegradable hydraulic oils, less practice-relevant contact conditions have been considered. The research of Castro [4] focused on the formation of tribological lubricating films in a pin-on-disc tribometer. Murakami [5] considered the tribological behavior of bio based hydraulic fluids using a four ball test. In this paper, contact mechanisms close to application are simulated with the help of a disc-on-disc tribometer, which increases the significance of the results for real powertrains of hydraulic displacement units.

The paper is structured as follows; first, the state of the art is described. For that, the fundamentals of lubrication film building will be described. Based on this, the influence of the different chemical structures of the base oils on the pressure-viscosity behavior is discussed. The two test methods are then presented and explained. The different test cycles are also described, as are the test parameters. The measurement results are then presented and analyzed. Particular attention is paid to analyzing the differences between the different hydraulic oils in terms of friction reduction and lubricant film formation. Furthermore, it will be discussed to which extend the results can be transferred to application-related hydraulic components and what influence they have on the operation of the respective components. Finally, the paper is concluded with a summary and an outlook on future research in this area.

2. STATE OF THE ART

Hydraulic fluids are important design elements in hydraulic systems. The specific properties of hydraulic fluids have a significant influence on the functionality, operational safety and environmental compatibility of hydraulic systems. In addition to the hydraulic power transmission, hydraulic fluids are subject to further function-maintaining tasks. One of the most important secondary tasks of hydraulic fluids is the reduction of friction and wear of components moving relative to each other. This is particularly important in components for power generation such as hydraulic displacement units. These contain highly stressed tribological contacts in which the surfaces are separated from each other by the hydraulic fluid.

The operating conditions and the loads imposed have a great influence on the friction-reducing behavior of hydraulic fluids. Viscosity is the most important physical property of hydraulic oils in terms of achieving a hydrodynamic or elastohydrodynamic lubricating (EHD) condition. It is a measure of the internal friction of the fluid.

According to Newton's law of friction (2.1), the shear stress τ formed between fluid layers is the product of the dynamic viscosity η and the shear rate S. [6]

$$\tau = \eta \cdot S \qquad (2.1)$$

The viscosity of hydraulic oils generally depends on various parameters, such as the shear gradient, temperature and pressure. For mineral oils and synthetic oils of comparable molecular masses, a Newtonian behavior is assumed. In this paper, the focus is on the viscosity-pressure behavior of hydraulic oils with different base oils.

2.1. Pressure-viscosity behavior

The oil's viscosity dependence on pressure can be explained based on its chemical structure. The branching and hydrocarbon chains, which move relative to each other, generate a resistance when sliding on each other, which represents the internal friction. If these chains are pressed closer together by applied pressure, a greater force is necessary to move them relative to each other. This means that the internal friction increases as the pressure rises. This applies to both mineral and synthetic oils. [7]

The first and still frequently used approach to describe the pressure-dependent viscosity is the exponential approach of Barus (2.2) [8]. The viscosity η_0 is obtained at a defined temperature under atmospheric pressure (p=0). α_0 is the pressure-viscosity coefficient, which will be discussed in more detail in the next chapter.

$$\eta(p) = \eta_0 \cdot \exp\left(\alpha_\eta \cdot p\right) \qquad (2.2)$$

Due to decreasing accuracy at very high pressures beyond approximately 5,000 bar, the Barus equation has been extended. In some approaches by various authors, the exponent of the Barus equation has been supplemented by potency and polynomial approaches [9]. A frequently used equation is the Roelands equation [10], which is often used to describe elastohydrodynamic contacts. In this paper, the Barus approach is considered because of its conciseness in the evaluations.

2.1.1. Pressure-viscosity coefficient

The pressure-viscosity coefficient described before is a fluid specific characteristic value and is mainly influenced by the composition (content of paraffin, naphthenic hydrocarbons and aromatics) of the base fluid. The added additives have a minor influence [11]. Since measured values for the coefficient are often not available, it is usually approximated. Pressure-viscosity coefficient estimates for ester-based and mineral oil-based are shown in **Table 2.1**. In general, the pressure-viscosity coefficient of mineral oils is higher than that of ester-based oils. This behavior was confirmed within the framework of the special research area 442 (SFB 442/German Research Foundation) [12] with the aid of a high-pressure viscometer.

The pressure-viscosity coefficient of a lubricant increases with increasing degree of branching and number of cyclic structures, presence of aromatics and increasing length of side chain branches. Naphthenic and paraffin-based mineral oils have extended branched chemical structures so that they interlock with increasing pressure, thus increasing the internal friction in the fluid. Ester molecules are composed of a flatter structure and have less extended branches, which causes a lower dependence on pressure. [13]

Table 2.1. Pressure-viscosity coefficient [14]

Oil type	Value	Unit
Paraffin-based mineral oils	$1.5 \ldots 2.4 \cdot 10^{-8}$	m^2/N
Naphten-based mineral oils	$2.5 \ldots 3.5 \cdot 10^{-8}$	m^2/N
Ester oils	$1.5 \ldots 2.0 \cdot 10^{-8}$	m^2/N

2.1.2. Previous investigations

Previous research also focused on the influence of the pressure-viscosity behavior of different base oils on the formation of lubricating films in tribological contacts. In [4] tests using a pin-on-disc tribometer showed an influence of the pressure-viscosity coefficient on the formation of the lubricating film. It was found that HEES formed a lubricating film of lower height compared to HLP, thus increasing the wear of the contact partners. [15] evaluated the wear of a hydraulic pump operated with the bio-based hydraulic oils HEES and HEPR compared to mineral oil. The best performance was achieved when tested with mineral oil, while a loss in efficiency could be identified when operating with HEES.

In this paper, the previous research work will be extended by different test procedures, which are presented in the following chapter.

3. TEST FLUIDS AND METHODS

In the following chapter, the test fluids and methods considered in this paper are presented, regarding setup, test properties and test evaluation.

3.1. Test Fluids

The test fluids of the investigations are a mineral oil based hydraulic oil (HLP) [16] and a biohydraulic oil based on a synthetic ester (HEES) [17]. The biohydraulic oil fulfils the

requirements of DIN 16807 [18]. The mineral oil tested contains zinc and is made of a paraffinic base oil. Some properties important for the investigations of the hydraulic oils are shown in Table **3.1**. Both hydraulic oils are of viscosity class ISO VG 49 according to DIN ISO 3448 [19].

Table 3.1. Fluid properties

Property	Mineral oil	Biohydraulic oil
Viscosity class [19]	46	46
Viscosity index (VI) [20]	105	190
Density (at 15 °C) [21]	875 kg/m³	920 kg/m³

3.2. Disc-on-disc tribometer

For tribological investigations, a disc-on-disc tribometer has been used. The tribometer represents an abstracted simulation of the contact mechanics of cylinder block/swash plate friction pairing. The comparability of the contact conditions of the tribometer used with those of hydraulic displacement units has already been investigated and confirmed in previous research work [22].

3.2.1. Setup and functions

Figure 3.1 shows a schematic drawing of the disc-on-disc tribometer. The type of tribometer test stand is also known as Siebel-Kehl tribometer [23]. The test rig consists of two test discs, stator and rotor, which are pressed onto each other. The corresponding pressure is generated by a hydraulic cylinder by initiating the compression force F_N. The rotor is driven by an electric motor at a rotational speed n and the torque M is transmitted to the stator. The corresponding torque is recorded at the stator with a torque sensor. The contact pair is operated in an oil bath whereby the contact surface is continuously supplied with oil. During the test, the oil bath temperature is regulated to a steady value. The tribometer was also used for investigations made in [24]. On optimization of the tribometer with regard to the avoidance of hysteresis during friction measurement was carried out within the work of [25].

The contact pairing consists of a hard-soft contact. The rotor is made of a 31CrMoV9 alloyed nitride steel. The stator is made of a non-ferrous metal with the designation CuZn40Al2Mn2Si. All counterparts have been lapped to a roughness of Ra 0.1.

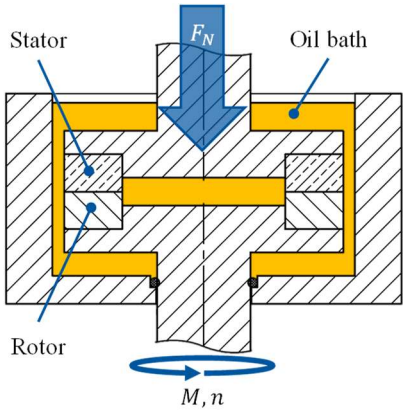

Figure 3.1. Disc-on-disc tribometer

The tribological measurement took place at an oil temperature of 60 °C, which was kept constant. Further technical parameters of the used test bench can be found in Table 3.2. The average contact diameter is 65 mm. The contact area is 1,021 mm² at a contact pressure of 1 MPa. During the test process, the rotational speed of 4,000 rpm is lowered until it comes to a standstill. The speed is reduced by 1 rpm per second. This makes it possible to reproduce a Stribeck curve of the test runs. Before the tests are carried out, the test specimens are run in at 1,000 rpm with a constant load of 1 MPa and 60 °C.

Table 3.2. Technical properties disc-on disc-tribometer

Parameter	Value	Unit
Mean contact diameter	65	mm
Area of contact	1,021	mm²
Friction track width	5	mm
Surface pressure	1	MPa
Rotational speed	0 to 4,000	1/min

3.2.2. Test evaluation

For evaluation, the frictional torque is recorded over the circumferential speed. This allows a Stribeck curve to be created and the frictional condition to be plotted as a function of speed. By plotting the friction coefficient, a statement can be made about the lubricity of the hydraulic oil.

3.3. High frequency reciprocating test (HFRR)

The test procedure using the high frequency reciprocating rig (HFRR) test is largely used to evaluate the lubrication properties of diesel fuels. The test is carried out according to the specifications of DIN EN ISO 12156-1 [26]. Due to the similar chemical structure of hydraulic oils and diesel fuels, the test method is used in this paper to identify the lubricity of hydraulic oils. The results obtained are analyzed with regard to the suitability of the test procedure according to [26] for evaluating the lubricating properties of hydraulic oils.

3.3.1. Setup and functions

Fig. 2 shows the test setup of the HFRR. An oscillating test ball with frequency f is pressed onto a stationary test plate under force F_G. The contact takes place in a temperature-controlled fluid bath. During HFRR testing, the machine's drive forces the ball to perform a sinusoidal movement.

The applied force creates a contact surface in the contact between the ball and the plate. Within this contact, a mixed friction condition occurs during the operation of the test rig. This means that solid contact occurs with simultaneous liquid friction. The proportion of the predominantly occurring friction condition depends on the lubricity of the respective lubricant. Among other effects, the formation of a load-bearing lubricant film is the decisive factor for the pressure viscosity behavior of the lubricant under investigation.

With the aid of the hertzian pressure theory, the contact pressure of the ball-on-disc contact can be calculated under steady-state contact conditions [27] The normal load induced by an attached 200 g weight creates a circular contact surface with a diameter of 67 µm. With (3.1), the maximum pressure of the ball disc contact can be calculated [6]. The force F_G applied to the contact amounts to 2.04 N. The two contact bodies are made of the steel ISO-683-17-100Cr6 [28]. The material-dependent characteristic values thus amount to a Poisson´s ratio ν of 0.3 and a modulus of elasticity E of 210,000 N/mm² [29]. This results in a maximum hertzian pressure of 835.68 N/mm². [30]

$$p_{max} = \frac{1}{\pi} \cdot \left[\frac{1.5 \cdot F_G}{r^2} \cdot \left(\frac{E}{1-\nu^2} \right)^2 \right]^{\frac{1}{3}} \qquad (3.1)$$

Due to the high loads on the contact, boundary friction occurs in the contact surface of the lubricated contact pair. This means that part of the load is carried by the lubricating fluid film, which is generated by hydrodynamic mechanisms. The other part is carried by solid contact of the test specimens.

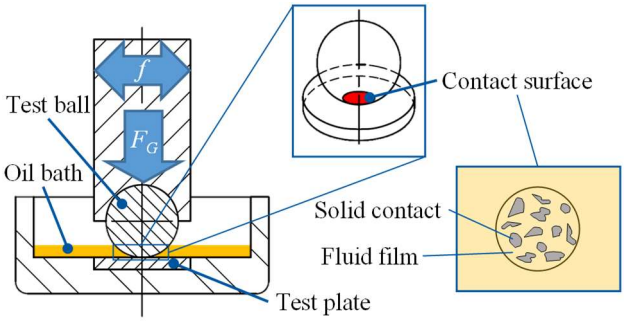

Figure 3.2. HFRR test setup

The test properties are chosen regarding DIN EN ISO 12156-1 (Table 3.3). The temperature of the oil bath is regulated to 60 °C during the testing process. The test fluid volume is 2 ml per fluid. The frequency of oscillation is 50 Hz, which results in a sliding speed of 0.05 m/s with a stroke length of 1 mm. The test procedure is carried out for 75 min for each run.

Table 3.3. Test setup HFRR

Parameter	Value	Unit
Oil temperature	60	°C
Fluid volume	2	mL
Frequency	50	Hz
Load	200	g
Stroke length	1	mm
Testing time	75	min

To analyze the lubricity of the fluid, the wear of the HFRR samples is determined after the tests have been carried out. For this purpose, the planar dimensions of the wear scar on the sample are recorded with an optical microscope. Then, the wear scar diameter (WSD) is calculated by averaging the length of the scar in sliding direction (y) and perpendicular (x) to it (3.2).

$$WSD = \frac{x+y}{2} \qquad (3.2)$$

4. TESTING RESULTS

In the following, the results of the tests executed are presented and discussed.

4.1. Disc-on-disc tribometer

4.1.1. Stribeck curves

To analyze the lubricity and the formation of a load-bearing lubricant film, the coefficient of friction μ is plotted against the rotational speed n, resulting in a so-called Stribeck curve. The coefficient of friction μ is calculated by dividing the frictional force F_R and the normal force F_N (4.1).

$$\mu = \frac{F_R}{F_N} \qquad (4.1)$$

Figure **4.1** shows the Stribeck curves of test runs in the disc-on-disc tribometer with HLP and HEES. The tests are recorded with a contact pressure of 1 MPa and at a regulated temperature at 60 °C. The tests were repeated three times to for every oil. The Stribeck curves obtained for all measurements of the different oils show in common the same trend. For this reason, two measurements are plotted as examples in Figure **4.1**.

The static friction (n≈0) when using HEES is significantly lower than when using HLP. This can be explained by the comparatively high polarity of the ester-based oil, which allows achieving a friction-reducing effect compared to HLP, especially in the boundary friction range. [31] When the speed is increased, the coefficient of friction drops significantly with HLP. With HEES, it drops less intensely. Thus, from 650 rpm, the coefficient of friction of HLP drops below that of HEES. From 2,750 rpm, both oils reach almost identical values again. All measurements taken show an abrupt reduction in coefficient of friction at 2,750 rpm, which makes it seem plausible that this is caused by factors related to the test

stand. In general, the Stribeck curve of HEES shows a much less pronounced trend than the curve of HLP. A notch point can be seen with the HEES from 2,300 rpm, but the coefficient of friction rises much more slowly (0.07 to 0.09) than with HLP (0.03 to 0.09). Which means that the fluid friction is much less developed by using HEES. The small difference between the coefficient of static friction and the coefficient of friction at high rotational speeds suggests that the contact pairing is operated in the mixed friction range for most of the test procedure. This indicates that the resulting lubricating film is smaller than when using HEES.

A possible explanation for this behavior is the lower pressure-viscosity coefficient of HEES. Especially at the high stressed roughness peaks, this can lead to solid body contact due to lower lubricant film thickness, which leads to a correspondingly higher coefficient of friction. Ongoing research underline this statement. This relationship is to be further investigated by the HFRR tests.

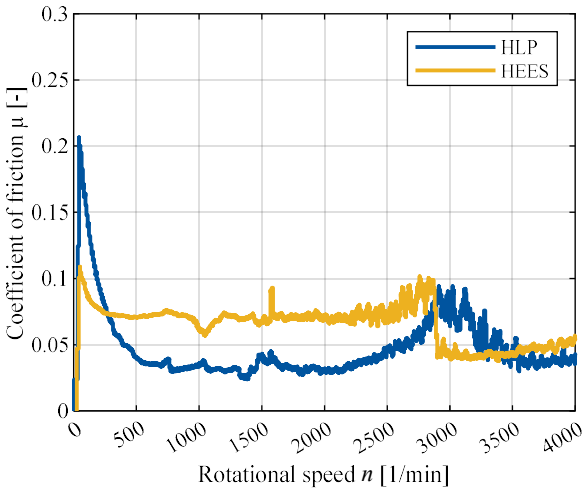

Figure 4.1. Stribeck curve at 1 MPa

4.2. High frequency reciprocating test (HFRR)

4.2.1. Wear scar diameter (WSD)

The WSD was determined over all tests carried out in accordance to [26]. For each fluid, five tests were performed. **Figure 4.2** shows the mean values of the wear calotte in x and y (sliding direction) coordinate and the respective standard deviations. The attached table shows the mean value for the WSD for the two hydraulic oils. The investigations show that in x and y direction, the wear area is larger when using HLP than when using HEES. On average, the wear area is about 15 % larger with HLP.

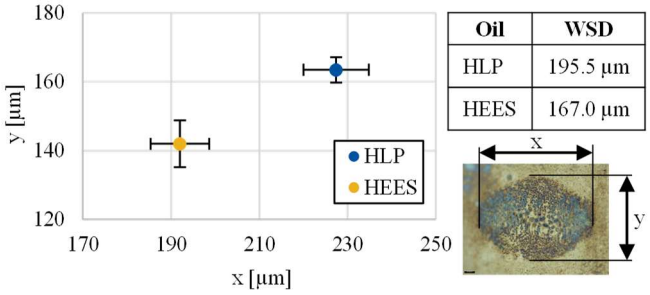

Figure 4.2. Wear scar diameter (WSD) with standard deviations

4.2.2. Wear examination

Following the test evaluation according to [26], the test specimens were examined under a 3D light microscope. For this purpose, the wear calotte was scanned 3-dimensionally and then the curvature was unwound and displayed as a flat surface. This makes it possible to display and analyze the surface profile of the test specimens. Figure **4.3** shows the surface profile of an HFRR test ball carried out with HLP and HEES. The geometry was recorded over a length of 130 µm in sliding direction. This is the area of greatest wear. In the adjacent areas, the wear is significantly less and mainly discoloration can be seen. Measurements were taken with a width of 20 µm and the mean value was calculated.

The contour of the resulting dome is similar when using both oils. An elliptical wear pattern is created with the long side in direction of the sliding movement. Three light-colored stripes can be seen in the center of the wear dome of the HEES test ball. All test specimens considered showed a similar pattern of wear. When observing this area under the 3D microscope, a significantly increased material removal could be detected in this area. This indicates an insufficiently formed lubricating film, which leads to solid body contact and increased wear. These grooves have a depth of up to 0.374 µm in relation of the ideal spherical shape (HLP: 0.254 µm). Beyond this significant wear in the center of the dome, less wear and abrasion can be observed on the rest of the dome when using HEES compared to HLP. In this area surrounding the contact point, which is subject of significantly lower loads, the higher viscosity of HEES due to the higher VI leads to an improved friction reduction compared to HLP, which may explain lower wear in this area.

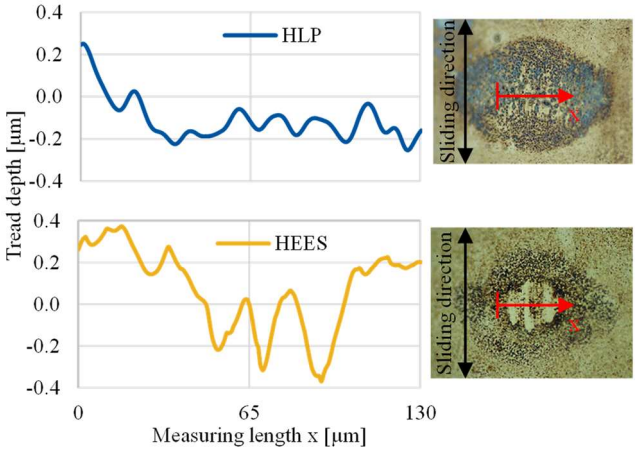

Figure 4.3. Wear contour ball HFRR

In addition, the discs used in the test were measured with the 3D light microscope. This procedure was also carried out by [30] in order to better describe the influence of diesel fuels. For this purpose, the surface profile of the worn area was recorded as in the previous investigations for the ball. Figure **4.4** shows the wear tracks for HLP and HEES as well as the surface profile transverse (x) to the sliding direction in the middle of the worn area. The disc operated with HLP is clearly more discoloured, which can possibly explained by different surface-active additives. When looking at the surface contour, it can be seen that contour with HLP has generally a deeper wear track than the disc operated with HEES. Furthermore, it can be seen that the wear contour of HLP is smoother than that of HEES, where some peaks can be seen, which can explain the groves that have formed in the ball.

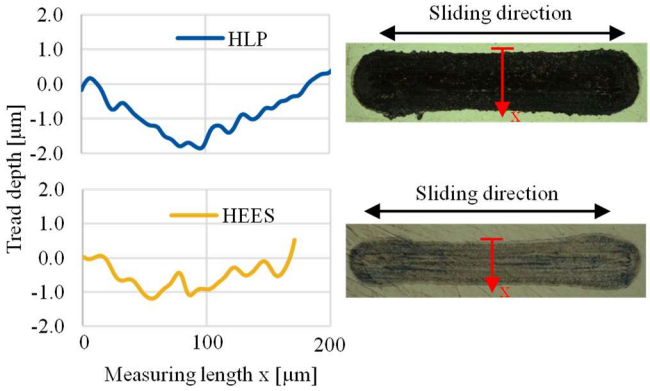

Figure 4.4. Wear contour disc HFRR

In general, the WSD results and the surface contours show contradictory results with regard to the wear properties of the fluids. On the one hand, the WSD of HEES is smaller than that of HLP. On the other hand, the test specimens of HEES show deep grooves of wear. In general, differences in the wear behaviour between the different fluids can be identified, so that the suitability for the wear testing of hydraulic fluids with the HFRR test method is

given. The extensive evaluation of the test with hydraulic oils represents a potential for further research work.

5. CONCLUSION AND OUTLOOK

The research carried out shows differences in the formation of load-bearing lubricating films between mineral oil based and ester-based lubricating films. Both the investigations in the disc-on-disc tribometer and the high frequency reciprocating (HFRR) test show differentiated results with respect to friction and wear of the two hydraulic oils considered.

Based on the results of the disc-on-disc tribometer test, an increased coefficient of friction was determined when using HEES compared to HLP. In the speed range between 600 and 2,300 rpm, the maximum coefficient of friction is 40 % higher than that of HLP. Furthermore, the flat course of the Stribeck curve of the HEES indicates that the contact pair is driven at boundary friction. For this reason, a possible explanation can be that the HEES formed a thinner lubricating film during investigations in the tribometer. In conclusion, the contact pair was operated in mixed friction over a wide speed range, whereas the formation of a hydrodynamic lubricating film can be identified from the measurement data when using HLP.

The results of the HFRR test showed that the use of HEES results in the formation of a 15 % smaller wear area (WSD) than with HLP. Further investigations using a 3D microscope showed that there is significantly deeper wear locally in the center of the dome of the ball, while using HEES. The grooves formed were almost twice as deep as with the use of mineral oil. A possible explanation for this could be the formation of an insufficient lubricating film due to the lower pressure viscosity dependence of the ester.

In general, it can be summarized that synthetic ester have different lubricating properties in relation to mineral oils. This must be taken into account especially when designing a hydraulic oil for use in systems in order to ensure safe and long-term failure-free operation. This leads to different friction conditions and challenges, especially in tribological contacts of hydraulic displacement units.

Subsequent research will relate to the influence of the results shown on the operation of hydraulic displacement units. Possible wear and losses occurring when using ester-based hydraulic oils will be analyzed. In addition to attributes during operation such as efficiency and heat emission, another necessary consideration is the service life and wear of the displacement units due to lubricant film formation in the tribological contacts. Furthermore, it is planned to increase the contact pressure during the disc-on-disc tribometer tests and to evaluate the influence on mineral oil based and ester-based hydraulic oils. These studies help to determine the performance characteristics of bio-based hydraulic oils identify possible development potential and make hydraulic systems more sustainable.

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