Oliver Feuchtmüller*
Lothar Hörl
Frank Bauer

An Empirical Study on the Wear of Reciprocating Hydraulic Rod Seals Using 15 Different Oils

Hydraulic oils differ in their chemical composition and viscosity substantially. Their influence on the wear of rod seals is neither fully understood nor can be predicted precisely. The primary objective of this study was to evaluate the influence of various oils and their wetting properties on the wear of rod seals. In an empirical study, the wear of 30 commercially available polyurethane U-cups was investigated with 15 different oils using a recently built test rig. The wear of rod seals increased with lower viscosity and higher polarity of the oil. Both oil properties are combined in a new parameter which helps to assess the lubricity of oil and the wear of rod seals.

Keywords: Hydraulic oils, Hydraulic rod seals, Lubricants, Wear of hydraulic rod seals

Received: July 27, 2022; *revised:* September 09, 2022; *accepted:* September 23, 2022

DOI: 10.1002/ceat.202200351

1 Introduction

Hydraulic rod seals are critical machine components of great technical and economic importance. A rod seal can be seen as a tribological system consisting of a seal ring, a rod, and a lubricant. A poorly designed rod seal results in a decreased averaged lifetime and low efficiency of hydraulic cylinders. If the rod seal fails, leakage and further consequences such as downtime and environmental pollution are unavoidable. The requirements of sealing systems related to safety and reliability are increasing steadily. Consequently, even existing sealing solutions must be reconsidered and improved. One challenge arises from new operating conditions and applications, e.g., the use of new or alternative hydraulic oils.

At outstroke and instroke, the rod drags oil into the sealing gap forming a lubrication film, which depends on the rod speed and the dynamic viscosity of the oil [1, 2]. The lubrication condition in the sealing gap directly influences friction, wear, and leakage. It follows that the hydraulic oil plays a key role in the smooth operation of a rod seal and the entire hydraulic system.

The oil film thickness in the sealing gap of practical relevant polyurethane U-cups is in the submicron to nanometer scale [3–5]. On this microscopic scale, properties of the oil can differ considerably from bulk properties due to interfacial phenomena at the solid-liquid interface [6, 7]. Those interfacial phenomena can be quantified using wetting parameters such as the contact angle, the work of adhesion or the spreading coefficient between the oil and the solid [8–10]. These wetting parameters depend on the surface energy (or surface tension) of the solid γ_s[1)] and the fluid γ_l. The surface energy of a material $\gamma = \gamma^p + \gamma^d$ is the sum of a polar part γ^p and a dispersive part γ^d. Both parts result from intermolecular forces [11]. The polarity $\chi^p = \gamma^p/\gamma$ of a material is defined as the quotient of the polar part and the total surface energy. The determination of the surface energy of a fluid or a solid is described in DIN 55660 [12–14].

Experimental work indicated the influence of the polarity of various base oils on the properties of polymers, which are used in sealing technology [15, 16]. The influence of wetting parameters on wear of rotary shaft seals was investigated and confirmed in [17–20]. Martinez [21] demonstrated the dependence of friction on measured contact angles in reciprocating and lubricated contacts. He did not observe any noteworthy wear in his 25-min friction tests. Correlations between friction and the spreading parameter, polarity and work of adhesion were observed in further studies, e.g., [22, 23].

Further studies demonstrated that wetting phenomena influence friction [24, 25] leakage [26], and wear [27] of reciprocating rod seals as well. However, the influence of wetting phenomena (e.g., surface energy and polarity of hydraulic oil) on the wear of a rod seal has been poorly investigated and is not yet understood. It is still a challenge to predict the wear of rod seals when using oils with different properties. Consequently, the development of new sealing systems is associated with time-consuming and expensive empirical tests. Despite decades of theoretical and empirical research [28, 29], optimization potential related to wetting phenomena is not considered so far. Advanced quantitative parameters are required to assess

1) List of symbols at the end of the paper.

Oliver Feuchtmüller, Lothar Hörl, Dr.-Ing. Frank Bauer
oliver.feuchtmueller@ima.uni-stuttgart.de
University of Stuttgart, Institute of Machine Components (IMA), Pfaffenwaldring 9, 70569 Stuttgart, Germany.

the lubricity of hydraulic oils and to predict the wear rates of rod seals.

The focus of this experimental study is on the influence of the surface energy, polarity, and dynamic viscosity of different oils on the wear of polyurethane U-cups. Wear tests were carried out using a recently developed and built test rig at constant operating parameters. The radial load and a geometrical parameter were measured to quantify the wear of the U-cups. Wetting properties of the sealing material, hydraulic rod, and various oils are provided. Based on the results, the wear of the U-cups can be correlated with wetting parameters of the sealing system.

2 Materials and Methods

2.1 Components and Characterization of the Sealing System

The sealing system used in this study consists of three main components: the seal ring, the rod, and the oil. Different parameters were determined to characterize the components.

2.1.1 Seal Ring

Commercially available and representative polyurethane U-cups from Freudenberg (Freudenberg Sealing Technology GmbH & Co. KG) labeled "T20 50x65x10" (Art.-No.: 40422194) were used. The sealing compound is based on a Shore 95 A polyurethane (95 AU V142). According to the datasheet, this compound is compatible with various oils such as typical HLP mineral oils, various fire-resistant oils (HFA, HFB, HFC, HFD), and HEES synthetic ester oils at an operating temperature of 25 °C. These U-cups fit for standardized housings defined in ISO 5597 [30] (bore diameter Ø 65 mm, rod diameter Ø 50 mm).

The surface energy of the sealing compound was determined based on contact angle measurements. A sufficient preparation process is a fundamental prerequisite to achieve a smooth and clean specimen surface. Any contamination or a certain roughness of the specimen can influence the measured contact angles considerably. Therefore, specimens were polished to a mirror smooth finish using an individual polishing process. Silicon carbide (SiC) paper was used in the polishing process. The grit size was increased stepwise from 800 to 4000. During the process, the abrasive paper was continuously rinsed with deionized water to remove any abraded particles. Care must be taken to ensure an evenly distributed material removal resulting in a flat surface.

An ionizer was used before the contact angle measurements on the surface to avoid electrostatical charge and to blow away any dust particles. Water and diiodomethane were used as reference liquids. The measurements were carried out following the recommendations in DIN 55660-2 [12]. The polar and dispersive parts of the surface energies were calculated using the approach from OWRK [31]. Polar and dispersive parts of surface energies of the reference liquids were adopted from Ström et al. [32].

The radial load F_{rad} of a seal is defined as the integral of the pressure in the sealing gap. Due to abrasive wear, the radial load of a hydraulic rod seal can decrease. Thus, the wear of such seals can be indicated and quantified by measuring the radial load. In this study, the radial loads of the seals were measured using the split mandrel technique, which is described in DIN 3761-9 (under revision) [33] and illustrated in Fig. 1a. The measurement device used has an automatic diameter control and is described by Feldmeth [34] in detail.

Before the measurements, the U-cups were mounted and stored at room temperature for approximately 24 h to simulate the installation conditions and to reduce relaxation effects. The radial load was measured 10 s after the U-cup was pushed over the split mandrel as it is suggested in [34]. The measurements were repeated five times for each U-cup to calculate a mean value. After each measurement, the U-cup was rotated by 90°.

The length of the beveled edge l was measured as a geometric parameter to indicate and quantify the wear of the U-cups used in this study, see Fig. 1b. Due to abrasive wear, the length of the beveled edge decreases ($l_{new} > l_{after\ test\ run}$). High wear is indicated by remarkably reduced lengths of the beveled edge. For the measurements, the IMA-Sealscanner® was employed (see [35]). This measurement device consists of a 2D laser scanner and a rotation drive to digitize the surface of the U-cup incrementally in three dimensions; see Fig. 1b. An individual algorithm was applied to determine the length of the beveled edge around the circumference automatically.

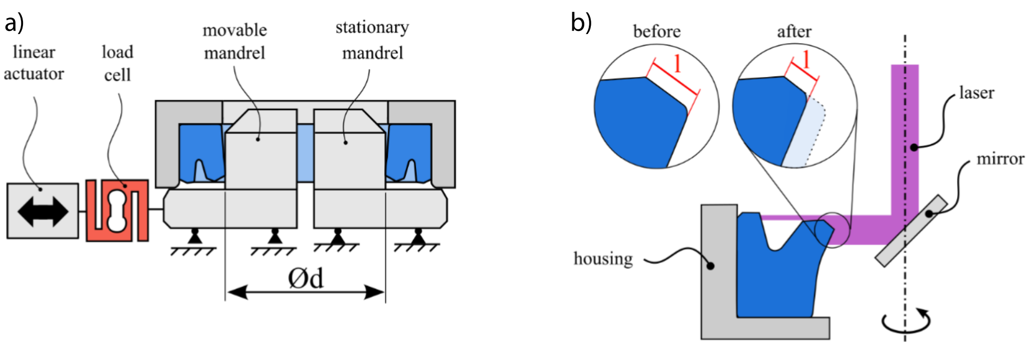

Figure 1. Schematic illustrations of the device to measure the radial load (a) and the analysis of the beveled edge of a U-cup measured with the IMA-Sealscanner® before and after a test run (b).

2.1.2 Rod

In this empirical study, specifically rough hydraulic rods were used. The hydraulic rods were hard-chromed and honed to a roughness of approximately Rz 3 µm (Ra 0.3 µm, Rmr (Rz/4) 70 %). The aim was to achieve rough surfaces with abrasive effect to accelerate the wear of the U-cups and to reduce the test time. In a previous research project such a rough honed rod caused moderate and well measurable wear at similar U-cups [36].

Fig. 2 shows the surface topographies, taken with a confocal microscope Keyence VK 9710, of three different hydraulic rods for illustration purposes. The rod with a common quality as specified by the company Weber-Hydraulik GmbH displays characteristic grooves in circumferential direction. Deep grooves which intersect in a certain angle are the result of the honing process and a characteristic feature of the rods used in this study.

The surface energy of a representative hydraulic rod was determined as described in DIN ISO 55660-2 [12]. Therefore, contact angles with water and diiodomethane were measured in a first step. In a second step, the surface energy was calculated using the approach from OWRK. Polar and dispersive parts of surface energy of the reference liquids were adopted from Ström et al. [32]. It must be noted that the surface roughness influences contact angles which are used to calculate the surface energy. In this study, a hard-chromed and polished rod was employed for contact angle measurements to eliminate the influence of surface roughness.

2.1.3 Oil

In total, 15 different sample oils were used in this study; see Tab. 1. The oils include various reference oils [37] from a German research association named Forschungsvereinigung Antriebstechnik e.V. (FVA) and typical hydraulic oils such as HLP mineral oil, fire-resistant HFC oil, biodegradable ester-based oils, fully synthetic oils based on polyalphaolefins, and silicone oils from various suppliers. The oils are based on different base oils and differ in their chemical composition and viscosity class.

The dynamic viscosity of each oil was measured using a common plate-to-plate rheometer (Type MCR 302 from Anton Paar GmbH) as described in DIN 53019 [38]. A shear rate of $100\,s^{-1}$ and a temperature of 25 °C were chosen.

The surface energies were analyzed using bubble pressure tensiometry; see [39] for details on the measuring method. A SITA Science Line T60 bubble pressure tensiometer was used. This bubble pressure tensiometer measures the dynamic pressure of a gas bubble at a flow rate in a liquid and calculates the surface energy automatically. The measurements were carried out at 23–25 °C and a bubble lifetime of 60 s. The measurement device was calibrated using distilled water before each measurement.

Table 1. The 15 different oils used for wear tests.

Name	Base oil	ISO VG	Supplier
FVA 2	Mineral	32	FVA
FVA 3	Mineral	100	FVA
Renolin B10 VG 32	Mineral	32	Fuchs Schmierstoffe GmbH
Renolin B20 VG 68	Mineral	68	Fuchs Schmierstoffe GmbH
FVA PAO 2	Polyalphaolefin	68	FVA
Renolin Unisyn 32 OL	Polyalphaolefin	32	Fuchs Schmierstoffe GmbH
Renolin Planto Tac 68	Ester	68	Fuchs Schmierstoffe GmbH
Renolin Plantohyd 32S	Ester	32	Fuchs Schmierstoffe GmbH
Renolin Plantosyn 68 HVI	Ester	68	Fuchs Schmierstoffe GmbH
Hydrotherm 46 M	Water-glycol	46	Fuchs Schmierstoffe GmbH
FVA PG 1	Polyglycol	46	FVA
FVA PG 3	Polyglycol	220	FVA
Renolin PG 32	Polyglycol	32	Fuchs Schmierstoffe GmbH
OKS 1050/0	Silicone	n/a	OKS Spezialschmierstoffe GmbH
OKS 1010/1	Silicone	n/a	OKS Spezialschmierstoffe GmbH

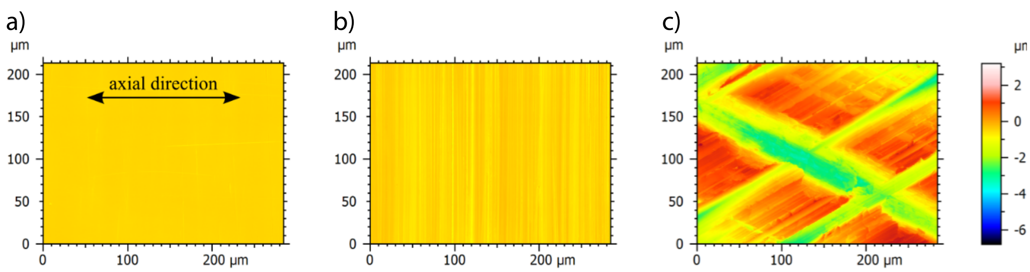

Figure 2. Surfaces of hydraulic rods with polished surface (a), common surface (b), and rough surface (c). 3D-topography analyzed with the Keyence VK 9710 confocal microscope.

The polar and dispersive parts of the surface energies were obtained using the pendant drop method as described in DIN 55660 [14]. Perfluorooctane with a surface energy of 14.0 mN m^{-1} was used as purely dispersive reference liquid. The density of each oil was determined using a pycnometer, since it was required as input for the analysis.

2.2 Wear Tests

Experimental tests were carried out on a recently built test rig.

2.2.1 Test Rig for Hydraulic Rod Seals

The test rig used is based on an arrangement as described in DIN 7986 [40] and illustrated in Fig. 3. Two test U-cups were mounted in a test chamber in opposite directions ("face to face"). Each test chamber was pressurized with hydraulic oil. A hydraulic rod was pushed and pulled through the chamber using an additional linear-hydraulic actuator to simulate outstroke and instroke of a typical hydraulic cylinder.

One special feature of the test rig used is the temperature unit which guaranteed a constant flow rate through hollow hydraulic rods. Thus, a defined and constant temperature of the rod surface was achieved. The oil temperature in the pressure chamber and the surface temperature of the rod were equal.

To pressurize the test chambers, additional hydraulic cylinders and a separate hydraulic pump were employed. The additional cylinders separated the oil circuits and avoided contaminations and mixing between different oils. Before the tests, all components were rinsed and cleaned properly using aceton and petroleum ether.

2.2.2 Operating Parameters

The test duration was 400 000 double strokes (approximately 22 days) which resulted in a cumulated sliding distance of 400 km for each setup. The operating conditions were constant during the test time. The oil pressure in the test chambers was set to 50 bar, the rod speed was 250 mm s^{-1}, and the temperature was set to 25 °C. Before starting the wear tests, the test arrangement and its components were checked during an additional short running-in time. The influence of this short running-in phase on the wear of the rod seals can be neglected.

3 Results

In this study, the dynamic viscosities and the surface energies of 15 oils were measured. Wear tests were carried out with 15 different oils and 30 U-cups (two U-cups in each test chamber). The wear of each U-cup was quantified by the radial load and the length of the beveled edge.

3.1 Analysis

The viscosities of the oils were measured at 25 °C, which corresponds to the temperature of the wear tests. The viscosities of the oils at 25 °C differ in a wide range; see Fig. 4a. FVA PG 3 has the highest viscosity with 489 mPa s and OKS 1050-0 has the lowest viscosity with 44 mPa s.

The polar and dispersive parts of the surface energy of the oils were determined using bubble pressure tensiometry and the pendant drop method. The results show that the mineral oils, polyalphaolefins, and esters have similar surface energies, polar and dispersive parts. Consequently, the polarities of those oils are similar. It is remarkable that the water-glycol-based Hydrotherm 46 M has a high surface energy and polarity compared to the other oils. In contrast, both silicone oils have lower surface energies. Fig. 4b shows the dispersive and polar part of the surface energy of each oil. Contact angle measurements on the seal compound and on a hydraulic rod were carried out to calculate the surface energies of both materials. The rod has a higher surface energy γ_s^d = 33.3 mN m^{-1}, γ_s^p = 20.5 mN m^{-1}) than the seal compound (γ_s^d = 39.6 mN m^{-1}, γ_s^p = 3.5 mN m^{-1}) and a higher polarity.

3.2 Wear

The radial load of each U-cup was determined after the endurance tests to indicate the wear. Remarkable differences were found depending on the oil in the test chamber. The measured radial loads were in the range of approximately 220 to almost 600 N. Fig. 5a shows the radial load for each U-cup and the mean value for both U-cups in one test chamber after the test runs.

Furthermore, the mean length of the beveled edge of each U-cup was measured after the endurance tests to indicate the wear. Depending on the oil, remarkable differences were identified. The lengths were in the range of approximately 1.26 mm, which is almost equal the length of a new U-cup

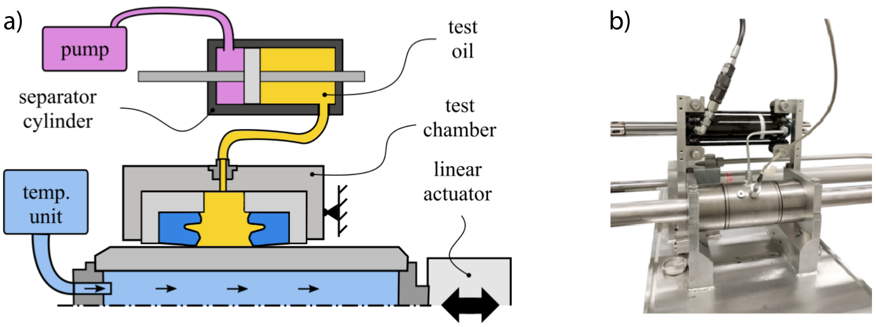

Figure 3. Schematic illustration (a) and a picture (b) of the test arrangement.

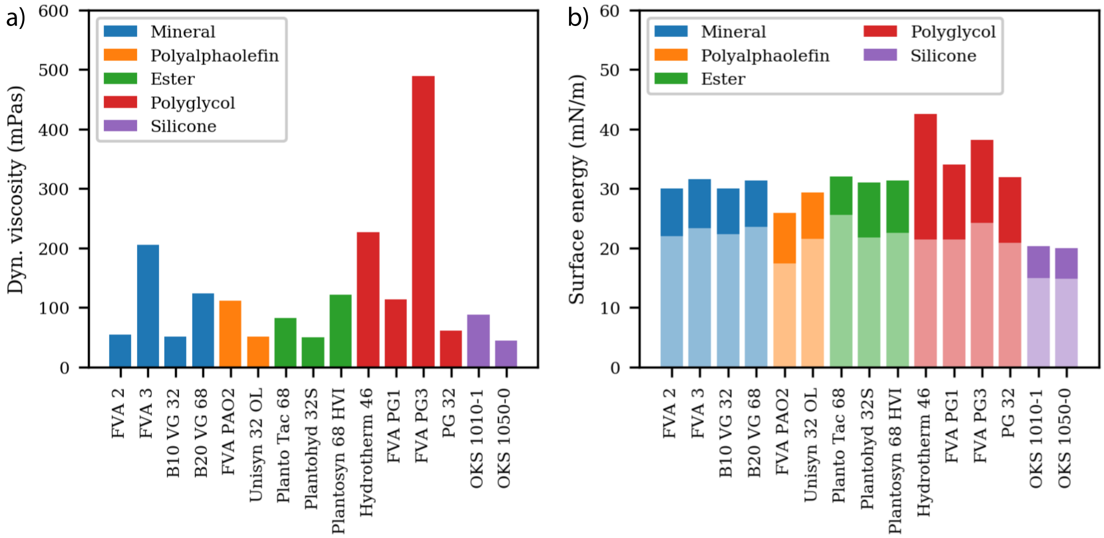

Figure 4. Properties of the used lubricants; dynamic viscosities at 25 °C (a), dispersive (light), and polar (dark) parts of the surface energies (b).

without wear, and 0.59 mm. The results are displayed in Fig. 5b.

4 Discussion

Based on the results, the wear of the U-cups after the test runs with different oils can be correlated with wetting properties of the sealing system. A combined and normalized parameter to quantify the wear of the U-cups is introduced.

4.1 Combined Wear Parameter

In this study, an almost linear correlation between the measured radial loads and the lengths of the beveled edges of the U-cups is found. Fig. 6a plots the radial load against the length of the beveled edge. The higher the radial load, the higher the length of the beveled edge. Both parameters can be combined to a single parameter, which is defined as the product of both parameters. This combined parameter is appropriate to quantify the wear of the U-cups. For the sake of simplicity, the combined parameter is normalized to describe the relative wear in %, as shown in Fig. 6b.

$$\text{Relative wear} = \frac{\max(F_{\text{rad}} l_{\text{after test run}}) - F_{\text{rad}} l_{\text{after test run}}}{\max(F_{\text{rad}} l_{\text{after test run}})} \tag{1}$$

The U-cup with the highest radial load and length of the beveled edge serves as a reference and has a relative wear of 0 % by definition. The wear of the other U-cups is given in % relative to the reference U-cup. A wear rate of 100 % is equivalent to a U-cup which has a radial load of 0 N or a U-cup which has no beveled edge due to high abrasive wear.

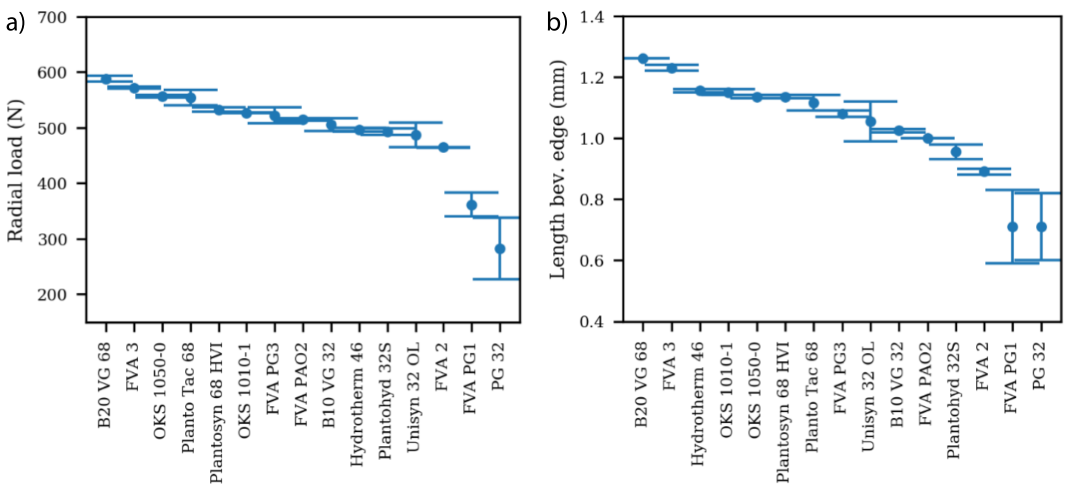

Figure 5. Measured radial loads (a) and lengths of the beveled edges (b) of the U-cups after the test runs.

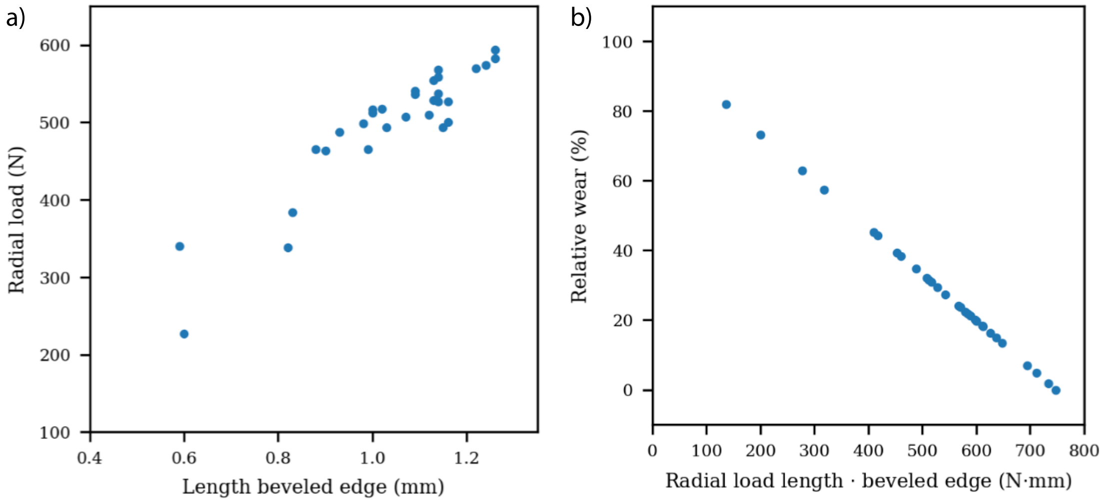

Figure 6. Correlation between the radial loads and the lengths of the beveled edges of the U-cups after the test runs (a). Relative wear plotted over the product of the radial load and the length of the beveled edge (b).

4.2 Correlations

For the wear tests, similar U-cups, rods, and operating parameters were chosen. The results confirm that the oil has a considerable influence on the wear of hydraulic rod seals. In this section, correlations between the wear and various parameters are discussed.

Fig. 7a shows the relative wear of the rod seals over the dynamic viscosity of the oil. The results indicate that the wear depends on the viscosity, if only oils with similar chemical properties are considered, e.g., FVA 2 with a viscosity of 54 mPa s leads to higher wear than FVA 3 with a viscosity of 205 mPa s. One possible reason of the influence of viscosity on wear may result from its influence on the oil film generation. The oil film thickness in the sealing gap and thus the lubrication condition depend on the viscosity of oil [3, 4]. It is assumed that a thick film separates the U-cup and the rod and reduces the wear rate.

If lubricants of different base oils are compared, this relationship is no longer valid, e.g., the use of polyglycol-based oils resulted in higher wear than the other oils. Consequently, the dynamic viscosity is a key factor which influences the wear of rod seals but not a sufficient parameter to quantify the lubricity of oil generally. Further properties of the oil must have an influence on the wear of rod seals.

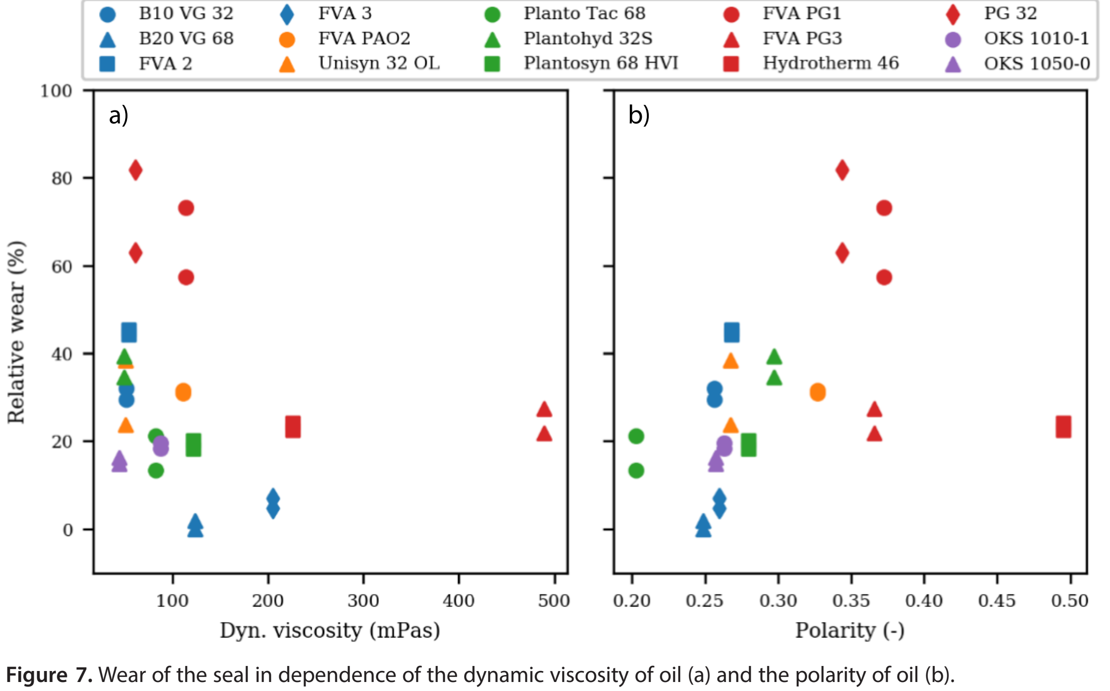

Figure 7. Wear of the seal in dependence of the dynamic viscosity of oil (a) and the polarity of oil (b).

Fig. 7b displays the relative wear as a function of the polarity of the oil. An almost linear correlation between the polarity of oil and the wear of the seals is revealed. One exception is the HFC hydraulic fluid Hydrotherm 46 M which has the highest polarity but caused a moderate wear rate. Nevertheless, the polarity of the oil should be considered when quantifying the lubricity of oil. It is conceivable that chemical interactions between the seal compound and the oil depend on the polarity of the oil. According to the results it can be assumed that oils with higher polarity have a negative effect on the wear resistance of the sealing compound.

Furthermore, the wear was correlated with the spreading parameter, work of adhesion, and total surface energy of each oil. The correlations were not better than the correlation between the polarity and the wear. That is why those parameters are not considered in the following section.

So far, the dynamic viscosity and the polarity of the oil were identified as key factors influencing the wear of rod seals. The tests revealed that oil with high viscosity and low polarity decreases the wear of rod seals. In a next step, both key factors are combined to a new parameter, which can be used to illustrate the wear of the seals depending on properties of oil. The new parameter

$$Lub_{\text{oil}} = \frac{\eta}{\chi_{\text{p}}^2} \quad (2)$$

includes the viscosity η and polarity χ_{p} of the oil. The quadrature of the polarity increases its weighting and indicates that the polarity of the oil has a strong influence on the wear of rod seals. Fig. 8 plots the wear of the seals against this new parameter Lub_{oil}. Obviously, the new parameter indicates the relation between properties of oil and the wear of the seals and is better suited than simple parameters such as the viscosity or polarity.

5 Summary and Conclusion

In this research project, correlations between the wear behavior of a rod seal and wetting parameters were investigated experimentally.

The analysis of wetting properties includes the determination of the dispersive and polar parts of the surface energies of 15 different oils. The samples include commercially available hydraulic oils and other oils made from different base oils (mineral oils, polyalphaolefins, esters, polyglycols, silicone oils). Dispersive and polar parts of the surface energy of a hard-chrome plated rod were determined by means of contact angle measurements. Reproducible results for a polyurethane compound were obtained by a grinding/polishing process and contact angle measurements.

The wear of rod seals was investigated in endurance tests, each with 400 000 double strokes which corresponds to a sliding distance of 400 km. Abrasive rods were used to increase the wear rates and to reduce the test duration. The wear behavior of 30 polyurethane U-cups was investigated with 15 different oils. The oils differ in viscosity, chemical composition, and wetting properties remarkably. The results highlight the influence of the oil on the wear of rod seals. To quantify the wear, a characteristic geometrical parameter of the U-cup and the radial load were used.

The wear behavior of the polyurethane U-cups was mapped as a function of a new parameter, which includes the dynamic viscosity and the polarity of oil. It is concluded that high wear rates can result from oils with lower viscosities and higher polarities. In practice, these new insights help to assess and compare the lubricity of hydraulic oils and the wear behavior of different sealing systems. When high wear rates occur, it is recommended to use oils with higher dynamic viscosity and lower polarity.

Furthermore, the use of abrasive rods is a promising approach for evaluating oils and sealing rings (geometry and materials) in accelerated test rig trials. When using rods of common quality, reduced wear can be expected under similar test and operating conditions.

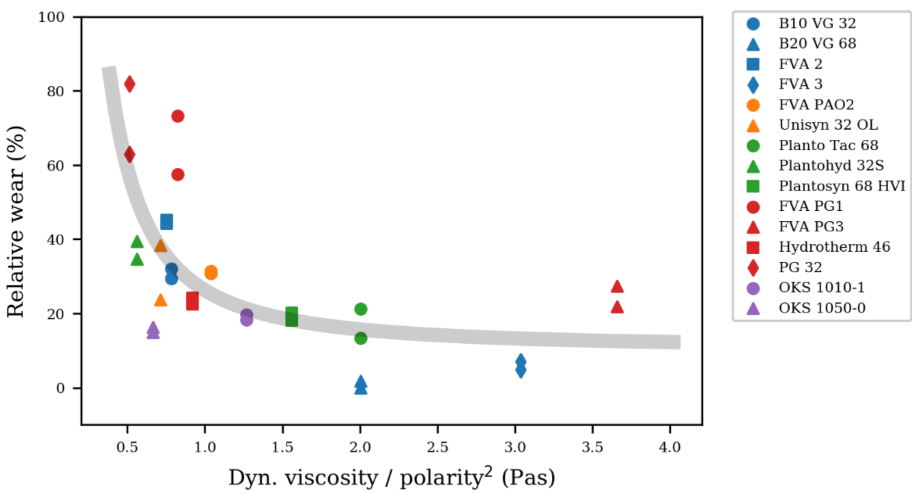

Figure 8. Correlation between the wear of the seal and a new parameter including the dynamic viscosity and the polarity of oil.

Data Availability Statement

Data available on request from the authors.

Acknowledgment

This work is part of the IGF project 20105 N/1 of the Forschungskuratorium Maschinenbau e.V. (FKM) and funded by the AiF as a support of the Industrielle Gemeinschaftsforschung (IGF, Industrial Collective Research) by the Federal Ministry for Economic Affairs and Energy (BMWi) on the basis of a decision by the German Bundestag. Open access funding enabled and organized by Projekt DEAL.

The authors have declared no conflict of interest.

Symbols used

F_{rad}	[N]	radial load of a seal
l	[mm]	length of the beveled edge of a U-cup
Lub_{oil}	[Pa s]	new parameter to map the wear of rod seals

Greek letters

γ	[mN m^{-1}]	surface energy
γ_l^d	[mN m^{-1}]	dispersive part of the surface energy of a liquid
γ_l^p	[mN m^{-1}]	polar part of the surface energy of a liquid
γ_s^d	[mN m^{-1}]	dispersive part of the surface energy of a solid
γ_s^p	[mN m^{-1}]	polar part of the surface energy of a solid
η	[Pa s]	dynamic viscosity
χ_P	[–]	polarity of material

References

[1] *Fluid Sealing Technology: Principles and Applications* (Eds: H. K. Müller, B. S. Nau), Marcel Dekker, New York **1998**.
[2] R. Flitney, *Seals and Sealing Handbook*, 6th ed., Butterworth-Heinemann, Woburn, MA **2014**.
[3] O. Feuchtmüller, L. Hörl, F. Bauer, *Tribol. Int.* **2021**, *162*, 107102. DOI: https://doi.org/10.1016/j.triboint.2021.107102
[4] O. Feuchtmüller, N. Dakov, L. Hörl, F. Bauer, *Lubricants* **2021**, *9 (9)*, 95. DOI: https://doi.org/10.3390/lubricants9090095
[5] L. Hörl, W. Haas, U. Nißler, *Sealing Technol.* **2009**, *(12)*, 8–13. DOI: https://doi.org/10.1016/S1350-4789(09)70594-X
[6] D. Y. C. Chan, R. G. Horn, *J. Chem. Phys.* **1985**, *83 (10)*, 5311–5324. DOI: https://doi.org/10.1063/1.449693
[7] M. L. Gee, P. M. McGuiggan, J. N. Israelachvili, A. M. Homola, *J. Chem. Phys.* **1990**, *93 (3)*, 1895–1906. DOI: https://doi.org/10.1063/1.459067
[8] T. Young, *Philos. Trans. R. Soc. London* **1805**, *95*, 304–310. DOI: https://doi.org/10.1098/rstl.1805.0005
[9] W. D. Harkins, A. Feldman, *J. Am. Chem. Soc.* **1922**, *44 (12)*, 2665–2685. DOI: https://doi.org/10.1021/ja01433a001
[10] J. W. Drelich, L. Boinovich, E. Chibowski, C. Della Volpe, L. Hołysz, A. Marmur, S. Siboni, *Surf. Innovations* **2020**, *8 (1–2)*, 3–27. DOI: https://doi.org/10.1680/jsuin.19.00007
[11] S. Wu, *J. Adhes.* **1973**, *5 (1)*, 39–55. DOI: https://doi.org/10.1080/00218467308078437
[12] DIN 55660-2:2011-12, *Beschichtungsstoffe – Benetzbarkeit – Teil 2: Bestimmung der freien Oberflächenenergie fester Oberflächen durch Messung des Kontaktwinkels*, Beuth, Berlin **2011**.
[13] DIN 55660-3:2011-12, *Beschichtungsstoffe – Benetzbarkeit – Teil 3: Bestimmung der Oberflächenspannung von Flüssigkeiten mit der Methode des hängenden Tropfens*, Beuth, Berlin **2011**.
[14] DIN 55660-4: 2012-04, *Beschichtungsstoffe – Benetzbarkeit – Teil 4: Bestimmung des polaren und dispersen Anteils der Oberflächenspannung von Flüssigkeiten aus einer Grenzflächenspannung*, Beuth, Berlin **2012**.
[15] G. van der Waal, *Lubr. Sci.* **1985**, *1 (4)*, 280–301. DOI: https://doi.org/10.1002/jsl.3000010404
[16] M. Torbacke, *Lubr. Sci.* **2005**, *22 (2)*, 123–142. DOI: https://doi.org/10.1002/jsl.2005.22.2.123
[17] P. Schuler, W. Haas, *Sealing Technol.* **2016**, *2016 (6)*, 7–11. DOI: https://doi.org/10.1016/S1350-4789(16)30183-0
[18] P. Schuler, Einfluss von Grenzflächeneffekten auf den Dichtmechanismus der Radial-Wellendichtung, *Ph.D. Thesis*, Universität Stuttgart **2014**.
[19] P. Fricker, M. Baumann, F. Bauer, *Wear* **2021**, *477*, 203897. DOI: https://doi.org/10.1016/j.wear.2021.203897
[20] C. Burkhart, M. Heier, R. Merz, J. Lösch, C. Wagner, S. Thielen, B. Sauer, M. Kopnarski, H. Hasse, *62. Tribologie-Fachtagung*, online, September **2021**.
[21] L. Martínez et al., *Tribol. Int.* **2009**, *42 (4)*, 584–590. DOI: https://doi.org/10.1016/j.triboint.2008.06.016
[22] M. Kalin, M. Polajnar, *Tribol. Lett.* **2013**, *52 (2)*, 185–194. DOI: https://doi.org/10.1007/s11249-013-0194-y
[23] M. Minn, S. K. Sinha, *Wear* **2010**, *268 (7–8)*, 1030–1036. DOI: https://doi.org/10.1016/j.wear.2009.12.032
[24] M. Wangenheim, Untersuchungen zu Reibmechanismen an Pneumatikdichtungen, *Ph.D. Thesis*, Leibniz Universität Hannover **2012**.
[25] F. Fischer, H. Murrenhoff, O. Reinertz, *19th Int. Sealing Conf.*, Stuttgart, October **2016**.
[26] G. Barillas, U. Ridder, F. Kaiser, *20th Int. Sealing Conf.*, Stuttgart, October **2018**.
[27] E. Hörl, in *Elastomere Dichtungssysteme* (Eds: G. Streit et al.), 4th ed., Expert Verlag, Tübingen **2011**.
[28] G. K. Nikas, *Proc. Inst. Mech. Eng., Part J* **2010**, *224 (1)*, 1–23. DOI: https://doi.org/10.1243/13506501JET607
[29] B. S. Nau, *Proc. Inst. Mech. Eng. Part J: J. Eng. Tribol.* **1999**, *213 (3)*, 215–226. DOI: https://doi.org/10.1243/1350650991542956
[30] DIN ISO 5597, *Fluidtechnik – Zylinder – Maße und Grenzabmaße für Einbauräume für Kolben- und Stangendichtungen für hin- und hergehende Anwendungen*, Beuth, Berlin **2019**.

[31] D. K. Owens, R. K. Wendt, *J. Appl. Polym. Sci.* **1969**, *13*, 1741–1747. DOI: https://doi.org/10.1002/app.1969.070130815
[32] G. Ström, M. Fredriksson, P. Stenius, *J. Colloid Interface Sci.* **1987**, *119 (2)*, 352–361. DOI: https://doi.org/10.1016/0021-9797(87)90280-3
[33] DIN 3761-9:1984-1, *Radial-Wellendichtringe für Kraftfahrzeuge, Prüfung Radialkraft-Meßgerät-Digital*, Beuth, Berlin **1984**.
[34] S. Feldmeth, M. Stoll, F. Bauer, *61. Tribologie-Fachtagung*, online, September **2020**.
[35] M. Baumann, F. Bauer, *20th Int. Sealing Conf.*, Stuttgart, October **2018**.
[36] M. Stoll, L. Hörl, W. Haas, *Einfluss der Oberflächenrauheit von Hydraulikstangen auf Reibung und Leckage*, Abschlussbericht IGF Nr. 18054 N/1, Universität Stuttgart, Stuttgart **2017**.
[37] *Referenzöle für Wälz- und Gleitlager, Zahnrad- und Kupplungsversuche – Datensammlung für Mineralöle*, Forschungsvereiningung Antriebstechnik e. V., Frankfurt/Main **1985**.
[38] DIN 53019-1:2008-09, *Viskositmerie – Messung von Viskositäten und Fließkurven mit Rotationsviskosimetern – Teil 1: Grundlagen und Messgeometrie*, Beuth, Berlin **2008**.
[39] V. B. Fainerman, R. Miller, in *Drops and Bubbles in Interfacial Research* (Eds: D. Mobius, R. Miller), 1st ed., Elsevier, New York **1997**.
[40] ISO 7986:1997-07, *Hydraulic Fluid Power – Sealing Devices – Standard Test Methods to Assess the Performance of Seals Used in Oil Hydraulic Reciprocating Applications*, Beuth, Berlin **1997**.

