Influence of Lubricants on the Thermal Behaviour of Rotary Shaft Seals

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The thermal behaviour of rotary shaft seals is affected by the fluid which has to be sealed. The fluid lubricates the sealing contact and determines the frictional heat generated there. Additionally, the fluid influences the heat transfer to the environment. Both aspects were experimentally analysed by measuring the friction torque and the temperature on the air side near the sealing contact during test runs with 16 different lubricants. The measurement results show that the fluid affects the heat generation much more than the heat transfer. In general, the friction torque is higher for more viscous fluids. However, several exceptions restrict this finding.

1 Introduction

Many applications require seals that retain lubricants or other fluids within machines and prevent the entry of dirt into these machines. In applications with rotating shafts, mostly rotary shaft lip-type seals are used [1, 2]. The rotary shaft seal, the surface of the shaft and the fluid, which has to be sealed (often oil), form a tribological system [3], that is significantly affected by its periphery and the operating conditions.

Elastomeric rotary shaft lip-type seals are standardised in national and international standards [4-6]. They consist of a metal insert to which a sealing lip is attached, Figure 1. During assembly, the sealing lip and a garter spring are widened. The sealing lip is pressed against the shaft surface, forming a slim contact area between its sealing edge and the shaft. The width of the contact area is approximately 0.1 to 0.2 mm. The average contact pressure in the contact area is about 1 MPa.

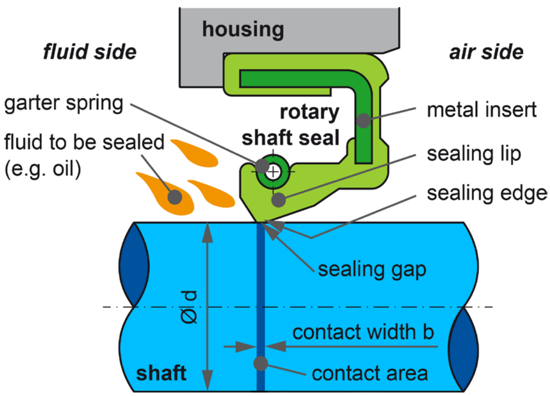

Figure 1: Sealing system consisting of rotary shaft seal, shaft and fluid [7]

During operation, a thin lubrication film arises in the contact area between the sealing edge and the shaft surface. This lubrication film separates the surfaces and forms the so-called sealing gap. A microscopic pumping mechanism prevents leakage [8].

Frictional heat and contact temperature

Energy is dissipated in the contact area during operation due to friction. This means, kinetic energy is converted to frictional heat. The temperature in the contact area is higher, the more frictional heat is generated and the poorer this heat is transferred away from the contact area. A high contact temperature is very harmful to the sealing system since it accelerates the aging of the elastomer as well as the deposition of oil carbon. Both mechanisms reduce the lifetime of the sealing system.

In order to achieve a long lifetime, moderate temperatures in the sealing system, and especially in the contact area, must be ensured during the development of new machines. A reliable and economic design of the sealing system is only possible if the temperature range in the contact area is known.

The contact temperature of a sealing system depends on its thermal behaviour which is affected by many influencing factors. These influencing factors either affect the heat generation (e.g. the surface topography of the shaft) or the heat dissipation (e.g. the thermal conductivity of the shaft material) or even both (e.g. shaft speed). Figure 2 shows the most important influencing factors, their interactions and their effects on the contact temperature. The lubricant determines some of the most relevant influencing factors (marked in yellow). In this work, the focus is on the lubricant's viscosity which strongly affects the lubrication condition in the sealing gap and thus the generation of frictional heat. Additionally, the lubricant's influence on the heat transfer is experimentally analysed. Further aspects, such as the wetting behaviour, are not in the main focus of this work, even though they have a significant influence [9].

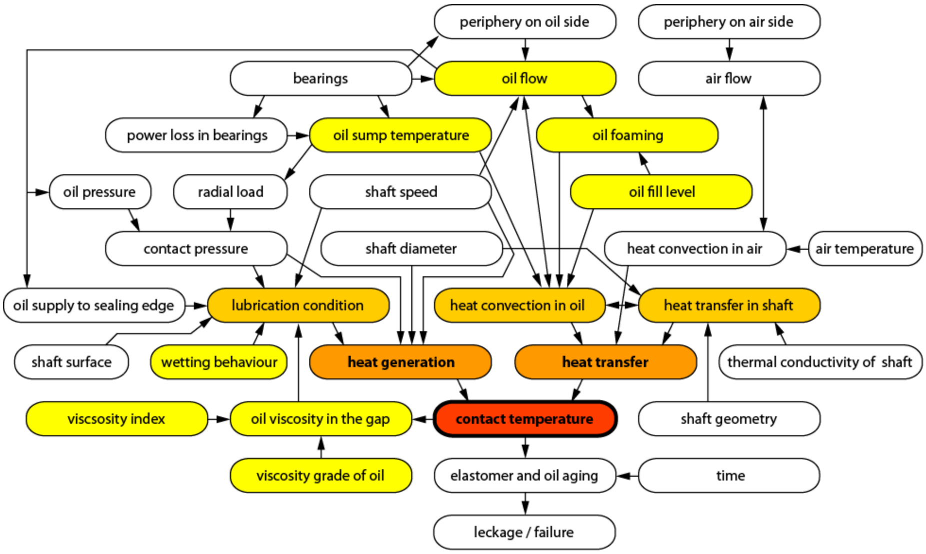

Figure 2: Factors influencing the contact temperature

2 Test method and test material

The thermal behaviour was analysed by performing combined measurements of the friction torque and the contact temperature at a test rig for rotary shaft seals.

2.1 Friction torque and contact temperature measurements

The friction torque measurements were performed at the high-speed friction torque test rig, Figure 3. The shaft sleeve is mounted on a shaft adapter and is driven by a spindle via an HSK collet chuck. The seal ring is pressed into a seal holder that is mounted at the test chamber. The test chamber is supported by an aerostatic bearing allowing it to rotate almost without friction. The friction torque of the sealing system can be measured via a load cell that supports the test chamber via a lever arm. The test chamber can be filled with lubricant and heated with cartridge heaters.

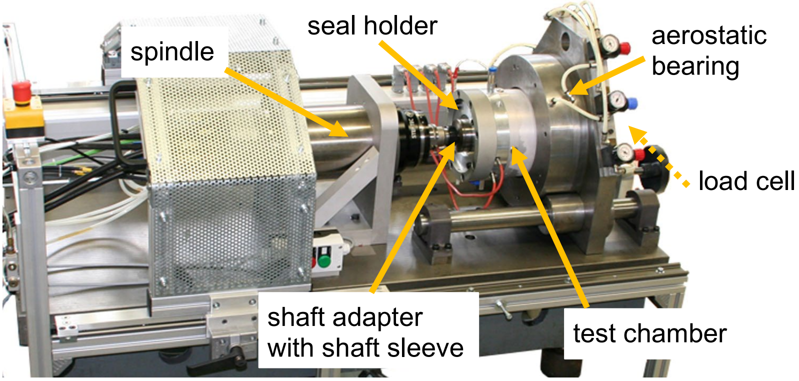

Figure 3: Sealing system consisting of rotary shaft seal, shaft and fluid

The test cycle consists of a 12 hours running-in period at 1000 rpm which is followed by 14 periods in which the shaft speed is increased from 500 to 10000 rpm in steps of approximately 25 % every 30 minutes. During the whole test cycle, the test chamber is heated up to 80 °C by the cartridge heaters. The oil fill level is centre of the shaft. The oil volume is approximately 1.15 litre.

The friction torque is calculated as the arithmetic mean over the last 10 minutes of each speed period. The contact temperature is measured on the air side near the sealing contact in the last few minutes of a speed period using a thermal imager Fluke Ti480-Pro (settings: emissivity = 0.95, transmission = 100 %; accuracy ±2 °C or ±2 %).

2.2 Rotary shaft seals

Standard rotary shaft seals of the type BAUM5X7 made of fluororubber 75FKM585 with the dimensions 100x80x10 mm manufactured by Freudenberg FST GmbH were used for all test runs. The seal rings were named with KT-F8-XXX where XXX stands for a consecutive number starting with 001. Since a new seal ring was used for every new test run, the name of the seal ring was also used as the ID of the test run.

The radial load was measured for all seal rings using the measuring methods B and C suggested in [10]. For each method 5 separate measurements are carried out and the arithmetic mean of the measurements No. 2 to 5 is referred to as the radial load. These measurements are either performed in new condition (Method B) or after 24 hours of storage on a mandrel with nominal diameter (Method C). Additionally, the radial load of 5 seal rings was measured according to method D in the temperature range between 40 and 160 °C in steps of 20 K. The radial load of the seal rings KT-F8-001 to KT-F8-090 according to method B is 40.2 N and 31.6 N according to method C, Figure 4. The measurements according to method D show a linear decrease of the radial load with increasing temperature down to 23.0 N at 160 °C.

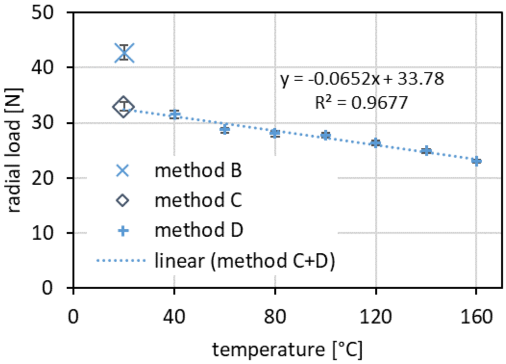

Figure 4: Radial load of the seal rings KT-F8-016 to KT-F8-020

2.3 Shaft sleeves

Plunge ground shaft sleeves made of bearing steel 100Cr6 were used as counterfaces for the seals. The thermal conductivity of 100Cr6 is 40 to 45 W/m·K [11], which is slightly lower than that of plain carbon steels such as C45E (47 to 52 W/m·K) and about 3 times higher than that of stainless steels such as X5CrNi18-10 (15 to 16 W/m·K) [12]. All shaft sleeves were subjected to a comprehensive surface analysis containing 2D and 3D roughness, microlead, macrolead and thread method. All sealing counterfaces were inconspicuous concerning lead and have 2D and 3D roughness parameters within the standard specifications. The roughness R_z was between 2.9 and 4.2 µm (measured in axial direction).

2.4 Lubricants

As lubricants, 16 different oils were used, Table 1. Unless otherwise noted, the lubricant properties were either taken from the technical or the safety data sheet of the respective lubricant. If the viscosity index (VI) was not given by the manufacturer, the VI was calculated using online tools such as [13]. At least 2 friction torque test runs were performed with each lubricant [14].

The lubricants are divided into 6 groups each containing at least 2 lubricants:

- Mineral oils
- Automotive oils (Engine oils and ATF with relatively low viscosity)

- Polyalkylene glycol-bases lubricants
- Lubricants based on synthetic hydrocarbons such as polyalphaolefin fluids
- Lubricants based on synthetic ester oils
- Silicone oils

Table 1: Lubricant properties

ID (short name)	Full name	Description (application and/or base oil)	Additives	Kinematic viscosity [Pa·s] at …		Viscosity index (VI)	Thermal conductivity at 40 °C [W/(m·K)]
				40 °C	100 °C		
FVA1	FVA 1 [15]	Mineral oil (paraffin-based solvent raffinates) [15]	No	15	3.4	97	0.128 [9]
FVA2	FVA 2 [15]		No	32	5.3	95	unknown
FVA3	FVA 3 [15]		No	95	10.7	95	0.138 [9]
CLP220	SEW GearOil Base E220 E1	Gear oil on mineral base	Yes	220	18.9	100	unknown
0W-30	Fuchs Titan Supersyn Longlife 0W-30	Engine oil	Yes	67.6	12.1	178	unknown
0W-20	Castrol BOT 920 0W-20		Yes	43.6	8.2	164	unknown
ATF	Shell L12108	Automatic transmission fluid (ATF)	Yes	24.9	5.6	175	unknown
PG1	FVA PG1 [15] (Clariant B11/50)	Polyalkylene glycol (PAG)	No	70	14	209	0.169 [9]
PG180	Mobil Glygoyle 22	PAG-based lubricant for extreme temperature applications	Yes	177	25.1	175	unknown
PAO2	FVA PAO 2 [15] (Mobil Gargoyle Arctic SHC 226E)	Refrigeration oil on polyalphaolefin (PAO) basis	No	62	10.0	148	0.138 (at 50 °C) [15]
HC150	Nabtesco RV Oil SB 150	Synthetic hydrocarbon oil and mineral oil	unknown	158	19.4	140	unknown
E150-1	Fuchs Cassida GLE 150	Gear lubricant for food industry	unknown	150	19	144	unknown
E150-2	Klüberbio RM2-150	Biodegradable stern tube oil based on synthetic ester oil	unknown	150	18	135	0.167 [9]
E220	Panolin EP Gear Synth 220	Biodegradable gear oil based on synthetic ester oil	Yes	220	24.0	136	unknown
SI50	OKS 1050/0	Silicone oil (Polydimethylsiloxane)	No	50 (at 25 °C)		n.a.	0.150 [9]
SI100	OKS 1010/1		No	100 (at 25 °C)		n.a.	0.154 [9]

The dynamic viscosity η of the lubricants was measured using an MCR302 rheometer manufactured by Anton Paar Germany GmbH. The measurements were performed using a plate-plate system with a diameter of 25 mm and an initial gap height of 0.5 mm. The lubricant was applied at room temperature and heated to 200 °C with a heating rate of 1 K every 5 min. The shear rate (at the outer diameter) was 100 1/s. Due to thermal expansion, the gap height decreased with increasing temperature. The rate of the gap reduction was measured to -0.48 µm/K which was used for compensating the measured viscosity. The kinematic viscosity ν is calculated by dividing the dynamic viscosity by the density ρ.

$$\nu(\vartheta) = \frac{\eta(\vartheta)}{\rho(\vartheta)} \tag{1}$$

The kinematic viscosity is calculated at temperatures of 40 and 100 °C for plausibility check. The calculated values differ on average by less than 5 % and in maximum by 13 % from the manufacturers' specification in the data sheets.

3 Results

First, the measurement results will be analysed regarding the aspect of heat generation. For this purpose, the friction torque is observed. The second part covers the heat transfer. For this analysis a self-defined parameter called thermal resistance is used.

3.1 Heat generation/ friction torque

Figure 5 shows the raw data of the exemplary test run KT-F8-023 using 0W-30 as lubricant. After a short heating-up period of approximately half an hour, the friction torque reaches a steady state for the rest of the running-in period. During the measuring period, the friction torque changes immediately after each speed change.

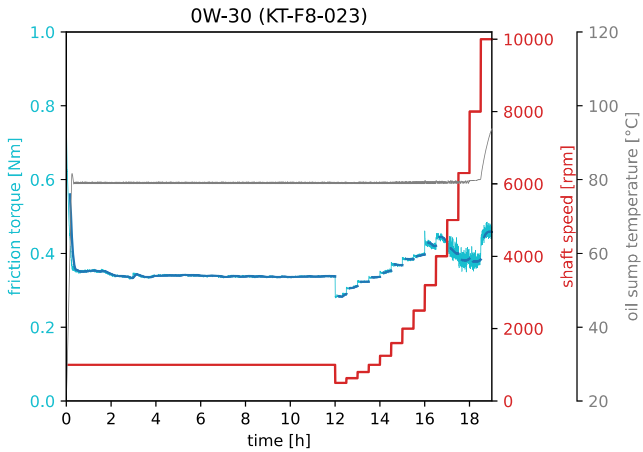

Figure 5: Friction torque and oil sump temperature over time during test run KT-F8-023 with lubricant 0W-30

During the last speed step, the heat generation of the sealing system exceeds the heat transfer by natural convection resulting in the test chamber and the containing oil heating up more than the nominal oil sump temperature of 80 °C.

Some of the lubricants show much more fluctuations in the friction torque during the test run even in the periods of constant operating parameters. Figure 6 shows such an example obtained with PG1 as lubricant. Both, during the running-in period (first 12 hours) and within each speed step, no steady-state is reached. This might be caused by a relatively poor wetting behaviour (compared to mineral-oil-based lubricants of the same viscosity, as already observed [16]) resulting in mixed lubrication.

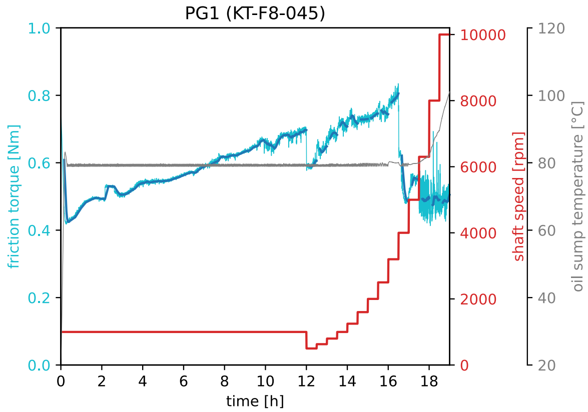

Figure 6: Friction torque and oil sump temperature over time during test run KT-F8-045 with lubricant PG1

By averaging the friction torque over the last 10 minutes of each speed step, the friction torque can be plotted as a function of the shaft speed. Figure 7 shows the friction torque measured in 4 test runs with ATF as lubricant. The 4 individual measurement curves agree very well.

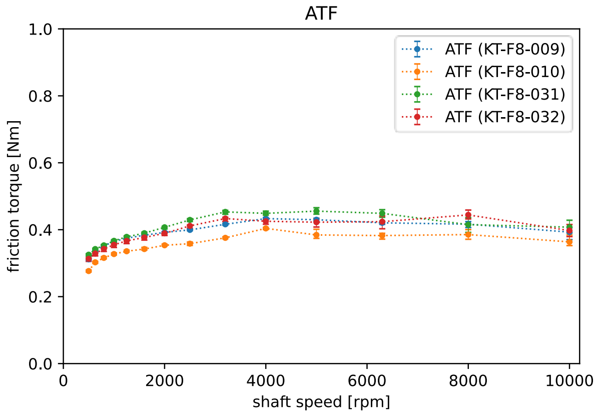

Figure 7: Friction torque over shaft speed using ATF as lubricant

However, there are lubricants that show significant deviations between the individual test runs, such as the lubricant PG1 shown in Figure 8. Lubricants with a significant deviation between the individual test runs often also show remarkable fluctuations of the friction torque in the averaging period of each speed step. This is indicated by the scatter bars in in Figure 8. Both, the deviations between the individual test runs and the fluctuation during each averaging period might be an indicator, that the sealing system is not optimally lubricated.

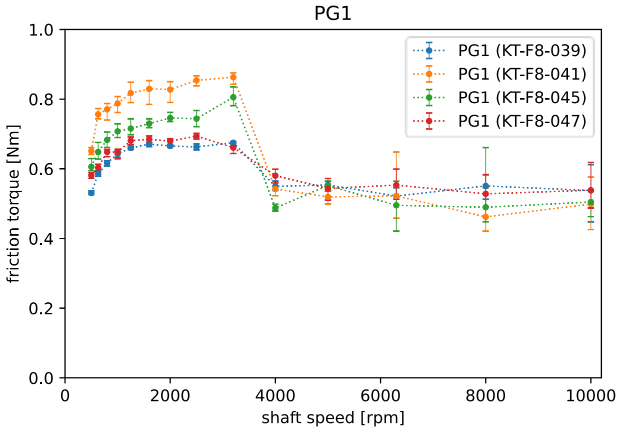

Figure 8: Friction torque over shaft speed using PG1 as lubricant

The friction torque of all lubricants is shown in Figure 9. The friction torque degressively increases for most of the lubricants with increasing shaft speed up to a shaft speed of approximately 3200 to 4000 rpm. At higher shaft speeds, the friction torque remains nearly constant (e.g. automotive lubricants), decreases slightly (e.g. mineral oils) or decreases significantly (e.g. silicone oils). The lubricant PG180 is an exception: With this lubricant, the friction torque considerably increases in the speed range from 6000 to 10000 rpm. The highest friction torques are obtained with the silicone oil SI100, the lowest with the 0W-20 engine oil.

Influence of viscosity on friction torque

Comparing the three mineral reference oils FVA1, FVA2 and FVA3, which only differ in their viscosity, the friction torque increases with increasing viscosity. However, the even more viscous CLP220 lubricant shows the lowest friction torque of all mineral oils (up to a shaft speed of 4000 rpm). Another evidence, that the lubricants' viscosity is not the only factor determining the friction torque is found for the synthetic ester-based lubricants: The lubricants E150-1 and E150-2 are both of the same viscosity grade with nearly the same viscosity-temperature behaviour. However, they significantly differ in their friction behaviour. The average friction torque obtained with E150-2 is about 25 % higher than those obtained with E150-1. This reveals that also other factors (in addition to the viscosity) influence the friction behaviour of the sealing systems – one of these might be additives within the lubricants.

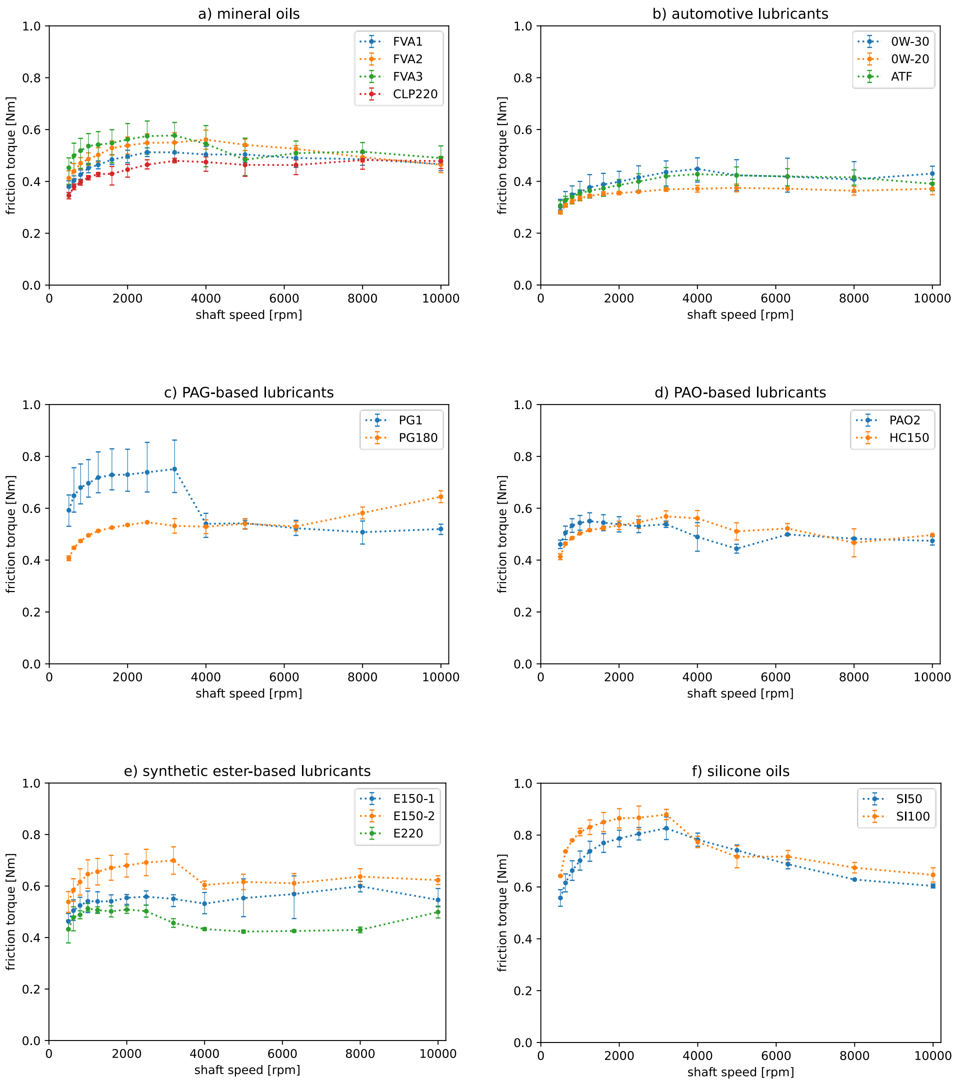

Figure 9: Friction torque of the lubricants as a function of shaft speed

Considering all 16 lubricants, the influence of the dynamic viscosity on the friction torque was systematically analysed. For each of the 14 speed steps, the correlation of the averaged friction torque with the dynamic viscosity at 8 temperatures between 40 and 200 °C was analysed by calculating the coefficient of determination R^2. The highest values for R^2 and thus good correlations are obtained for a shaft speed of 10000 rpm and temperatures of 100 °C or higher for the dynamic viscosity. However, this correlation seems to be rather random since R^2 is much lower at the next lower shaft speed of 8000 rpm. In contrast, the second region of acceptable correlations seems to be much more expanded in the speed and temperature range and thus more reliable. This region is found in the range of lower shaft speeds (between 630 and 2500 rpm) and temperatures around 160 °C where R^2 is in a range between 0.42 and 0.45. Figure 11 and Figure 12 show two exemplary correlations.

Temperature [°C] for dynamic viscosity	Shaft speed [rpm] 500	630	800	1000	1250	1600	2000	2500	3200	4000	5000	6300	8000	10000
20	0.01	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.01	0.01	0.01	0.08
40	0.01	0.01	0.00	0.01	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.04	0.17
60	0.08	0.09	0.08	0.08	0.07	0.05	0.05	0.05	0.02	0.01	0.03	0.04	0.12	0.34
80	0.20	0.23	0.22	0.22	0.21	0.18	0.18	0.17	0.13	0.10	0.13	0.15	0.24	0.50
100	0.31	0.35	0.34	0.35	0.33	0.31	0.31	0.30	0.25	0.22	0.24	0.27	0.33	0.58
120	0.36	0.41	0.40	0.41	0.41	0.39	0.39	0.38	0.33	0.31	0.33	0.35	0.37	0.61
140	0.38	0.43	0.43	0.44	0.44	0.42	0.43	0.42	0.37	0.37	0.38	0.40	0.39	0.60
160	0.38	0.44	0.43	0.45	0.45	0.44	0.45	0.44	0.39	0.40	0.41	0.42	0.40	0.59
180	0.37	0.43	0.43	0.45	0.45	0.44	0.45	0.44	0.40	0.42	0.42	0.44	0.41	0.58
200	0.36	0.42	0.42	0.44	0.44	0.44	0.45	0.44	0.40	0.43	0.43	0.44	0.41	0.57

Figure 10: Coefficient of determination R^2 for the correlation of friction torque at different shaft speeds and the dynamic viscosity at different temperatures

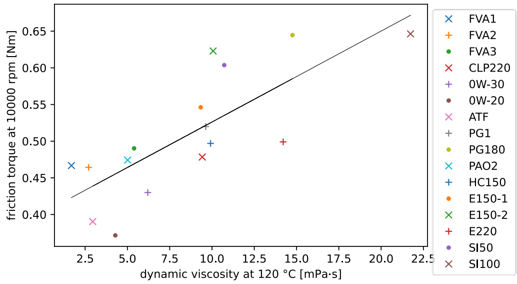

Figure 11: Correlation of friction torque at 10000 rpm and dynamic viscosity at 120 °C

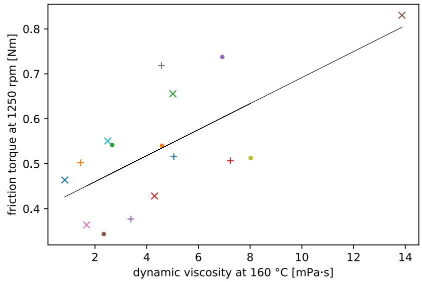

Figure 12: Correlation of friction torque at 1250 rpm and dynamic viscosity at 160 °C

The correlation analysis shows, that lubricants with a high dynamic viscosity in the expected range of the contact temperature tend to cause high friction torques. It is reasonable that better correlations are observed when using the viscosity at a higher temperature range, since the contact temperature was in this range during the test runs. However, there is still no explanation why the correlation is better in some speed ranges than in others.

3.2 Heat transfer/ thermal resistance

The frictional heat flux $\dot{Q}$ generated in the contact area equals the power loss P_R of the sealing system and is the product of the friction torque M_R and the angular velocity which is the shaft speed n multiplied by 2π.

$$\dot{Q} = P_R = M_R \cdot 2\pi\, n \tag{2}$$

Assuming a relatively constant friction torque (for one lubricant), the heat generation is proportional to the shaft speed.

In order to transfer the frictional heat from the contact area to the environment, there must be a temperature difference $\Delta\vartheta$ between the contact area (with temperature ϑ_c) and the oil sump (with temperature ϑ_0) as the most important part of the environment.

$$\Delta\vartheta = \vartheta_c - \vartheta_0 \tag{3}$$

According to Fourier's law, this temperature difference is proportional to the heat flux $\dot{Q}$ which has to be transferred and which corresponds to the power loss P_R.

$$\Delta\vartheta \sim \dot{Q} \quad \text{or} \quad \Delta\vartheta \sim P_R \tag{4}$$

In analogy to Ohm's law in electricity, a proportionality factor between the temperature/ voltage gradient and the heat/ electrical flux is defined: The thermal resistance R_0 is defined as the ratio of the temperature difference $\Delta\vartheta$ and the specific power loss (which is the absolute power loss P_R related to the shaft circumference πd).

$$R_0 = \frac{\Delta\vartheta}{\frac{P_R}{\pi d}} = \frac{\vartheta_c - \vartheta_0}{\frac{P_R}{\pi d}} = \frac{\vartheta_c - \vartheta_0}{\frac{M_R \cdot 2n}{d}} \tag{5}$$

The thermal resistance describes the capability of the sealing system to transfer the generated heat from the contact area away to the environment. It is a parameter of the sealing system, its operating parameters (e.g. oil fill level) and its environment. In contrast to the temperature difference $\Delta\vartheta$, the thermal resistance R_0 can be used to describe the heat transfer independent of the heat generation. Thus, the thermal resistance allows to compare sealing systems regarding the heat transfer only. Low values for R_0 indicate a good heat transfer, high values indicate a restricted heat transfer to the environment.

Figure 13 shows the temperature difference measured during the test run KT-F8-071 using 0W-30 as lubricant. As expected, the temperature difference (related to the oil sump) linearly increases with the shaft speed and the generated heat, respectively. The thermal resistance, however, is nearly constant over the entire range of shaft speed.

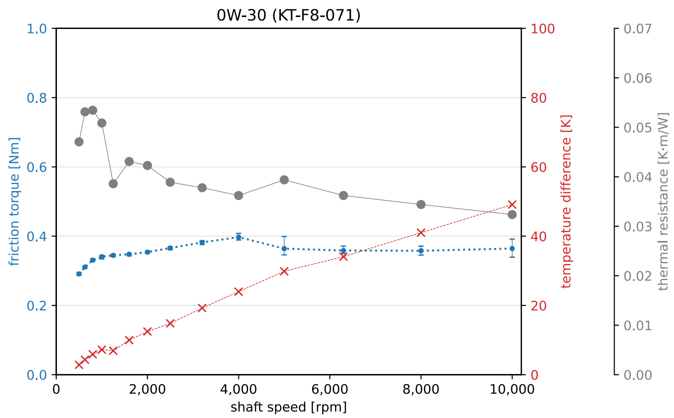

Figure 13: Temperature difference and thermal resistance as a function of shaft speed for test run KT-F8-071 using 0W-30 as lubricant

Figure 14 shows the thermal resistance as a function of the shaft speed for the different lubricants. In general, the thermal resistance decreases degressively with increasing shaft speed. At a shaft speed of 2500 rpm, the thermal resistance is in the range of 0.36 to 0.46 K·m/W. At a shaft speed of 8000 rpm, the thermal resistance is between 0.27 and 0.38 K·m/W. With increasing shaft speed, the heat transfer from the shaft to the oil sump becomes better and thus the thermal resistance decreases. The thermal resistance depends on the used test setup. For other test setups or real applications, the thermal resistance may vary from these values.

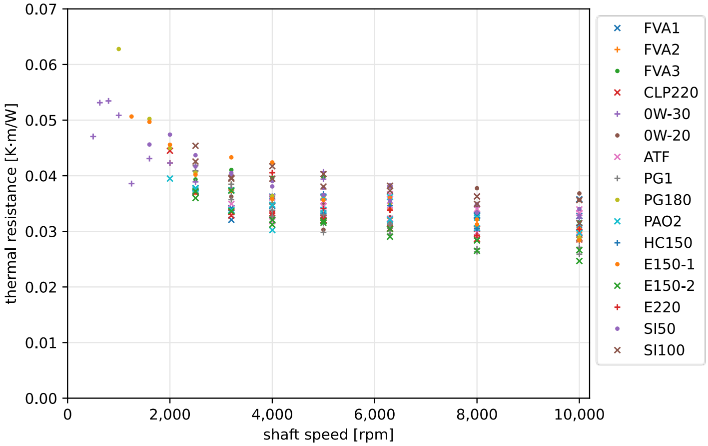

Figure 14: Thermal resistance for various lubricants as a function of shaft speed

For a closer examination, the shaft speed of 5000 rpm was chosen exemplary, Figure 15. At this shaft speed, the averaged thermal resistance per lubricant is between 0.031 and 0.040 K·m/W. Since the individual lubricants deviate from the average of all lubricants (0.035 K·m/W) by less than 15 %, the influence of the lubricant on the heat transfer can be assumed to be negligible.

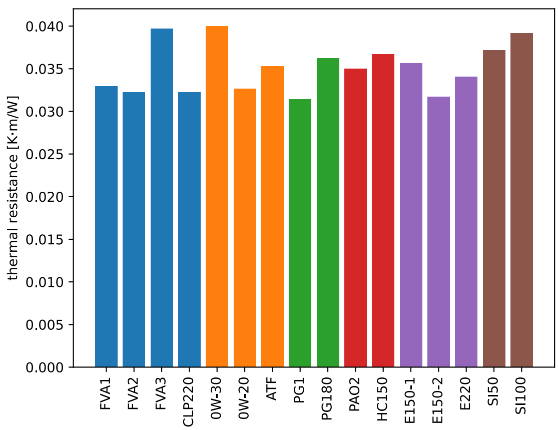

Figure 15: Thermal resistance at 5000 rpm (average of all test runs per lubricant)

4 Summary and conclusion

The thermal behaviour of rotary shaft seals is affected by the fluid which has to be sealed. The fluid lubricates the sealing contact and determines the frictional heat generated there. Additionally, the fluid may affect the heat transfer to the environment.

Both aspects were experimentally analysed by measuring the friction torque and the temperature on the air side near the sealing contact during test runs with 16 different lubricants. The lubricants contained mineral oils with and without additives, automotive oils (engine oils and ATF), as well as lubricants based on polyalkylene glycol (PAG), polyalphaolefins (PAO) and synthetic esters. The test runs were performed with seal rings made of fluororubber and plunge ground shafts (80 mm diameter).

The measurement results show, that the lubricant strongly affects the heat generation. The highest friction torques are obtained with the silicone oil SI100, the lowest with the 0W-20 engine oil. The friction torque obtained with SI100 is more than twice of the torque obtained with 0W-20. In general, the friction torque is higher for more viscous fluids. However, there are several exceptions that restrict this finding. In addition, there are further factors influencing the friction behaviour of the sealing systems – these might be additives within the lubricants and the wetting behaviour.

In order to analyse the aspect of heat transfer separated from the aspect of heat generation, the so-called thermal resistance is defined. The thermal resistance describes the capability of the sealing system to transfer the generated heat from the contact area to the environment. The thermal resistance progressively decreases with increasing shaft speed due to a better heat transfer from the shaft to the oil sump. At a shaft speed of 5,000 rpm, the thermal resistance is in the range of 0.031 to 0.040 K·m/W for the used test setup. The thermal resistance of each lubricant deviates by less than 15 % from the average of all lubricants. Thus, the influence of the lubricant on the heat transfer is much lower than the one on heat generation, where the friction torques vary by a factor of two.

The results obtained in this study will be used to extend the web application "InsECT". With the extended tool, engineers will be able to better consider the fluid to be sealed and to estimate the contact temperature of rotary shaft seals more accurately.

5 Acknowledgements

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FKM Forschung im VDMA

6 Nomenclature

Variable	*Description*	*Unit*
d	Shaft diameter	[mm]
M_R	Friction torque	[Nm]
n	Shaft speed	[rpm]
P_R	Power loss in the sealing contact due to friction	[W]
$\dot{Q}$	Heat flux	[W]
R_0	Thermal resistance	[K·m/W]
η	Dynamic viscosity	[Pa·s]
λ	Thermal conductivity	[W/(m·K)]
ν	Kinematic viscosity	[mm²/s]
ρ	Density	[kg/m^3]
ϑ	Temperature	[°C]
ϑ_0	Temperature in the oil sump	[°C]
ϑ_c	Temperature in the sealing contact	[°C]
$\Delta\vartheta$	Temperature difference (between oil sump and sealing contact)	[K]

7 References

[1] Müller, H. K.; Nau, B. Fluid Sealing Technology - Principles and Applications. New York : Dekker, 1998.

[2] Horve, L.: Shaft Seals for Dynamic Applications. New York : Dekker, 1996.

[3] Bauer, F.: Tribologie. prägnant und praxisrelevant. Springer Vieweg, 2021. ISBN 978-3-658-32919-8.

[4] DIN 3760. Rotary shaft lip type seals. September 1996. (in German)

[5] DIN 3761. Rotary shaft lip type seals for automobiles. 15 Parts. November 1983 to January 1984. (in German, withdrawn in March 2017)

[6] ISO 6194. Rotary shaft lip-type seals incorporating elastomeric sealing elements. September 2007.

[7] Feldmeth, S.; Braeuer, P.; Franke, J.; Bauer, F.: Temperature measurement in the sealing contact – Sealing systems are getting smarter. XXII. Dichtungskolloquium. Vulkan. Steinfurt/Münster 9/2021, p. 15-26, ISBN 978-3-8027-2240-0.

[8] Bauer, F.: Federvorgespannte Elastomer-Radial-Wellendichtungen. Grundlagen der Tribologie & Dichtungstechnik, Funktion und Schadensanalyse. Springer Vieweg, 2021. ISBN 978-3-658-32921-1.

[9] Schuler, P.: Einfluss von Grenzflächeneffekten auf den Dichtmechanismus der Radial-Wellendichtung. Ph.D. thesis. University of Stuttgart, Institute of Machine Components (IMA), 2014. http://dx.doi.org/10.18419/opus-4575

[10] Feldmeth, S.; Stoll, M.; Bauer, F.: How to measure the radial load of radial lip seals. Tribologie und Schmierungstechnik, Expert, Vol. 68, 3-4/2021, p. 5-12, ISSN 0724-3472. https://doi.org/10.24053/TuS-2021-0014

[11] Schaeffler Technologies AG & Co. KG: Super Precision Bearings (SP1). November 2019. https://www.schaeffler.com/remotemedien/media/_shared_media/08_media_library/01_publications/schaeffler_2/catalogue_1/downloads_6/sp1_de_en.pdf, visited in July 2022.

[12] VDI-Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (Hrsg.): VDI-Wärmeatlas. 11. Auflage. Springer Vieweg, 2013. https://doi.org/10.1007/978-3-642-19981-3.

[13] Klüber Lubrication München SE & Co. KG: Viskositätsindex Rechner. Website: https://www.klueber.com/de/de/produkte-service/tools-apps/viskositaetsindex-rechner-1/ visited in May 2022.

[14] Zepf, D.: Experimental analysis of the thermal behaviour of rotary shaft seals. M.Sc. thesis. University of Stuttgart, Institute of Machine Components (IMA), 2022. IMA-2022-31-DT. (in German, expected to be completed in August 2022)

[15] Laukotka, E.: FVA-Heft 660. Referenzöle - Datensammlung. Forschungsvereinigung Antriebstechnik e.V. Frankfurt, 2007.

[16] Fricker, P.; Baumann, M.; Bauer, F.: How different lubricants affect the wear of steel counter-faces in radial lip sealing systems. In: Wear, 2021, S. 203897. DOI: 10.1016/j.wear.2021.203897.

[17] University of Stuttgart, Institute of Machine Components (IMA): InsECT. https://insect.ima.uni-stuttgart.de/

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