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Influence of rolling-element bearings on rotary shaft seals

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Rotary shaft seals are commonly used to seal machines with rotating shafts and prevent the leakage of fluids into the environment. Failures of elastomeric rotary shaft seals are often the result of overheating in the contact area between the sealing edge and the shaft. High temperatures in this contact area occur due to high frictional power and/or inadequate heat dissipation. The surrounding of the seal has a significant impact on the temperatures occurring in the contact area. Especially rolling-element bearings near the seal strongly influence the contact temperature. The frictional heat generated in the bearings results in an increase in the temperature of the fluid near the seal, consequently raising the contact temperature. Tapered roller bearings pump oil. Depending on the installation situation, this can lead to a higher or lower fluid level next to the seal. Tests are conducted on a high-speed friction torque test bench. This investigation involves four different bearing arrangements, as well as a reference variant without bearings. In these test runs, both the frictional torque and the temperature near the contact area are examined. In addition to the experiments on the test bench, Conjugate Heat Transfer (CHT) simulations are conducted with the same installation conditions. The simulated temperatures are compared with the measured temperatures. In most cases, the correlation is good. However, for one bearing arrangement, the correlation is significantly worse. These differences are investigated further.

1 Introduction

Rotary shaft seals and rolling-element bearings are two of the most commonly used machine components. In many technical systems, rotary shaft seals are installed near rolling-element bearings. In this context, they have to seal the fluid (used for lubricating and cooling of the bearing) at the point where a shaft exits a housing [1]. Studies by Kunstfeld [2] have already demonstrated that additional elements near the seal influence the temperature. In this paper, the influence of rolling-element bearings on rotary shaft seals in relation to temperature and frictional torque is analysed further.

1.1 Rotary shaft seals

Rotary shaft seals are commonly used to seal machines with rotating shafts. In the majority of applications, these seals must contain a fluid within a designated space and prevent leakage into the surrounding environment. It is necessary that all components within the tribological system of the rotary shaft seal operate in unison to ensure this function. The tribological system extends beyond the seal ring itself and includes the corresponding counter face (the surface of the shaft), and the sealed fluid (lubricant or operational fluid). The tribological system of a typical rotary shaft seal is seen in Figure 1.

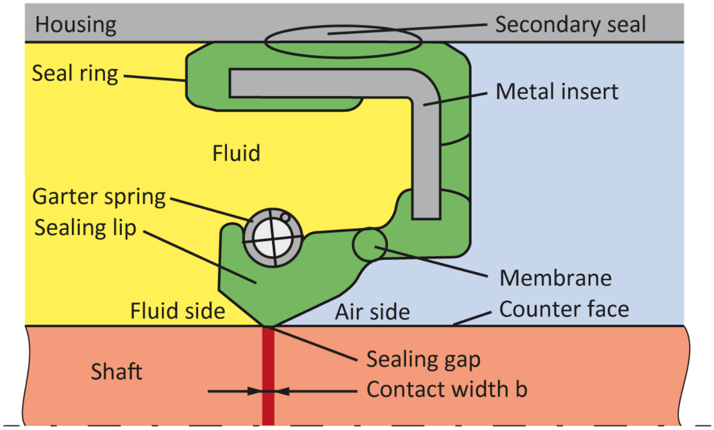


Figure 1: Sealing system of a rotary shaft seal

The structure of a rotary shaft seal is defined in ISO 6194-1 [3] and ISO 6194-2 [4]. The seal ring consists of a metal insert to which an elastomeric sealing lip is moulded.

The area, where the seal ring touches the shaft is called contact area. The contact area typically has a contact width of 0.1 to 0.2 mm. The sealing lip and the garter spring are expanded, when the seal ring is mounted on a shaft. This results in an average contact pressure of approximately 1 MPa in the contact area. When the shaft rotates, elasto-hydrodynamic processes create a thin lubricating film (<1 µm). This film lifts the sealing lip, and is commonly referred to as the sealing gap.

The rotation of the shaft in combination with the asymmetric distribution of pressure within the sealing gap creates a pumping effect from the air side towards the oil side. This pumping effect can be explained by the distortion principle and wiping edge principle. Any fluid that gets to the air side is pumped back to the fluid side retaining the fluid within the system and preventing leakage [1].

1.2 Rolling-element bearings

In most cases rolling-element bearings are used to support rotating components like shafts. These bearings are designed to facilitate smooth and efficient motion by minimizing friction between moving parts. Unlike plain bearings that rely on sliding contact, rolling-element bearings use small, rolling elements (such as balls or cylindrical rollers) that reduce friction and wear. The basic structure of a rolling-element bearing typically consists of an inner race, an outer race, and the rolling elements themselves. The inner and outer races are the ring-shaped structures that house the rolling elements. A cage ensures the proper alignment and movement of the rolling elements. The rolling elements, positioned between the races, distribute the load and enable the smooth rotation of the bearing [5].

Rolling-element bearings come in various types, each tailored to specific applications and load requirements. They can be split into two general types, ball bearings and roller bearings [6]. Three of the most common types are deep groove ball bearings, cylindrical roller bearings and tapered roller bearings. In this paper the influence of these three types on rotary shaft seals is analysed further.

1.3 Contact temperature

Overheating in the contact area between the lip of the rotary shaft seal and the shaft can lead to failures of the seal. The increased temperatures in the contact area are typically the result of high frictional power and/or inadequate heat transfer out of the contact area [7]. One of the typical failure modes of a rotary shaft seal is the hardening of the sealing lip, which is usually the result of excessive temperatures. Hardening prevents the distortion of the sealing edge and thus disables the pumping effect via the distortion principle [1]. Hardening of the sealing lip in combination with increased mechanical stress, for example by an excentric shaft, can result in the development of axial cracks. Elevated temperatures near the sealing lip can also induce fluid decomposition. This decomposition often results in oil carbon deposits in or near the contact area. The accumulated oil carbon can disturb the sealing mechanism, ultimately leading to leakage. [1, 8].

A failure of a rotary shaft seal can be both ecologically and economically significant. Information about the resulting contact temperature is crucial to avoid overheating. This awareness plays a pivotal role in preventing failures, enhancing product reliability, and prolonging the lifespan of the seal.

To choose an appropriate seal for a particular application, an understanding of the potential contact temperatures in later use is essential. An increase in temperature of about 10 K can half the anticipated service life [9]. To determine the potential contact temperature there are three different approaches: The measurement on a prototype, the simulation of the entire system or using estimation methods [10]. Measurements on a prototype and Simulations are more accurate. However, it is often necessary to quickly and easily estimate the contact temperature. In these cases it is useful to use less precise but simple estimation methods.

1.4 Frictional torque

In most applications, no special attention is paid to the frictional torque of seals. The frictional torque of a single seal of usually less than one newton meter appears very low compared to other components of a typical technical system. However, the friction in the contact area directly influences the contact temperature. Especially in combination with high rotational speeds, minimizing friction and thus frictional power is advantageous. In addition to the power losses at the seal, significant losses also occur at the rolling-element bearings. This power loss heats the oil and the shaft additionally. Some rolling-element bearings pump fluid in axial direction [11]. Hence, the bearings have a significant impact on the operating conditions of the seal. They influence not only the temperature but also the oil level and potentially the pressure of the oil near the seal. The altered operating conditions affect both the heat generation in and transfer out of the contact area.

Exposure of a rotary shaft seal to a higher pressure on the fluid side than on the air side leads to an increase in the radial load on the sealing lip. Consequently, this enlarges the contact area between the sealing lip and the shaft, leading to additional friction and increased contact temperatures and wear [12].

2 Setup

All tests are conducted on a high speed universal friction torque test bench [13]. The basic design of the test bench consists of a motor spindle and a test chamber in which the seal ring is mounted. The motor spindle enables speeds of up to 24000 rpm. The test shaft is mounted on the motor spindle through a hollow shaft taper (HSK). The test chamber is supported by an aerostatic bearing. Due to the aerostatic bearing, the chamber can rotate almost frictionless around its centre axis. The chamber is supported by a lever arm on a force sensor. The torque at the seal is transferred to the chamber and to the force sensor. The frictional torque is calculated by multiplying the length of the lever arm and the measured force. The frictional torque is recorded by the measuring computer.

Figure 2 shows the design of the high speed universal friction torque test bench.

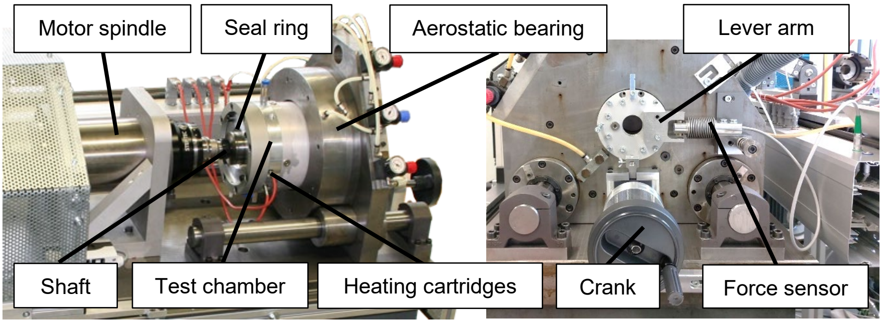


Figure 2: Setup high speed universal friction torque test bench [13, 14]

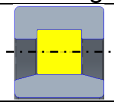
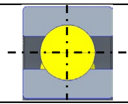
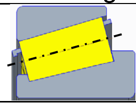
Before the experiment, the test chamber is filled (with the fluid to be examined) up to the centre of the shaft and then closed with a vent for ventilation. Temperature control is achieved through heating cartridges and a temperature sensor, which are installed in the chamber. They are controlled and evaluated by the control software. The test chamber can be moved along linear guides using a crank. This way different positions of the sealing lip can be set on the shaft.

2.1 Tested components

For all experiments, rotary shaft seals according to ISO 6194 [3] with dimensions of 100x80x10 are used. The seal rings are made out of fluoro rubber.

Four different rolling-element bearing arrangements are investigated. The bearing next to the seal is varied between a ball bearing, a cylindrical roller bearing, a tapered roller bearing in an X arrangement, and a tapered roller bearing in an O arrangement. An overview of the used bearings is provided in Table 1.

Table 1: Used rolling-element bearings.

	Rolling-element bearing		
Type	Cylindrical roller bearing	Ball bearing	Tapered roller bearing
Cross section			
Inner diameter d [mm]	85	85	85
Outer diameter D [mm]	130	130	130
Width B [mm]	22	22	29
Max speed in test [rpm]	6.300	6.300	3.200

To mount the rolling-element bearings in the test chamber, a new mounting structure was designed and manufactured. The adapted test chamber is seen in Figure 3.

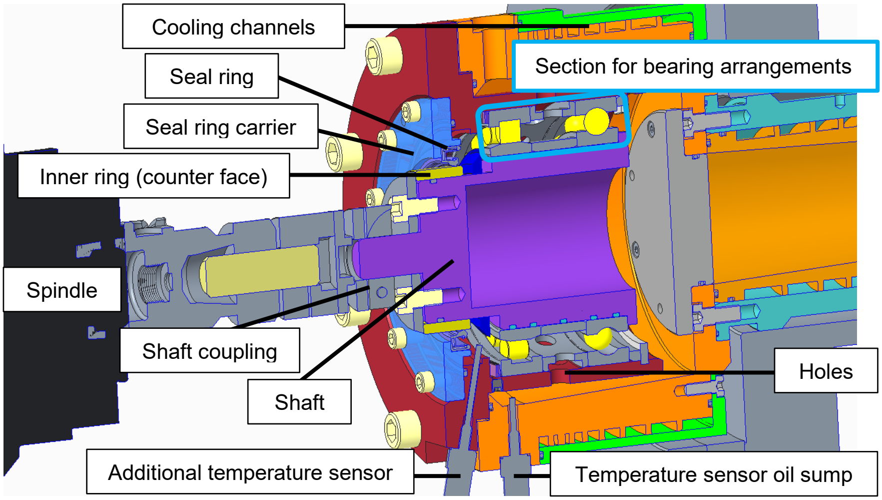
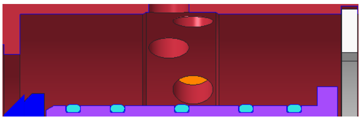
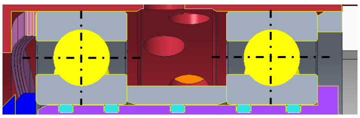
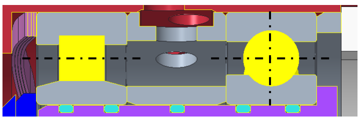
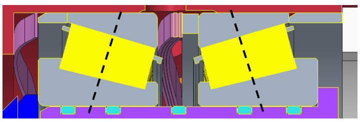
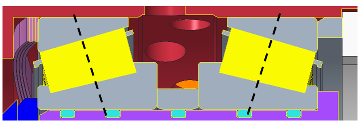


Figure 3: Adapted test chamber to mount rolling-element bearings

The bearings can be installed in to the adapted test chamber. A corresponding shaft body allows for the installation of the bearing inner rings on the shaft. 24 holes with a diameter of 10 mm allow for the circulation of the oil between the bearings.

The shaft counter face of the seal is realized with a plunge gound inner ring of a needle bearing. A seal ring carrier made out of transparent PMMA enables the visual observation of the oil flow near the seal. An additional temperature sensor is installed in the gap between the bearing and the seal. This sensor measures the oil temperature near the seal. In the case of the bearing variants, the shaft body is supported by the bearings installed in the test chamber. For these variants, a coupling is used on the air side to prevent the bearings from binding (misalignment to the motor spindle). The coupling can compensate for any misalignment that may occur. The four different rolling-element bearing arrangements, as well as a reference Variant without bearings are shown in Table 2.

Table 2: Tested arrangements.

Abbreviation	**Meaning / Explanation**	**Picture**
Ref-SV	Reference variant for surrounding variation: Variant without bearings as a reference to investigate a single seal.	
Ba	Ball bearing: Deep groove ball bearing next to the seal (Bearing: 6017)	
Cyl	Cylindrical roller bearing: Cylindrical roller bearing next to the seal (Bearing: NU1017-XL-M1 + 6017)	
O	O arrangement Tapered roller bearing in an O arrangement next to the seal (Bearing: 32017 X)	
X	X arrangement Tapered roller bearing in an X arrangement next to the seal (Bearing: 32017 X)	

2.2 Testing procedure

All tests are conducted with a new seal ring and counter face. Before the tests, the chamber is filled with oil up to the centre of the shaft. This is a deviation from the typical setup for bearing tests as described in ISO 15312, where the oil level only reaches the centre of the rolling element in the lowest position [15]. The oil sump temperature is set to 80°C. Due to the frictional power loss in the bearings all bearing variants were tested with an actively cooled test chamber. The tests follow a specific load collective [16], which consists of two phases. The first phase is a preconditioning phase during which the shaft rotates with a speed of 1000 rpm. This phase ends after twelve hours. After the preconditioning phase the second phase starts. During this phase all the measurements are conducted. The rotational speed is varied in this phase. Each speed is held for thirty minutes and increased by about twenty percent after each speed, starting at 500 rpm and ending at 6300 rpm. Figure 4 shows the load collective. The variants with tapered roller bearings (O and X) are only run up to 3200 rpm due to speed limitations of the bearings.

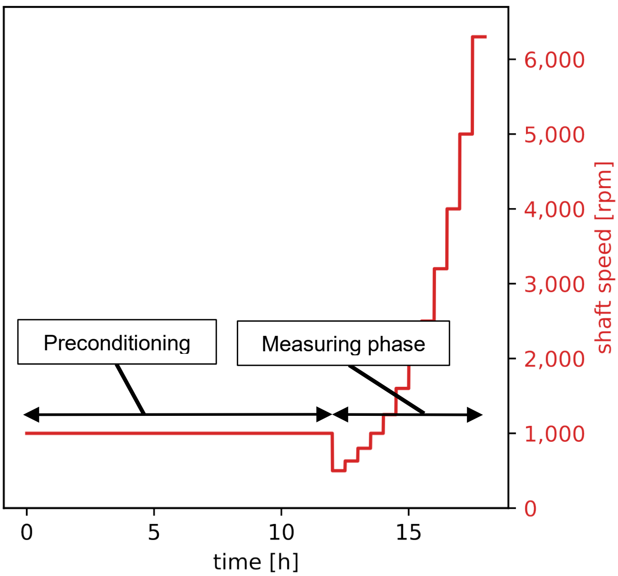


Figure 4: Load collective

During the whole test the frictional torque is measured and documented by the test bench. At the end of each speed interval the Temperature at the airside near the sealing contact is measured with a thermal imager (Fluke Ti480-Pro; accuracy: ±2 °C or ±2 % [17]; emissivity = 0.95, transmission = 100 %). For the sake of simplicity, this temperature is referred to as the contact temperature in the following, even if the temperature in the direct contact area between the sealing lip and the shaft can't be measured.

3 Results

Figure 5 shows the measured frictional torque of all variants over the rotational speed. For each variant at least two individual test runs are conducted. The scatter bars show the maximum and minimum frictional torques of the individual test runs at the respective shaft speeds for each variant.

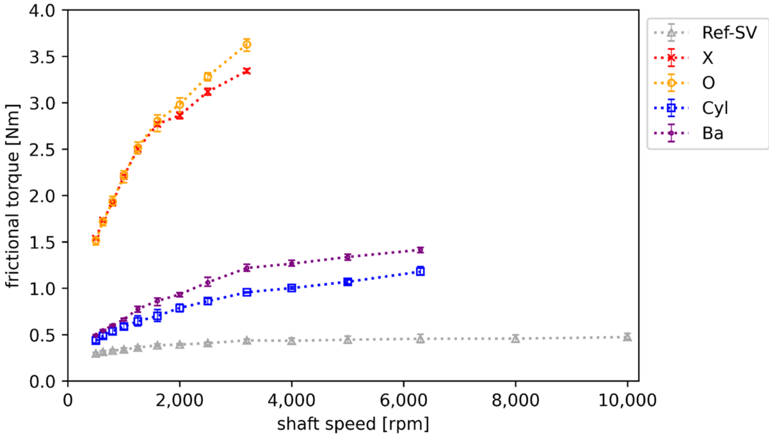


Figure 5: Torque over shaft speed

The reference variant (Ref-SV) shows the lowest frictional torque. The bearing variants show a trend of increasing frictional torques with speed, which flattens with higher speeds. Of the variants with bearings the cylindrical roller bearings (Cyl) has the lowest frictional torque. Slightly higher frictional torques are measured with the deep groove ball bearings (Ba). The largest frictional torques occur with the tapered roller bearings. In this case, the X arrangement (X) shows a slightly flatter trend at higher speeds compared to the O arrangement (O). As expected the tapered roller bearings show a strong pumping behaviour from the side of the smaller diameter to the side of the larger diameter, significantly influencing the oil fill level near the seal. Figure 6 shows the resulting oil levels at the lowest speed of 500 rpm for the X and O arrangements.

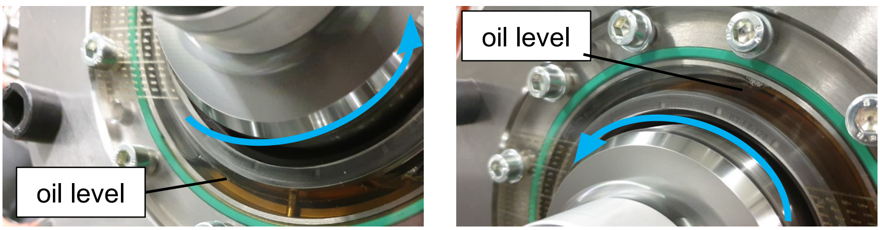


Figure 6: Oil levels 500 rpm for the X (left) and O (right) arrangement [18]

Already at the lowest speed of 500 rpm, the X arrangement shows a fill level below the lowest point of the sealing edge. In the O arrangement, the space between the bearing and the seal is almost completely filled with oil. This becomes even more apparent with increasing speeds. The other two bearing arrangements, as well as the reference variant of the surroundings variation (Ref-SV), show no pumping effect. Figure 7 shows the Temperature at the sealing edge over the shaft speed.

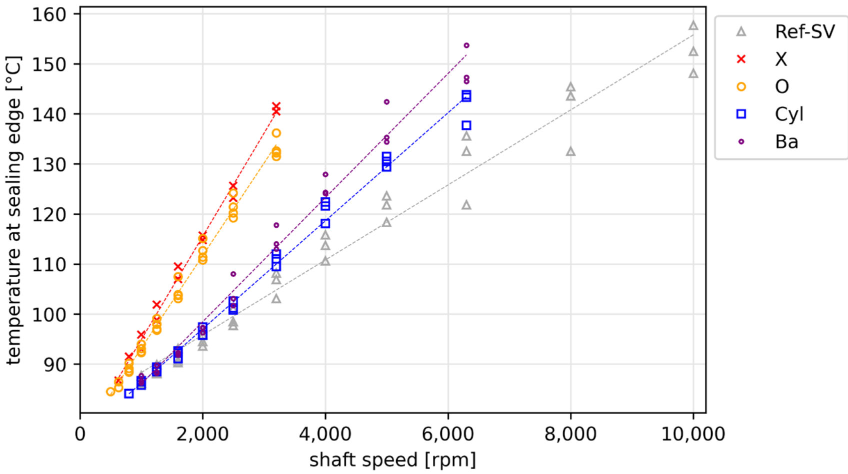


Figure 7: Temperature at the sealing edge over shaft speed

The highest temperatures occur at the X arrangement closely followed by the O arrangement. The variants with the cylindrical roller bearing (Cyl) and the deep groove ball bearings (Ba) show lower temperatures. The lowest temperatures occur without bearings (Ref-SV).

The temperature of the oil near the seal is relevant for the heat transfer out of the contact area. The temperature difference between the oil near the seal (additional sensor, see Figure 3) and the oil sump at 80 °C is shown in Figure 8. For all variants the temperature difference increases nearly linear with the shaft speed. The variants with bearings show higher temperatures of the oil near the seal than the reference variant. the O arrangement (highest torque) has the highest temperature difference.

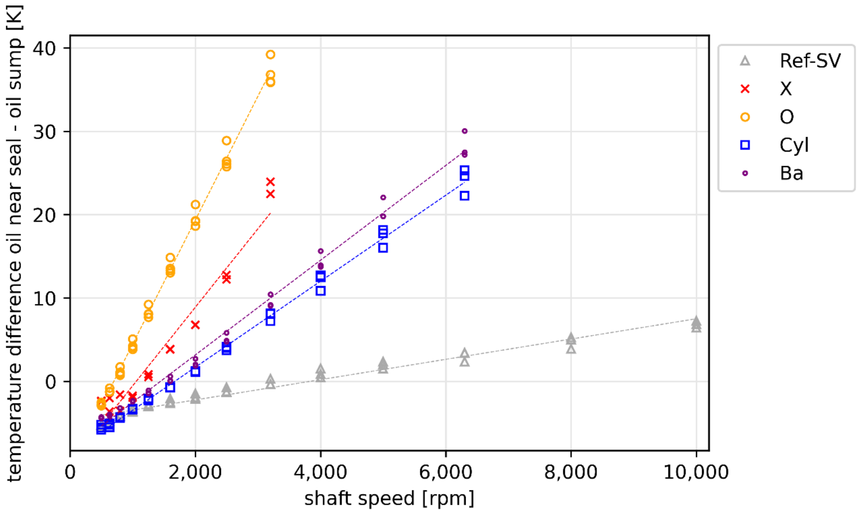


Figure 8: Temperature difference between the oil near the seal and the oil sump over shaft speed

In Figure 9 the temperature difference between the sealing edge and the oil near the seal is plotted over the shaft speed.

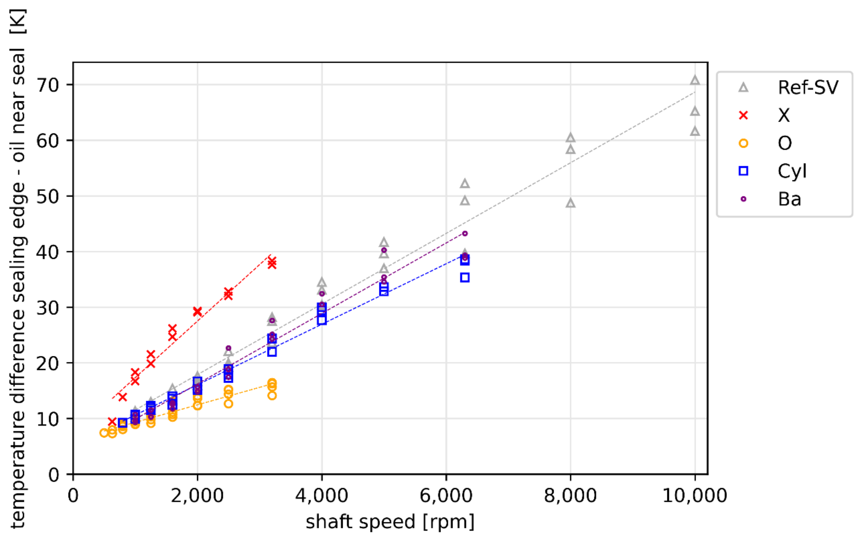


Figure 9: Temperature difference between the sealing edge and the oil near the seal over shaft speed

The highest temperature differences between the sealing edge and the oil near the seal occur in the X arrangement (X). Due to the low oil level near the seal, the heat generated in the contact area cannot be transferred out of the contact area effectively, leading to particularly high temperature differences and the highest contact temperatures. The O arrangement (O) has the second highest contact temperatures, however the lowest temperature differences to the oil near the seal occur at this variant. Due to the high oil level near the seal, the heat generated in the contact area can be transferred out of the contact area more effectively, resulting in lower temperature differences between the sealing edge and the oil near the seal. The variants with no pumping effect (Ref-SV, Cyl and Ba) show very similar temperature trends with respect to speed. The temperature differences are slightly lower for the cylindrical roller bearing (Cyl) compared to the deep groove ball bearing (Ba) and the reference variant (Ref-SV).

4 Simulation of the contact temperature

In addition to the experiments on the test bench, the different rolling-element bearing variants were simulated [19]. The numerical fluid flow simulations were conducted in Ansys CFX 2021 R2. Due to the heat exchange between solid and liquid domains in the model, the Conjugate Heat Transfer (CHT) method was used. This method allows the calculation of temperature distributions in both the solid and fluid domains. Due to the interaction of oil and air in the oil chamber, a multiphase flow approach was chosen for this area. The simulation closely matches the results from the temperature measurements (difference < 5 K) at the test bench except for the O arrangement (max. difference of 26 K). It is suspected that the higher temperatures of the measurements in comparison to the simulation is due to a pressure build up in the area between the bearing and the seal. This would lead to increased contact pressure in the sealing contact and higher frictional torque at the seal, explaining the difference to the simulation without increased torque.

5 Additional Measurements

In order to further investigate the pressure build up in the area between the bearing and the seal of the O arrangement additional tests are conducted. Instead of the second temperature sensor the space between the bearing and the seal is connected to a vertical tube. Using the principle of a liquid column manometer the pressure is determined by the difference in oil level in the tube. The setup is shown in Figure 10.

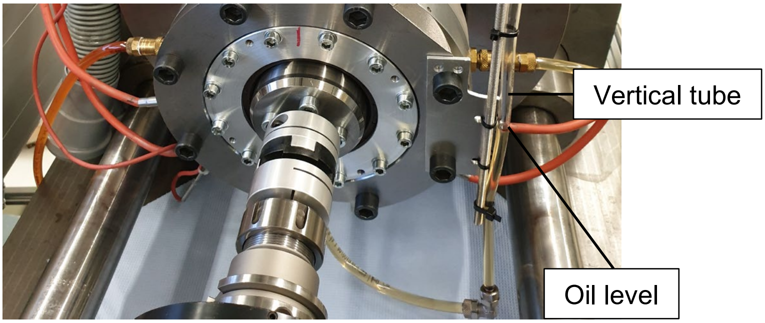


Figure 10: Setup for pressure measurement

The pressure was measured in two test runs. The pressures at different shaft speeds can be seen in Figure 11.

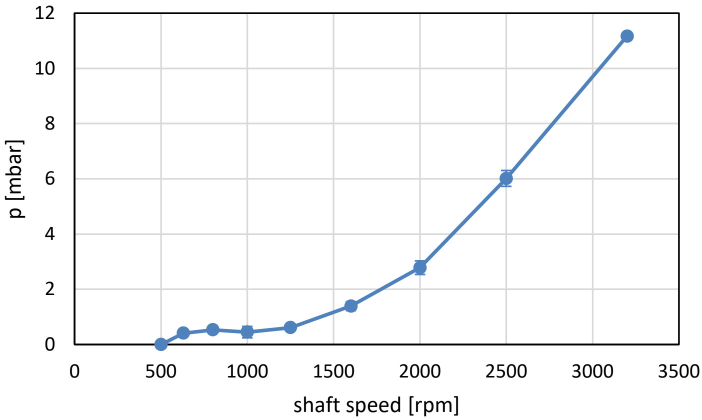


Figure 11: Pressure in area between seal and bearing over shaft speed

The pressure increases with the shaft speed. For speeds over 1500 rpm there is a significant increase in pressure. The radial load of a seal ring increases with pressure. For the tested ring the radial load increases by about 0,08 N per mbar [20]. For the tested seal ring the radial load should increase by about 0,9 N for the highest shaft speed of 6300 rpm. The radial load of a seal at 80 °C (oil sump temperature) is about 28 N. Therefore the radial load increases by about 3 %.

In order to get a better understanding of the frictional torque of the seal under the pressurized condition, additional tests are conducted. In these tests the reference configuration is used and the oil chamber is pressurized. The pressure is set according to the measurements (see Figure 11). The frictional torque of the pressurized and unpressurized test runs is shown in Figure 12.

The pressurized seal (Ref+p) shows higher frictional torque than the unpressurized seal (Ref). For the slower shaft speeds the pressure is very low, thus the frictional torque is similar for these speeds. For the highest shaft speed of 6300 rpm the frictional torque increases by about 4 % compared to the unpressurised variant. The increase in frictional torque (4 %) is very similar to the increase in radial load (3 %)

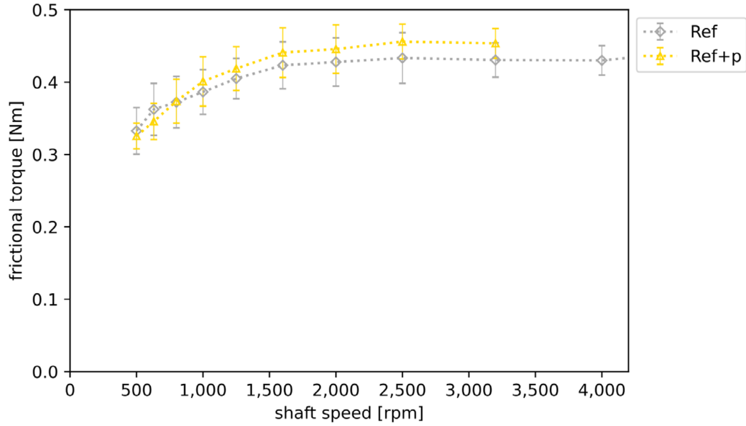


Figure 12: Seal torque over shaft speed with and without pressure

6 Simulation of the O arrangement with measured torque

The simulation of the O arrangement of chapter 4 is updated with the measured torque at the seal. In this simulation the frictional power is calculated with the frictional torque at the respective shaft speed and is applied directly in the contact area instead of the previously used frictional power model (standard seal, no pressure). Figure 13 compares the measurements with the updated simulation results.

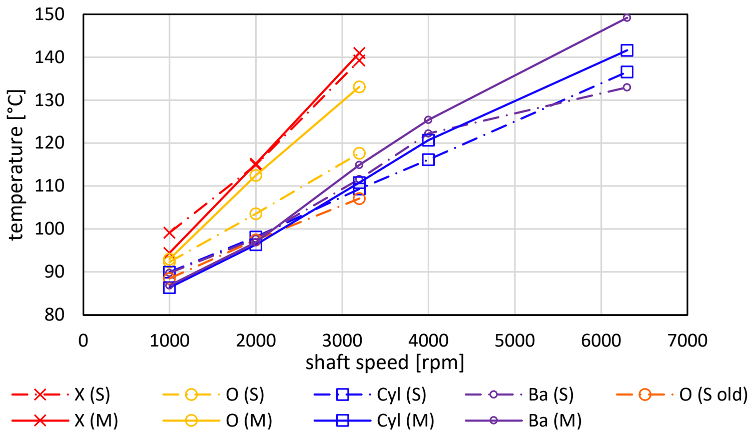


Figure 13: Maximum temperature at the airside of the sealing lip: simulation results (dashed) compared to the measured temperatures (full colored).

In order to get a comparable result from the simulations the maximum temperature at the airside of the sealing lip is compared with the measurements. These simulated results are marked with an (S). The results from the measurements are marked with an (M). The simulation closely matches the results from the temperature measurements at the test bench. The O arrangement O (S) still produces colder simulation results with a maximum deviation of 15 K, however this is still an improvement compared to the deviation of 26 K of the previous simulation O (S old). The frictional torque of the seal ring during the tests with the O arrangement was likely higher. A potential cause for this could be the higher temperature of the oil next to the seal in comparison to the tests with a pressurized single seal (Figure 12) and the different oil levels at the seal (centerline for single seal and flooded for O arrangement).

The simulations depend on the frictional torque of the seal and the bearings. Therefore, measurements on a test bench are still necessary.

7 Conclusion

Rolling-element bearings near the seal strongly influence the contact temperature. The heat generated in the bearings due to friction results in an increase in the temperature of the fluid near the seal, and consequently, an increase in the contact temperature. Tapered roller bearings have the highest torque compared to the other variants and pump oil. Depending on the installation situation the pumping leads to a higher or lower fluid level near the seal. The X arrangement significantly lowers the fluid level in the area near the seal, causing the highest contact temperatures. The O arrangement significantly increases the fluid level and has the highest overall torque (including the bearings and the seal). For higher shaft speeds the O arrangement increases the pressure at the seal leading to higher frictional power at the sealing edge. However, the O arrangement has a lower contact temperature than the X arrangement due to better cooling. The variants with no pumping effect (Ref-SV, Cyl and Ba) show very similar temperature differences to the oil next to the seal, which increase with speed. Since the temperature in the contact area cannot be measured directly, simulations can be used to get an understanding of the temperature in the contact area.

8 Acknowledgements

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