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Influence of Wear on the Frictional Torque of Rotary Shaft Seals

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The tribological system of a rotary shaft seal is subject to friction and wear. In most cases, these aspects are analysed separately. Wear is usually evaluated on endurance test rigs where multiple seals are tested simultaneously over long periods of time (a few days to a few months). After these tests, the wear of the seal ring and shaft can be analysed. This enables the prediction of wear over time for a specific seal under a specific load collective. Friction is usually tested on friction torque test benches, where the frictional torque of the sealing system is analysed under specific conditions. These tests usually last a few hours and contain different operating conditions. Due to the comparatively short time on the friction torque test benches, these seals show almost no wear. In this paper, the influence of wear on the frictional torque of rotary shaft seals is analysed further.

1 Introduction

During the operation of a rotary shaft seal, the entire tribological system is subject to various wear mechanisms. Wear can occur on the sealing lip, the counter face (shaft surface) and the sealed fluid [1].

The sealed fluid acts as a lubricant in the contact area. A change in the lubricant due to tribological stress is described by Kuhn as lubricant wear or rheological wear [2].

In many cases, the counter face shows only minimal wear. If the surface finish of the counter face complies with the values described in [3] and the contact patch is not mirror-polished, wear on the mating surface is not harmful [4].

Wear on the sealing lip of an elastomer rotary shaft seal can result from various wear mechanisms. This wear is frequently caused by fatigue, friction and rolling-in. Material fatigue occurs due to cyclic loading [5]. Friction can cause the elastomer to deform locally beyond its tensile strength and tear. The further propagation of a crack then requires less energy than the crack initiation [5]. A high coefficient of friction can lead to the elastomer rolling in and tearing. The material that has already torn adheres to the shaft until it rolls into the contact surface. This leads to further tearing and ultimately to the roll breaking off [1], [5], [6].

The effect of a worn sealing edge on the frictional torque of the seal will be examined in more detail in section 3.2. Both friction in the sealing contact and wear of the sealing edge have been studied multiple times. The focus in these studies is either on wear, e.g. [7], or on the frictional torque of the seal, e.g. [8] or [9]. Wear is usually investigated on endurance test benches with long test durations, but without frictional torque measurements. The investigation of the frictional torque is usually limited to shorter test durations on friction torque test benches. So far, the relationship between wear of the sealing edge and the frictional torque has not been investigated in detail.

The frictional torque tends to increase with wear. This behaviour is evident in [10] and [11]. The findings are based on experimental and simulated data.

In order to experimentally investigate the influence of wear on the frictional torque, a sealing system must be allowed to wear on a friction torque test bench. Under normal operating conditions, it can take several days to weeks for the desired wear to develop. Such an approach is impractical due to limited test bench capacity. In the event of insufficient lubrication, increased wear occurs at the sealing edge. If the sealing edge runs dry on the shaft, significant wear is visible after just a few minutes. This behaviour has been investigated in preliminary studies, which are discussed in section 3.1.

2 Setup

The used friction torque test bench is shown in Figure 1. The frictional torque of the seal is measured within the shaft.

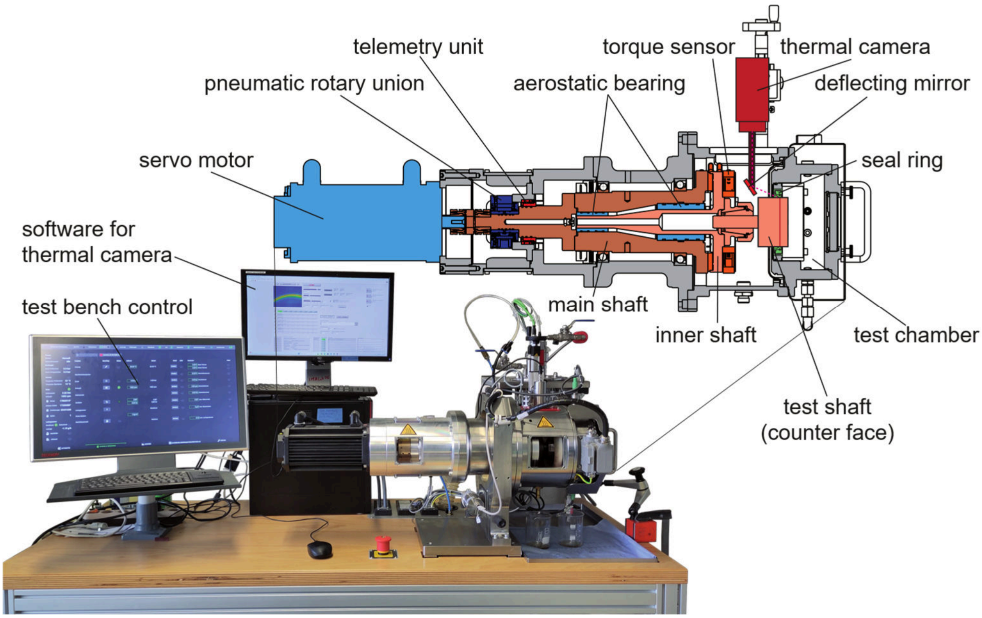

Figure 1: friction torque test bench

The main shaft is driven by a servo motor. The inner shaft is located inside the main shaft. The inner shaft is suspended by an aerostatic bearing within the main shaft. It can therefore rotate nearly without friction. The inner shaft is connected to the main shaft via a torque sensor. The torque is transmitted between the main shaft and the inner shaft solely via the torque sensor and can be measured directly. The test shaft is connected to the inner shaft by a collet. The frictional torque of the seal is thus transmitted to the inner shaft. During operation at a constant speed, the torque applied to the torque sensor equals the frictional torque of the seal. To transmit the measured torque, as well as the compressed air for the aerostatic bearings, a telemetry unit and a pneumatic rotary unit are installed between the main shaft and the fixed structure.

The seal ring is installed in the test chamber and sits on the test shaft. The test chamber is filled with the sealed fluid prior to the tests. The fluid sump is temperature-controlled using heating elements and temperature sensors installed in the test chamber.

A thermal camera, Optris Xi 400 [12], is used to measure the temperature at the air side of the sealing lip. The emissivity is set to 0.95, and the transmission is set to 100 %. The background temperature is measured internally by the camera. Due to the limited space inside the test bench, the optical path of the thermal camera must be redirected using a deflecting mirror. The measurement data from the thermal camera is recorded with a separate software. After the test run, the measurement data from the test bench and the thermal camera are synchronised with a separate program.

2.1 Tested components

For all experiments, rotary shaft seals according to ISO 6194 [13] with dimensions of 100x80x10 are used. The seal rings are made out of fluoro rubber (FKM) and nitrile rubber (NBR).

As counter face of the seal, a plunge ground shaft sleeve is used (inner ring of a needle bearing), which meets the specifications described in [3]. The shaft sleeve is assembled on the main body of the test shaft. The fully assembled shaft can then be mounted in the test bench.

For all tests with lubrication, an engine oil with the specification SAE 0W-30 [14] is used.

2.2 Test conditions for frictional torque measurements

The fluid sump in the test chamber is set to the centre of the shaft. During the test, the fluid sump temperature is maintained at 80 °C. A standard speed profile is used for all tests, in which the shaft speed is varied over time. The speed profile consists of two parts. The first part is a preconditioning phase during which the shaft rotates with a constant speed of 1000 rpm. This phase ends after twelve hours. The second part is the measuring phase, during which all the measurements are conducted. In this phase, the shaft speed is varied. Each speed is held for thirty minutes and increased by about 25 % after each speed step. The measuring phase starts with a speed of 20 rpm and ends with a speed of 3000 rpm. The used speed profile is shown in Figure 2.

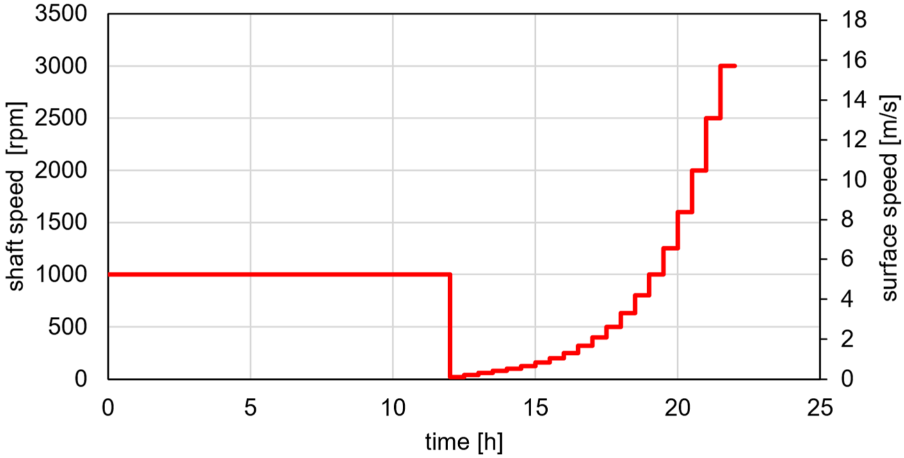

Figure 2: Speed profile

Throughout the entire test, the frictional torque is measured and documented by the test bench. The temperature at the airside near the sealing contact is measured with the thermal camera and documented as well. For further analysis, all measured parameters are averaged over the last ten minutes of each speed step. They can then be compared for the individual shaft speeds.

3 Results

In preliminary studies, the ideal conditions for abrasive wear on seal rings in short time periods are determined. Using this knowledge, wear is created on seal rings prior to frictional torque measurements. The results of the preliminary studies and the measurements are presented in the following.

3.1 Preliminary studies on seal ring wear

In preliminary studies, sealing systems with FKM and NBR seal rings, each with an inner diameter of 80 mm, were examined. The tests were conducted without the sealed fluid and, therefore, without lubrication.

The test shafts are cleaned with acetone prior to assembly. The sealing systems are then run at constant speed for different durations. Following these tests, the wear width of the sealing rings is measured. After the measurement, the sealing system is reassembled, and the test chamber is filled with the engine oil of the specification SAE 0W-30 [14]. The sealing system is then subjected to the test conditions described in section 2.2. After the test, the wear width is measured again. The results of these tests are shown in Table 1.

Table 1: Wear width of seal rings depending on conditions during dry run

conditions during dry run			average wear width [mm]	
material	shaft speed [rpm]	duration [min]	after dry run	after load collective
FKM	3000	30	0.55	0.69
FKM	3000	60	1.01	1.00
FKM	3000	120	1.28	1.30
NBR	1000	120	0.44	0.44
NBR	1000	600	0.98	1.00
NBR	1000	1440	1.41	1.43

The tests show that the wear width changes only slightly during the load collective with oil lubrication. It can therefore be assumed that the wear does not change during the test series and that the wear measured after a test with oil lubrication and prior dry running is equal to the wear resulting from dry running only.

3.2 *Effect of wear on the frictional torque*

The following procedure is used to investigate the effect of wear on the frictional torque. The seal ring and the test shaft are mounted in the friction torque test bench. The seal ring is worn by running the system dry at a constant speed for a specified duration. The test chamber is then filled with oil. For the tests, the test conditions described in section 2.2 are used. Figure 3 shows the frictional torque of sealing systems with FKM seal rings with varying wear widths.

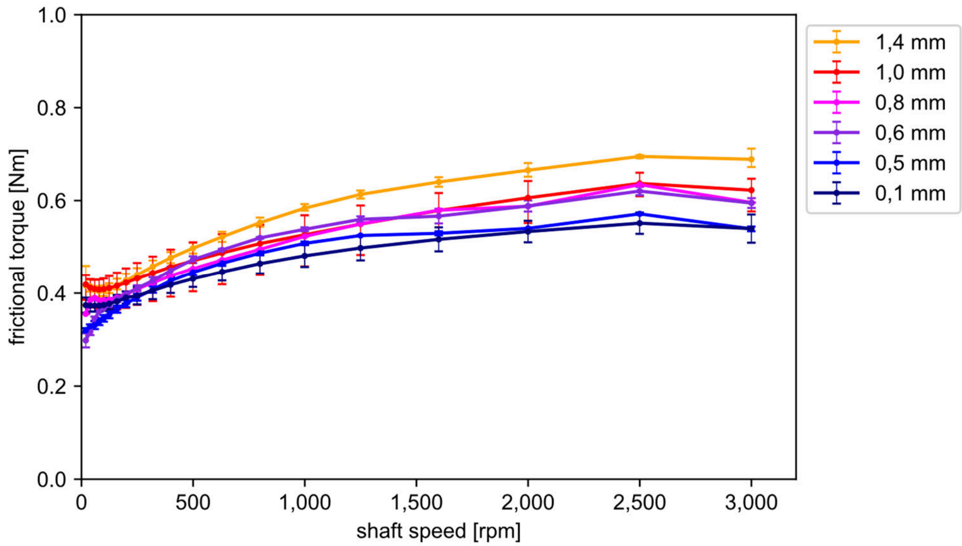

Figure 3: Frictional torque of sealing systems with FKM seal rings with varying wear widths

The seal rings are divided into groups based on their average wear width. Each group consists of at least two tests with different seals. The average torque of each group is shown as a dot. The scattering within each group is shown with scatter bars.

In sealing systems with FKM seal rings, the frictional torque increases with the shaft speed. The increase in frictional torque levels off as the shaft speed increases further. The frictional torque tends to increase with increasing wear. Exceptions to this are the wear widths of 0.5 mm and 0.6 mm, which have the lowest frictional torque at shaft speeds below 200 rpm.

Figure 4 shows the frictional torque of sealing systems with FKM seal rings with varying wear widths.

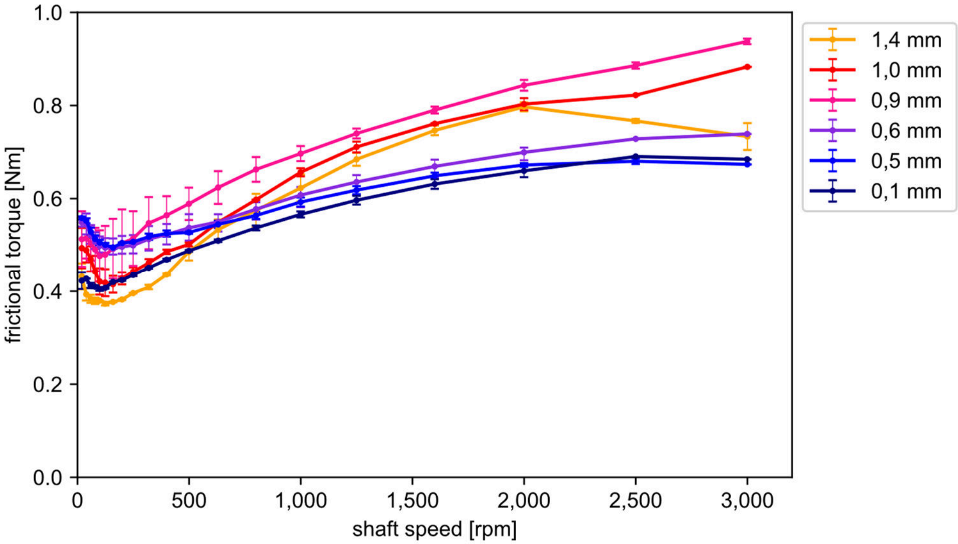

Figure 4: Frictional torque of sealing systems with NBR seal rings with varying wear widths

In the low-speed range below 120 rpm, the frictional torque decreases with increasing shaft speed for sealing systems with NBR seal rings. The lowest frictional torque is reached at around 120 rpm. At higher shaft speeds, the frictional torque increases with the shaft speed. The curve flattens out for higher shaft speeds.

The frictional torque tends to increase with increasing wear. The highest torque is reached at a wear width of approximately 0.9 mm. With further wear, the frictional torque decreases again. This decrease is particularly pronounced at shaft speeds above 2000 rpm.

For shaft speeds of less than 120 rpm, the maximum frictional torque is reached at a wear width of 0.5 mm. As the wear width increases, the frictional torque decreases.

With wear, not only the wear width, but also the inner diameter of the seal ring increases. As a result, the interference between seal ring and shaft as well as the radial load decreases. During the tests on the test bench, the temperature of the seal rings is approximately the same as the oil sump temperature. The radial load of the seal rings is therefore measured at a temperature of 80°C. The measured radial loads are shown in Figure 5. The radial loads are measured after the test using Method D, as described in [15].

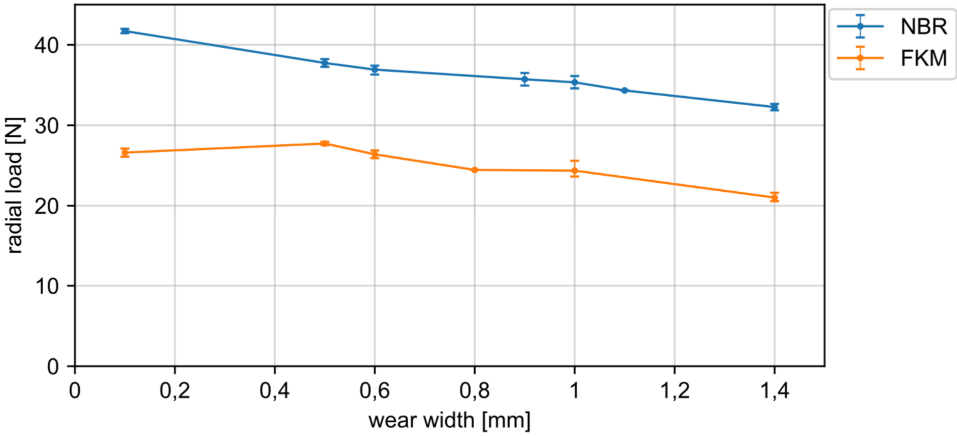

Figure 5: Radial loads at 80°C for different wear widths

The radial load decreases primarily linearly with the wear width. So far, the increase in wear width has been analysed in combination with a reduction in radial load. To examine the effect of the wear width independently of the radial load, the coefficient of friction is used. Figure 6 shows the coefficient of friction for sealing systems with FKM seal rings as a function of surface speed for different wear widths.

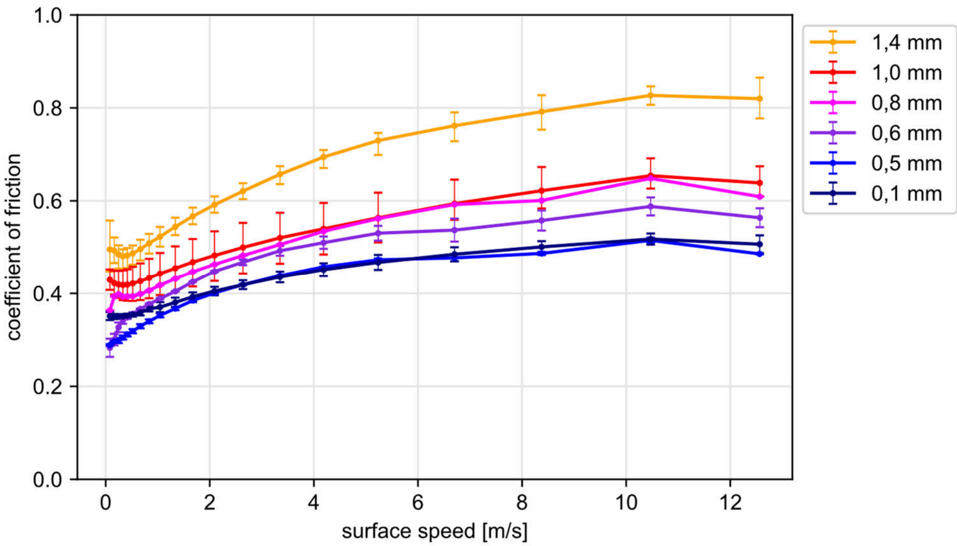

Figure 6: Coefficient of friction in relation to surface speed for FKM seals

As with the frictional torque, the coefficient of friction increases degressively with increasing surface speed. The coefficient of friction also increases as the wear width increases. Exceptions to this are seals with minimal wear at low surface speeds of less than 2 m/s. In this speed range, the coefficient of friction of a seal with a wear width of 0.1 mm is higher than that of a seal with a wear width of 0.5 mm. For higher surface speeds, the coefficients of friction for these small wear widths are identical. As the wear width continues to increase, the coefficient of friction increases across all surface speeds. At wear widths of 0.8 mm and higher, the coefficient of friction decreases at low surface speeds up to 0.5 m/s with increasing speed. At higher surface speeds, the curve rises degressively with the surface speed. This behaviour indicates mixed friction at low surface speeds. At lower wear levels, this does not occur at the tested surface speeds.

Figure 7 shows the coefficient of friction of sealing systems with NBR seal rings as a function of surface speed for different wear widths.

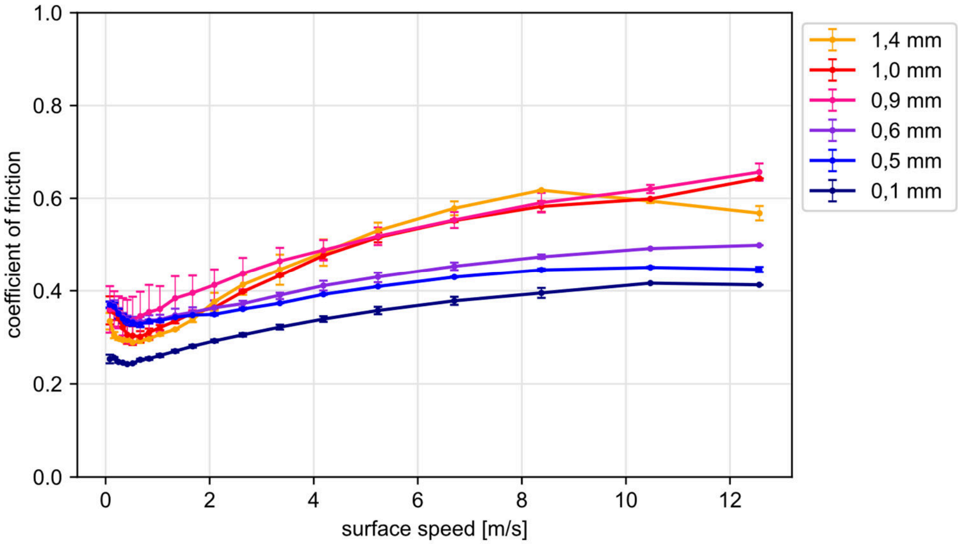

Figure 7: Coefficient of friction in relation to surface speed for NBR seals

At low surface speeds, the coefficient of friction decreases as the surface speed increases, reaching a minimum at around 0.5 m/s. This suggests mixed friction in the speed range below 0.5 m/s. As the surface speed increases, the coefficient of friction rises degressively. This suggests fluid friction. The coefficient of friction increases with increasing wear width up to a wear width of approximately 0.9 mm. For wear widths between 0.5 mm and 0.9 mm, the coefficients of friction in the mixed friction range are virtually identical. In the range of fluid friction (surface speeds over 0.5 m/s), the coefficient of friction increases more with increasing wear. For wear widths above 0.9 mm, the coefficient of friction decreases in the mixed friction range. In the range of fluid friction, the coefficient of friction rises to a similar level as at a wear width of 0.9 mm.

3.3 Effect of wear on the contact temperature

To investigate the temperature in the sealing contact, the temperature of the sealing edge is measured at the air side as close as possible to the sealing contact. The actual contact temperature cannot be measured in this way, as the contact area is not visible to the thermal camera. The measured temperature is slightly below the contact temperature, but exhibits similar behaviour.

Figure 8 shows the temperatures at the sealing edge of all seals over the surface speed.

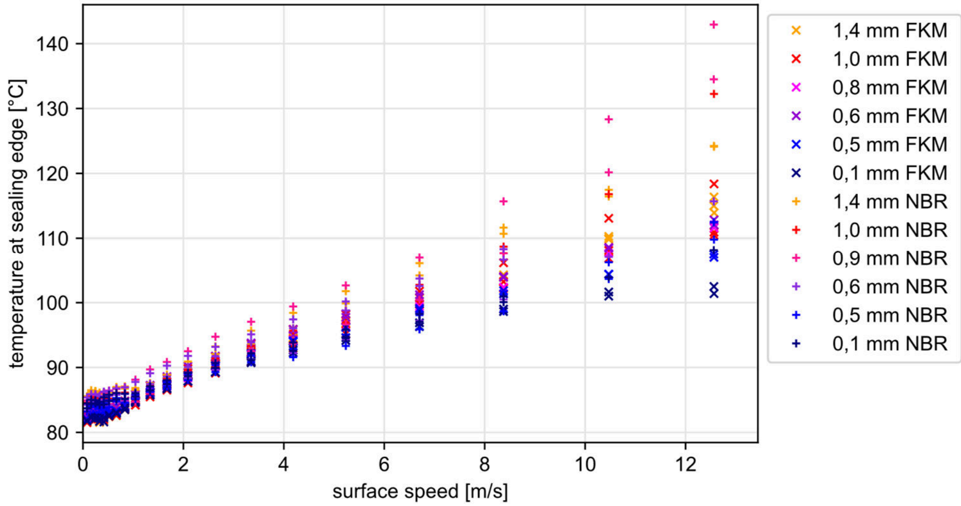

Figure 8: Temperature at the sealing edge over surface speed for FKM and NBR seal rings with varying degrees of wear

As the surface speed increases, the temperature at the sealing edge rises. The temperature rise is the lowest for the FKM seals with a wear width of 0.1 mm and the highest for NBR seal rings with a wear width of 0f.9 mm. This behaviour is as expected. The frictional power depends on the frictional torque and the surface speed. The NBR seal rings with a wear width of 0.9 mm have the highest frictional torque (see Figure 4). This results in the highest frictional power and the highest contact temperature. To better illustrate this behaviour, Figure 9 shows the temperature difference between the sealing edge and the fluid sump over the power loss related to the shaft circumference.

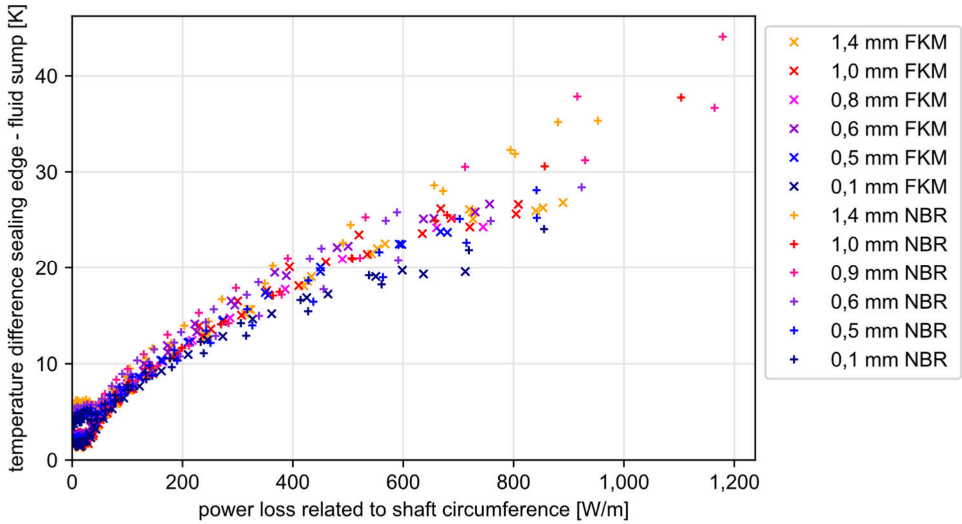

Figure 9: Temperature difference between sealing edge and fluid sump over power loss related to shaft circumference

The temperature difference between the sealing edge and the fluid sump increases as the power loss related to the shaft circumference increases. The trend in all test series is, as a first approximation, linear and has a gradient of approximately $0.03\,\frac{K \cdot m}{W}$. This gradient can be interpreted as the thermal resistance. The thermal resistance describes how effectively heat can be transferred away from the contact area. The curve is slightly flatter for smaller wear widths than for larger wear widths. The thermal resistance, therefore, increases with increasing wear width. This means that heat dissipation from the contact worsens slightly as the wear width increases.

4 Summary and Conclusion

Dry running can create wear on seal rings within a short time period of a few hours. This enables the creation of seal rings with a specific wear width. If a seal is worn in this way and subsequently lubricated by a fluid, the additional wear during standard test conditions for frictional torque measurements is negligible.

For seal rings made of FKM, the frictional torque of the seal increases with increasing wear. For seal rings made of NBR, the frictional torque of the seal increases with increasing wear up to a wear width of approx. 0.9 mm. With further wear, the frictional torque decreases again.

With increasing wear, not only the wear width, but also the inner diameter of the seal ring increases. As a result, the interference between seal ring and shaft and the radial load decreases.

For FKM seal rings, the coefficient of friction increases as the wear width increases. For NBR seal rings, the coefficient of friction increases up to a wear width of approximately 0.9 mm. For wear widths exceeding 0.9 mm, the coefficient of friction decreases in the range of mixed friction (surface speeds below 0.5 m/s). In the range of fluid friction, the coefficient of friction remains at a similarly high level to that at a wear width of 0.9 mm.

The contact temperature rises with increasing circumferential speed and increasing frictional torque. As wear increases, the contact temperature also tends to rise. The highest contact temperatures are reached with NBR seal rings with a wear width of 0.9 mm. As the wear width increases further, the contact temperature decreases. Heat dissipation from the contact area is also influenced by the wear width. As the wear width increases, heat dissipation worsens slightly.

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