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Performance characterization of marine lip seals: Contact temperature and frictional torque

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ABSTRACT

This paper investigates the performance of marine thruster lip seals. Marine lip seals have received much less attention in the published literature than lip seals used in other fields. The main differences between marine and non-marine lip seals are the harder material compound, the higher oil pressure, and the larger diameters. The lip seals that are studied in this paper have a nominal diameter of 300 mm. The counterface is a tungsten-carbide-coated stainless steel shaft liner. The studied lip seals and their counterfaces are commonly used in marine applications. A full-scale test device is built to study the effects of marine-specific parameters on the behavior of the lip seal, using a parameter sweep of the oil temperature, oil pressure, and rotational speed. Air bearings are used to allow for a more accurate measurement of the frictional torque. The subsurface temperature at the contact is measured using a wireless temperature probe inside the rotating shaft liner. In future work, the subsurface temperature is used in a thermal finite element model to estimate the average contact temperature.

1. Introduction

Lip seals are a common type of seal of rather rudimentary construction, used in many fields such as automotive and marine applications. They have been in use since the 1930s [1] and were introduced to the marine field in the 1950s when wooden water-lubricated bearings were replaced with metallic, oil-lubricated bearings [2]. They enable reduced maintenance costs and downtime; however, oil leakage became a problem and is estimated to total 80 million liters per year [2]. This corresponds to an average of 6 L per day for each vessel, or an estimated 1 L per day for a shaft diameter of 100 mm [2]. This level of leakage can be considered substantial, considering that well-designed lip seals should not leak at all during operation. The leakage levels in marine lip seals stem mainly from the harsher operating conditions, characterized by the seawater, its contaminants, the water pressure and temperature fluctuations, and the larger sliding speeds due to the larger seal diameters. All these factors are not usually studied in laboratory tests, and some of these conditions will be studied in future work.

Generally, lip seals do not leak while in operation. The sealing occurs based on the principle of reverse pumping [3]. An oil film separates the lip seal and the counterface, providing lubrication. Concurrently, the tangential deformation of the surface asperities of the lip shapes them into a vane-like pattern. The surface roughness of the contact patch resembles two counteracting screw pumps, resulting in a net oil flow toward the oil side. This mechanism is extremely sensitive to the conditions at the contact zone and is affected by the interference fit [4], the surface roughness [5–7], the local hardness of the rubber [8], and any misalignments between the shaft and the lip seal or out of roundness of the shaft [9–11].

A typical marine lip seal package consists of several seals in succession, as shown in Fig. 1. The arrangement and number of the seals depends on the specific vessel, its thruster type, and the type of application. The water-facing seal, the leftmost seal in Fig. 1, always has the same orientation. The sealed side is the seawater. The other seals can be oriented either way and are separated by oil chambers, which are pressurized above or below the seawater pressure. Ultimately, reverse pumping or leakage creates a pressure-driven flow to one of the chambers, usually the one in the middle. This chamber is used as a collection point for leakages in some seal packages, acting essentially as a fail-safe mechanism. In practice, most seals are exposed to seawater, albeit to a different level. Fluid mixing has been observed with a fully flooded seal [12].

Marine lip seals usually operate with a pressure difference over the seal. The sealed side (spring side) has the higher pressure. The pressure difference is typically 0.2–0.3 bar and can intermittently reach 1 bar. The larger marine lip seal diameters result in an increased sliding speed at the contact, and the seals are more significantly affected by angular misalignments [10,11]. In addition, the seals operate fully submersed, while in most non-marine lip seal applications, the lip has an air-oil interface, i.e., an air side and an oil side.

The published literature seldom describes marine seals. To the best knowledge of the authors, only Borras and colleagues [10,11,13,14]

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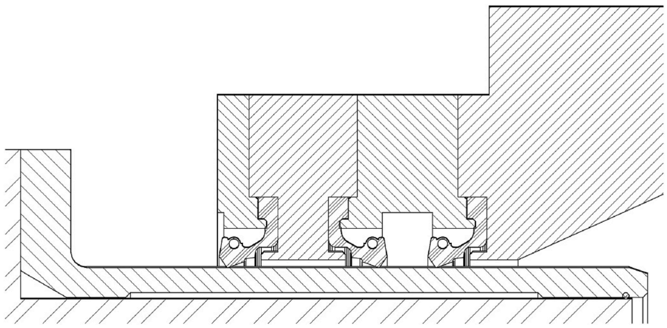

Fig. 1. Example of marine lip seals.

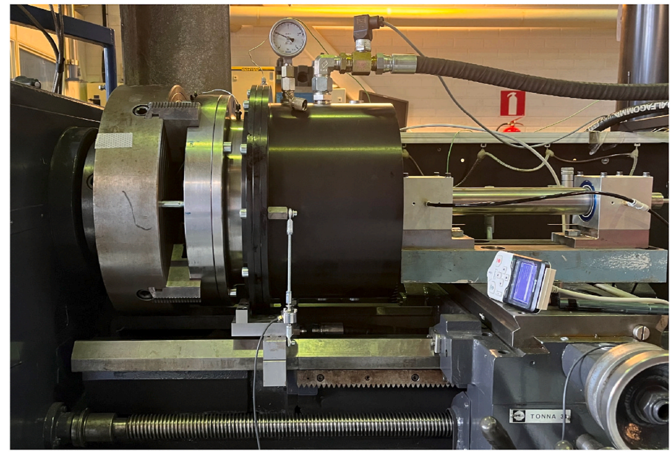

Fig. 3. Photo of the test device.

have studied them. The sensitivity of the lubricating oil film to many operational parameters, in addition to the harsh operating environment of marine lip seals, creates challenges in understanding how they operate. In this paper, a marine lip seal is studied using state-of-the-art methods to establish a comparative benchmark to seals in the published literature, which usually have an air-oil interface. The performance of the lip seals is characterized by the frictional torque and the subsurface temperature at the contact. The subsurface temperature provides a good indication of the contact temperature, which is important since the contact temperature is detrimental to the oil film and the material of the lip at the contact. In later work, the contact temperature will be calculated and the same seals will be studied in fully submersed conditions. The submersion will have both oil-oil and oil-water interfaces.

2. Methods

A test device was built to test large-size lip seals. A cross-sectional view of the device is shown in Fig. 2, and a photo of the actual test device is shown in Fig. 3. The test device was constructed on a Colchester Mascot 1600 lathe. The lathe provided good mechanical support, a movable carriage for ease of assembly and disassembly, and a powerful motor. The lathe motor was connected to an ABB ACS 880 variable frequency drive (VFD), which allowed for fine speed control. The tool carriage was modified to support two air bearings that carry the static shaft. The air bearings provided sufficient support and ensured minimal friction on the shaft. The seal housing was attached at one end of the static shaft. At the other end of the shaft, a 20 mm ball was placed in contact with a flat end of a hardened steel rod placed in the tailstock, thus creating a low-friction thrust bearing. Both air bearings had insignificant friction. The thrust bearing, a ball-on-flat contact between two polished hard steels, had negligible frictional toque. A thrust air bearing will replace the ball-on-flat thrust bearing in future tests to further minimize the effect of parasitic friction on the frictional torque measurement.

An FE60 shaft stub was placed inside the chuck. The shaft stub was chosen to have a solid section to simulate the thermal conductivity in marine thrusters. An AISI 316 shaft liner coated with tungsten carbide was used as the counterface. This liner is commonly used in marine thrusters. The surface roughness of the shaft was measured using a Mahr MarSurf PS 10 portable surface roughness measurement device. Five samples were taken at different radial locations on the liner. For each sample, a 1.5 mm length was measured axially that consisted of 3000 measurement points. The counterface of the liner, the lip seal, was part of the same sealing package. It had a nominal diameter of 300 mm and a cross section of 300 × 357 × 18 mm. Its radial interference fit was 4.5 mm. The lip seal was made from 80 NBR 94207, with a hardness of 80 Shore A. The seal did not have the typical L-shaped bracket but had a garter spring for increased radial force. An optical microscope with a 5x zoom lens was used to capture the contact surface of a new seal and a used seal.

The seal housing was a hollow cylindrical piece with a flanged end. Instead of brass, which is typically used in marine seals, it was made from 6082-T6 aluminum to reduce the lateral load on the radial air bearings. Two seal housing rings which housed the lip seal were made from the same material and were connected via bolts. The dimensions of the seal housing rings were made to match those of the commercial seal package. The gap between the shaft stub and the inner wall of the seal housing was large enough to dissipate the rotational viscous friction of the oil.

The frictional force was measured by restraining the seal housing

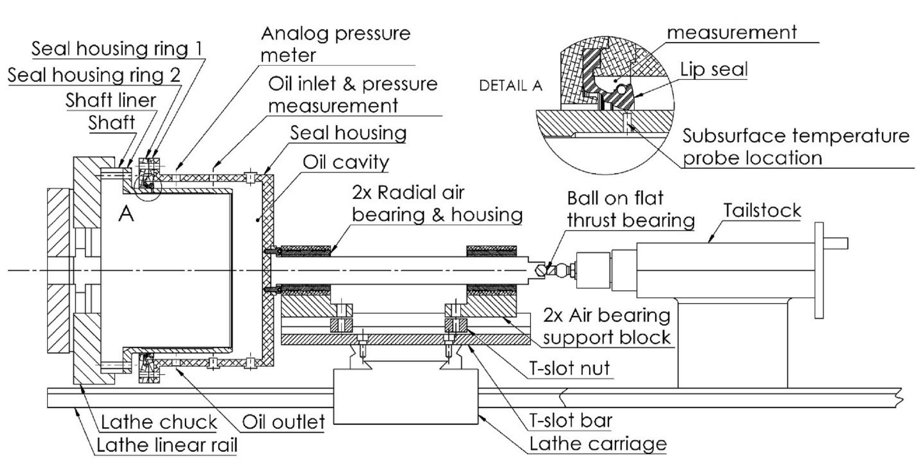

Fig. 2. Cross section of test machine.

assembly rotationally by a load cell in tension. The load cell was an HBM U9C with a maximum load of 200 N. Weights were used to calibrate it before the tests. The frictional torque was calculated by multiplying the measured force by the moment arm, which was measured to be 0.19 m.

The lubricating oil was Shell Omala S2 GX 100, an ISO grade 100 oil. The manufacturer properties showed that the oil had a kinematic viscosity of 100 mm/s^2 at 40 °C and 11.4 mm/s^2 at 100 °C. The oil was circulated through the seal housing via a gear pump, which was controlled by an ABB ACS 880 VFD. The oil passed through a heated oil tank. A 3 kW heater was placed inside the tank and was controlled by a commercial on/off type controller acting essentially as a thermostat. The controller measured the temperature inside the tank and turned on once the hysteresis limit was reached. The hysteresis of the controller was set to 1 °C. The temperature set point of the tank heater was chosen so that the oil in the seal housing was at the desired test temperature. The oil temperature was taken at the vicinity of the lip seal, as shown in Detail A in Fig. 2, with a K-type thermocouple. The thermocouple was calibrated in an ice bath and a boiling water bath.

The oil was circulated from the pump to the tank, then to the seal housing, and then back to the pump. The oil circuit had an oil line that bypassed the seal housing in order to maintain the thermal stability in the system. The speed of the oil pump was set to maintain the temperature of the oil near the lip seal at the desired setpoint. The flow rate of the circulated oil did not affect the frictional torque. The oil passed through three 10 μm filters placed on separate lines to remove any contaminants circulating in the system. The filters may have contained some wear particles from the lip seal, and this may be analyzed as part of future work.

The oil was pressurized by pneumatically pressurizing the oil tank. An SMC ITV20 electro-pneumatic regulator was used to regulate the pressure. The pressure was set in LabVIEW through a PID controller, which generated an input voltage signal to the regulator. The PID controller was necessary because the pressure in the seal housing changed when the oil was heated. The reference point for the PID control was the measured oil pressure inside the seal housing. The oil pressure was measured using a factory-calibrated WIKA A-10 pressure transducer.

LabVIEW was used to record the measurements and produce the voltage command signals for the electro-pneumatic regulator. The communication was facilitated by a cDAQ-9174 data acquisition module. Connected to this was an NI-9129 universal analog input-output module to record the pressure, temperature, and load cell measurements. An NI-9263 voltage module generated the input voltage signal for the electro-pneumatic regulator.

The subsurface contact temperature was measured using a K-type thermocouple, placed in the rotating liner 0.5 mm beneath the contact surface. The thermocouple was calibrated in an ice water bath and a boiling water bath. The thermocouple was connected to a Nokeval FTR264 wireless transmitter that was placed on the rotating shaft of the lathe. It transmitted the temperature reading at a set interval to a Nokeval FT20 receiver. The receiver was connected to a computer via a Nokeval DCS772 connector. The temperature readings were read and stored using LabVIEW via a USB connection and the Modbus protocol.

The concentricity between the shaft and the seal housing was measured using a dial gauge. Adjustments were made with shim plates placed between the air bearing housings and the T-slot bar. The final shaft to bore misalignment was 0.1 mm.

3. Results

A total of seven tests were performed. Three values were tested each for the oil pressure, oil temperature, and rotational speed. One parameter was varied while the other two were fixed. A collection container was placed below the seal housing to collect any leakage; however, no leakage was observed in any test. Any leakage may have taken the form of small droplets adhering to the shaft liner. The midpoint test, with a speed of 200 rpm, a pressure of 0.2 bar, and temperature of 50 °C, was performed only once instead of a triple repetition. Its results were used in all three comparisons. The results of the tests are shown in Figs. 4–6. All the results shown are averaged values over 12 h of runtime. The error bars represent the standard deviation of the measurements.

The liner had an R_a surface roughness of 0.4 μm, which was the average of six measurements around the liner. The lip seal running-in was done at a speed of 150 rpm, a pressure of 0.3 bar, and oil at room temperature. After 24 h, the frictional torque value decreased from 22.5 Nm to 20.8 Nm, after which the value remained steady.

3.1. *Temperature variation*

In the temperature variation tests, the speed was held constant at 200 rpm and the pressure at 0.2 bar. The oil temperature varied between 40 °C, 50 °C, and 60 °C. The results are shown in Fig. 4. The subsurface temperature increased with increasing oil temperature. At an oil temperature of 60 °C, the subsurface temperature was 95.3 °C. The frictional torque decreased with increasing oil temperature and varied between 14.6 and 17.0 Nm. The lowest frictional torque value for all tests was obtained in the 60 °C oil temperature test.

3.2. *Pressure variation*

In the pressure variation tests, the pressure values were 0.1, 0.2, and 0.3 bar; the latter is the most used pressure level in marine seals. The oil temperature was fixed at 50 °C and the speed was 200 rpm. The results are shown in Fig. 5. A higher oil pressure resulted in a higher subsurface temperature, with a more pronounced increase at 0.3 bar. At this pressure, the subsurface temperature was 92.1 °C. The frictional torque increased with increasing oil pressure. The frictional torque was 18.5 Nm in the test with 0.3 bar. The increase in the frictional torque was more pronounced in the 0.3 bar test compared to the other pressures.

3.3. *Speed variation*

In the speed variation tests, the seals were tested at speeds of 100, 200, and 300 rpm, which corresponded to sliding speeds of 1.57, 3.14, and 4.71 m/s, respectively. In all three tests, the oil temperature was 50 °C and the pressure was 0.2 bar. The results are shown in Fig. 6. The subsurface temperature increased with increasing speed and reached 101.5 °C at a speed of 300 rpm. The frictional torque also increased with speed to a maximum of 17.5 Nm at a speed of 300 rpm.

3.4. *Contact surfaces*

The shaft liner surface roughness had an R_a value of 0.4 μm before the test. The value was unchanged after the test. A sample of the surface roughness measurement is shown in Fig. 7. The unused lip seal had a V-shaped edge. After the tests, it developed a flatter edge, as shown in Fig. 8.

4. Discussion

4.1. *Effect of temperature*

In the 60 °C oil temperature test, the subsurface temperature reached 95.3 °C. This is a relatively high temperature that can reduce the elastic modulus of the NBR seal material [15]. In addition, the NBR material can swell, which results in an increased inner diameter and thus a reduced interference fit [15]. Stress relaxation also occurs under elevated temperatures in oil, which changes the geometry of the lip [15]. All these factors can reduce the reverse pumping ability of the seal and increase the chance of leakage. In this specific test, however, no leakage was observed. The increase in the subsurface contact temperature with higher oil temperatures was significant.

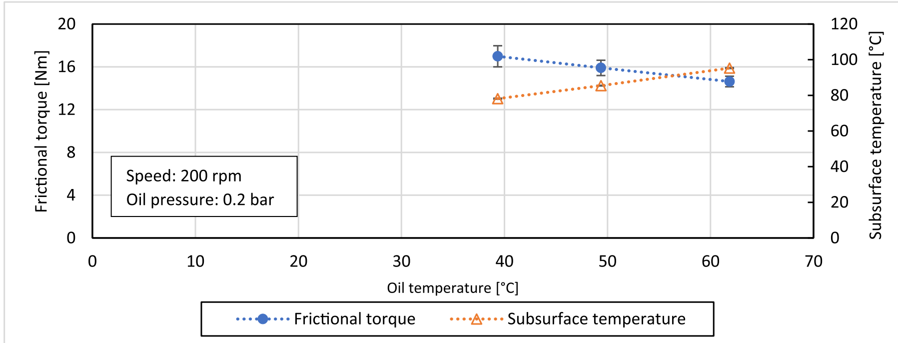

Fig. 4. Variation of frictional torque & subsurface temperature with oil temperature.

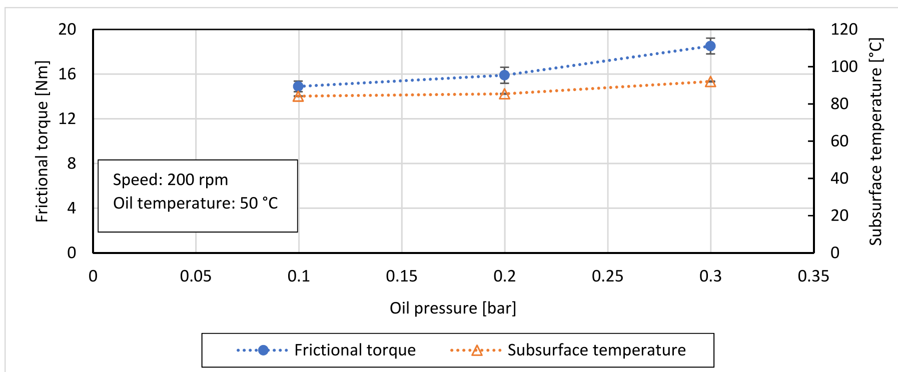

Fig. 5. Variation of frictional torque & subsurface temperature with oil pressure.

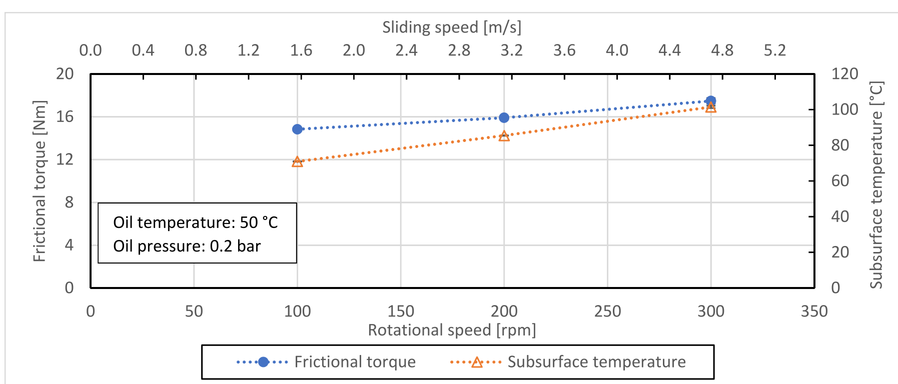

Fig. 6. Variation of frictional torque and subsurface temperature with rotational speed.

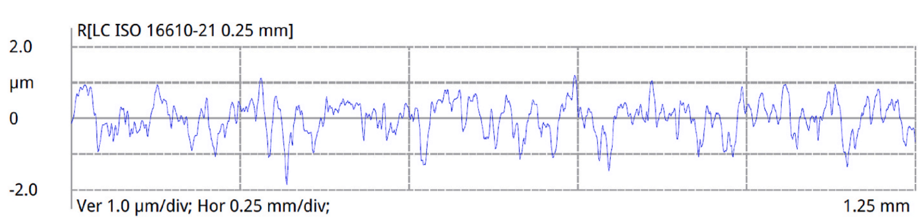

Fig. 7. Surface roughness measurement of shaft liner. $R_a = 0.471$ µm.

A decrease in frictional torque with an increasing oil temperature is expected due to a reduction in the viscosity of the oil, which in turn decreases the viscous shear component of the frictional torque. The other component of the frictional torque is due to the asperity contact. Furthermore, a lower viscosity results in a thinner film, which in turn may result in an increased asperity contact, increasing the wear. It also reduces the reverse pumping rate [16–18].

The effects of higher oil temperature in this study were a decrease in the frictional torque and an increase in the contact temperature. Yan et al. [15] reported the same effect of oil temperature on frictional

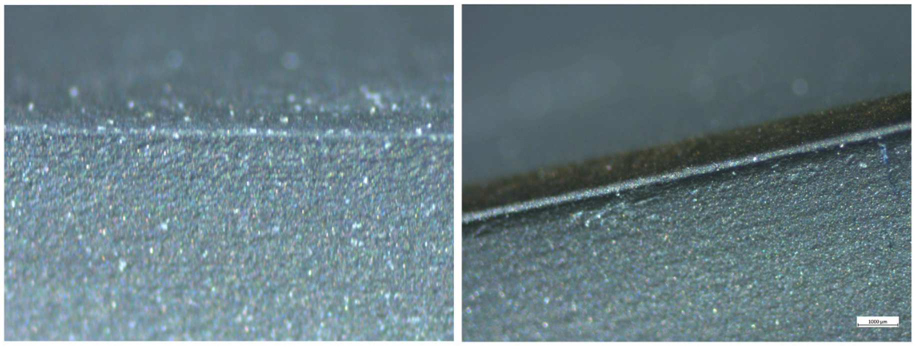
(a) (b)

Fig. 8. Microscopic images of sealing edge of (a) new and (b) used lip seal.

torque and contact temperature. They tested an inversely installed lip seal with a diameter of 45 mm and hydraulic oil #32, at a speed of 5.9 m/s.

4.2. Effect of pressure

The pressure had the most significant effect on the frictional torque. The highest frictional torque of all tests was measured with a pressure of 0.3 bar. Higher oil pressures resulted in a higher contact pressure and a wider contact length between the lip and the shaft [13,19]. The higher contact pressure increased the frictional torque [4]. The increase originated from the asperity contact component of the frictional torque, which can also increase the wear of the lip.

The subsurface temperature showed the least sensitivity to the oil pressure. The change was moderate from 0.1 bar to 0.2 bar. The increase at 0.3 bar to 92.1 °C was larger but not as pronounced as in the other tests. As discussed in Section 4.1, the temperature rise may still affect the properties of the rubber and the sealing conditions.

Borras et al. [14] tested a marine fluoroelastomer lip seal with a diameter of 200 mm at different pressures. The tests were conducted under fully flooded conditions and used an intermediate oil between ISO grades VG100 and VG 150. Both the frictional torque and the contact temperature increased with increasing pressure.

4.3. Effect of speed

In the 300 rpm test, the subsurface temperature, 101.5 °C, was the highest of the seven test runs. Such a high temperature can be detrimental to the sealing ability of the lip seal, as discussed in Section 4.1. A reduction in the contact pressures is expected at this temperature. This may explain why the increase in the frictional torque from the midpoint test was not as large as in the 0.3 bar pressure test. The increase in the subsurface temperature with each 100 rpm step was more significant compared to the other parameters, indicating a higher sensitivity to the speed.

The frictional torque also increased with increasing speeds. The increase was less pronounced compared to the pressure test due to the influence of the oil temperature on both the viscous shear and the contact force. The high temperature simultaneously reduced the contact pressure by reducing the elastic modulus and reduced the viscosity, resulting in a softer lip and a thinner film that generated less viscous shear. This combined effect likely explains why the highest subsurface temperature did not occur concurrently with the highest frictional torque. The increase in the frictional torque at higher speeds is an indication that the lip seal operates in the full film regime, since the frictional torque vs. speed curve in Fig. 6 is comparable to a typical Stribeck curve.

Borras et al. [14] reported an increase in contact temperature and a decrease in frictional torque with higher speeds in their test of a marine lip seal. The behavior of the frictional torque was explained by the oil film becoming thinner due to higher contact temperatures. The Stribeck curve for their seal showed that it operated after the friction minimum point, so a decrease in frictional torque may be expected.

Yan et al. [15] reported an increase of frictional torque and contact temperature at higher speeds. The contact temperatures were lower in their tests, even though the sliding speed range was similar. This may be attributed to the inverted installation of the lip seal or the interference fit. Regardless, the effect of speed was similar to the current study. El Gadari et al. [20] tested a fluoroelastomer lip seal and found that, for the same speed range as the current study, the contact temperature and frictional torque increased with higher speed. Guo et al. [21] tested an inversely installed NBR lip seal with a nominal diameter of 100 mm and with #32 hydraulic oil. The frictional torque increased with higher sliding speeds for the same speed range as the current study. Horve [9] obtained the same effect with six seals of different nominal diameters, ranging from 32 to 130 mm. The frictional torque and contact temperature always increased with higher speeds. Larger seals resulted in higher frictional torques. The distinction in the relation between size and contact temperature was not clear in Horve's tests, likely due to the high oil temperature of 93 °C.

4.4. General observations

The subsurface temperature values obtained in the tests are an indication of the contact temperature between the lip and the shaft. The subsurface temperature is a better indicator of the average contact temperature than the maximum temperature, since the thermocouple probe has a diameter of 1.5 mm. In contrast, a typical contact width is 0.1–0.5 mm [22,23]. In a future article, the subsurface temperature will be used in a thermal finite element model (FEM) to obtain the contact temperature of the lip. Nevertheless, it can be assumed based on published numerical models [16,18] that the contact temperature is higher by an order of 1 °C–10 °C. The same models showed temperature gradients on the lip surface of the order of 10 °C between the asperity peaks and the valleys. A similar temperature gradient was computed in those models inside the oil film, with the shaft surface being cooler. All of this indicates that the lip material at the contact zone operates under elevated temperatures, which may affect the wear mechanisms of the lip.

In an industrial report, identical lip seals used on a vessel were analyzed. All lip seals had an increased hardness. Some lip seals experienced thermal glazing and axial heat cracks. The increase in the hardness may have been due to strain hardening, sustained tangential deformation, elevated contact temperatures, or a combination of those factors. Some of the seals were exposed to pressure gradients as high as 1 bar. If these results are extrapolated, it shows that the seals would have experienced much higher frictional torques and contact temperatures. Circumferential grooves were observed on the liner in the same report. Generally, axial grooves are expected on the metallic shafts or liners after a long period of operation and can be visible to the naked eye. The grooves develop because the lubricated contact between the lip and the shaft is analogous to a polishing process, which reduces the roughness of the shaft at the contact zone [24]. This was not observed in the experimental work of this paper, likely due to the shorter running time.

The subsurface temperature measurement method presented here is rather novel. The proximity to the contact surface, along with the high thermal conductivity of the liner, provides a good indication of the contact temperature. A similar measurement approach was done by Stakenborg [25], albeit the seal was rotating and the shaft was stationary. Other researchers typically place a thermocouple on the seal or use an infrared thermal camera to measure the temperature of the seal. However, both methods measure at a location that does not provide a good indication of the contact temperature.

In the test results, the relation between frictional torque and subsurface contact temperature was not always clear. At a constant temperature, both increased simultaneously with higher pressures and speeds, albeit at different rates. This indicates a different codependence for pressure than for speed. On the other hand, the subsurface temperature and frictional torque were inversely proportional when the oil temperature was varied under a constant pressure and speed. Therefore, the frictional torque cannot be used as a reliable indicator of a high subsurface or contact temperature and vice versa. This can be expected from the theoretical literature, as thermal-elastohydrodynamic models [6,18,26] typically have complex formulae to compute the frictional torque and the contact temperature. The formulae usually contain a viscous shear term that accounts for viscosity change due to temperature. Some models also consider the effect of temperature on the mechanical properties of the lip, which affects the frictional torque, which consequently affects the contact temperature.

The lower frictional torque values with smaller size lip seals are a direct result of the smaller circumferential contact length. Since the radial force of a lip seal is measured in N/mm of circumferential length, a larger diameter lip seal has a larger radial force acting on it, which generates higher frictional force and heat. Therefore, a larger seal diameter will result in a higher contact temperature if all other parameters stay the same (e.g., sliding speed, oil sump temperature). In comparison, the contact temperature did not seem to be significantly affected by the diameter of the lip seal [9]. The aforementioned tests by other researchers show that the contact temperature can easily reach 100 °C, albeit at higher speeds than those of the current study. This can be attributed to the difference in interference fits, since the interference fit of the studied seal is rather high compared to those found in published literature.

5. Conclusion

In this paper, a marine lip seal was tested under seven sets of parameters that were representative of realistic operating conditions. The subsurface temperature, and by extension the contact temperature, increased with increasing pressures, speeds, and oil temperatures. The speed had the highest impact on the subsurface temperature. The frictional torque also increased with increasing oil pressures and speeds but decreased with increasing oil temperature.

The results were comparable to the findings of other researchers in the field. They also shed light on the sealing behavior of large-diameter lip seals, a seldom-discussed topic in the literature. In future work, the tests will be extended to cover all possible combinations of the tested parameters. Furthermore, the contact temperature will be calculated using a thermal FEM model.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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