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PTFE Lip-seals with Bi-directional Sealing Aids

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1 Introduction and Basics

PTFE lip-seals are used in various areas of sealing technology because of their outstanding chemical and thermal resistance. In opposition to elastomeric lip-seals a PTFE lip-seal does not develop an automatic pumping mechanism and therefore requires additional sealing aids. A commonly used sealing aid is the spiral groove which is very effective but only suitable for one rotational direction [3]. Until now there are no reliable PTFE lip-seals with bi-directional sealing aids in the market. There are numerous patents on bi-directional sealing aids (e.g. [1], [2]) but sealing manufacturers obviously fail to implement these patents in functioning seals. The reason for that is deficient understanding of the sealing and pumping mechanisms. Understanding the mechanisms allows estimating the qualitative sealing and pumping properties and thus optimizing PTFE lip-seals with bi-directional sealing aids on an early stage of development. This paper contains simulations and experimental results which help to increase the understanding of the sealing and pumping mechanisms.

A general-purpose PTFE lip-seal requires bi-directional sealing aids to work in both rotational directions and not only has to prevent dynamic but also static leakage. Preventing static leakage requires a continuously closed ring which separates the oil and the air side, see Fig. 1. Due to the rough surface of PTFE-Compounds the closed ring is not leak-tight. A small amount of fluid is able to pass through micro-channels formed in the contact area between seal and shaft [4]. To collect and pump leakage back to the oil side, bi-directional sealing aids are necessary. This implies that the sealing aids have to be at least partially on the air side.

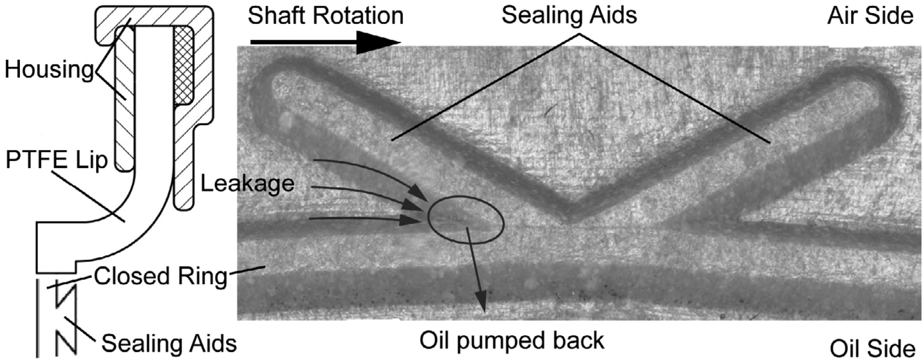


Fig. 1: Seal geometry and bi-directional sealing aids

For pumping fluid from the air to the oil side the closed ring has to be passed. This can be done the following ways:

- Channels can be formed locally in the closed ring if the drag flow pressure at the sealing aids is high enough.
- By hydrodynamic effects (and inertia) the closed ring can be lifted. This would lead to a gap in the contact area with the shaft.

Both of these effects can occur at the same time. In that case a dynamic gap with variable heights evolves in the contact area. To characterize this dynamic gap many different influences have to be taken into account. In the simulations below some of these influences will be discussed.

2 Simulation Results

From elastomeric lip-seals many bi-directional sealing aids are known, e.g. [5]. There are also several patents which describe various geometries for bi-directional sealing aids for PTFE lip-seals. But many of these geometries do not show reliable functionality in experiments, e.g. [6]. The reason for that can be easily understood by considering the capability of building up pressure by different sealing aid geometries. To generate a pumping effect the sealing aids have to build up pressure high enough to lift the closed ring at least locally. A typical PTFE lip-seal with a sealing diameter d = 80 mm, a radial force F_r = 180 N and a contact width b = 0.3 mm has an average contact pressure p_m of:

$$p_m = \frac{F_r}{\pi \cdot d \cdot b} \approx 24\, bar \left(= 2.4 \cdot 10^6\, \frac{N}{m^2}\right) \tag{1}$$

Fig. 2 and Fig. 3 depict the flow and pressure distribution around two different triangle-shaped sealing aids. The model consists of two parallel plates (distance 1 mm) and one sealing aid which builds up pressure towards a wall. The lower plate is moving at a velocity of 10 m/s (equivalent to 2387 rpm for d = 80 mm) and thus generates a drag flow. To ensure comparability of peak pressures the mesh sizing in the respective areas is almost uniform. It can be seen that the attainable maximum pressure (overpressure to the atmosphere) in both cases is about 0.6 bar (= $6.0 \cdot 10^4$ N/m^2). This is considerably less than the average pressure in the contact area calculated above. Therefore fluid will not be able to pass the closed ring. For this reason the basic geometry of a sealing aid is not crucial with regard to its ability to build up pressure.

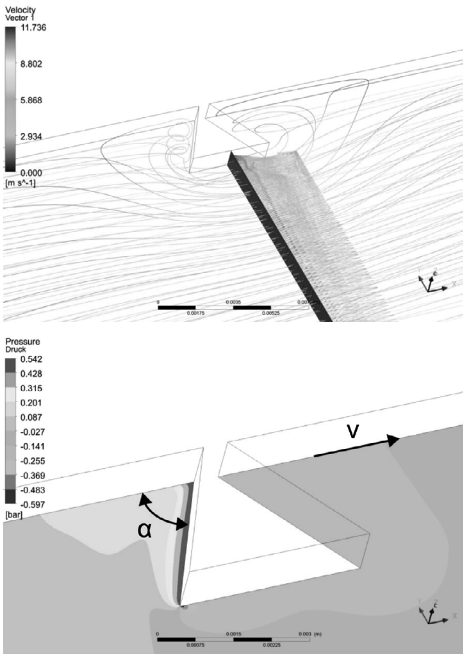


Fig. 2: Sealing aid no. 1, drag flow, v=10 m/s; α=60°

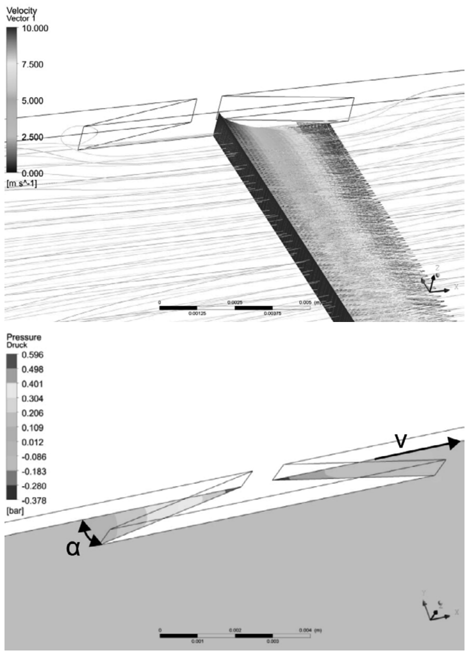


Fig. 3: Sealing aid no. 2, drag flow, v=10 m/s, α=10°

But how might the closed ring be passed? The simplest possibility is to make use of drag flow pressure in preformed channels. In Fig. 4 the drag flow in a closed rectangular channel is shown. The drag flow is completely redirected by the back wall and returns to the inlet driven by the built up pressure difference. An equilibrium state between drag and pressure flow evolves. The pressure distribution belonging to Fig. 4 is shown in Fig. 5 . In Fig. 6 and Fig. 7 the pressure distribution in wedge-shaped channels is depicted. In comparison to the straight channel the attainable pressure is much higher. The highest pressure can be realized in a double wedge-shaped channel as shown in Fig. 7.

To ensure comparability of peak pressures in Fig. 5 to Fig. 7 the mesh sizing is similar throughout the models. In addition to that the profile at the channels inlet is kept constant. But the pressure in Fig. 5 to Fig. 7 is still insufficient to overcome an average pressure of 24 bar (= $2.4 \cdot 10^6$ N/m^2) in the contact area. The reason for that is the relatively large cross-section of the channels used in the simulations. Reducing the cross-section (and keeping the length of the channels) would result in higher pressures.

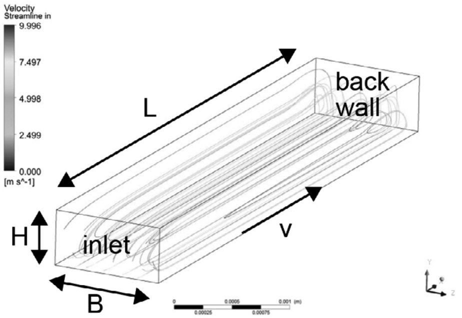


Fig. 4: Drag flow in straight channel, v=10 m/s, L=5 mm, B=1 mm, H=0.5 mm

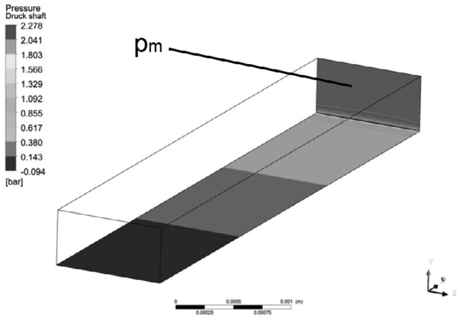


Fig. 5: Drag flow pressure in straight channel, average pressure at back wall **p_m=0.79 bar** (= $7.9 \cdot 10^4$ N/m^2)

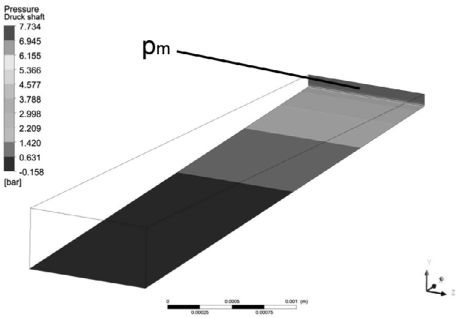


Fig. 6: Drag flow pressure in wedge-shaped channel, average pressure **p_m=3.75 bar** (= $3.75 \cdot 10^5$ N/m^2)

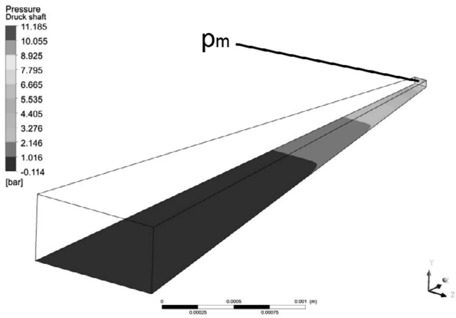


Fig. 7: Drag flow pressure in double wedge-shaped channel, average pressure **p_m=7.21 bar** (= $7.21 \cdot 10^5$ N/m^2)

The presented results point out that the pressure build-up capability of sealing aids does not depend on their general geometry but on their transition into the closed ring. This transition should be wedge-shaped to realize pressures high enough to overcome the closed ring. This could lead to the thought of reducing sealing aids to preformed wedge-shaped channels in the closed ring. But doing so would considerably decrease the sealing aids' capability to collect and hold leakage until it is pumped back onto the oil side.

Another important factor concerning the attainable flow rate by sealing aids is the pressure distribution in the contact area. The maximum contact pressure in the sealing gap often cannot be overcome by the sealing aids' drag flow pressure. As drag flow pressure instantly decreases with an increase in gap height the fluid naturally flows through the area with the lowest contact pressure. Therefore to attain pumping to the oil side the contact pressure distribution has to be appropriate. If this is not the case the fluid will be redirected to the air side, Fig. 8.

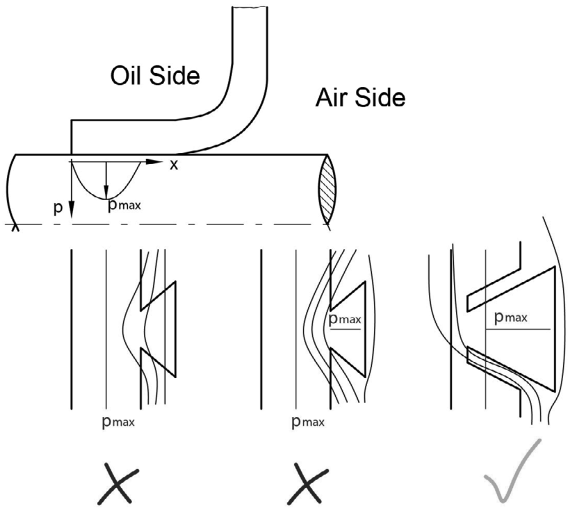


Fig. 8: Example of (in)appropriate pressure distribution in the contact area

In order to adjust the pressure distribution and geometry of the sealing aids properly, three-dimensional finite element analysis are necessary, Fig. 9. The contact area can also be studied experimentally by means of a glass hollow shaft as shown below.

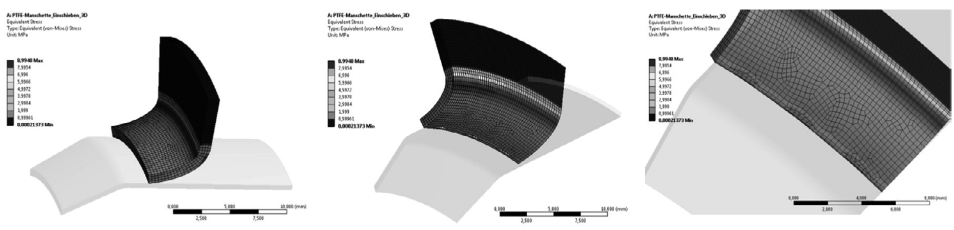


Fig. 9: Three-dimensional simulation of a 30° section of a PTFE lip

2.1 Conclusions – Simulation

With the help of relatively simple numerical models the fundamental functionality of bi-directional sealing aids could be explained. For a reliable pumping effect the transition between sealing aids and closed ring is crucial. The basic geometry of the sealing aid seems to be secondary. To attain high pressures at the closed ring wedge-shaped channels are a good choice. A proper pressure distribution in the contact area is crucial for the function of sealing aids.

Qualitative statements concerning the flow rate of bi-directional sealing aids require much more complicated models. To model the real processes correctly a two-way FSI (fluid-structure interaction) with cavitation is necessary. The surface roughness in the contact area is also to be taken into consideration as otherwise no hydrodynamic gap will evolve. In addition to that the mounting of the PTFE lip on the shaft has to be simulated in order to obtain the accurate pressure distribution in the contact area. Such a coupled simulation is still a great challenge regarding the model's complexity and the computation time. These challenges are pursued in several projects at the Institute of Machine Components.

3 Experimental Analysis

The PTFE lip-seals for the experimental analysis were manufactured at the Institute of Machine Components at the University of Stuttgart. The sealing aids were stamped into plain PTFE lips. The stamping press consists of two heatable plates and a hydraulic cylinder by which pressure is applied to the plates. The stamping die consists of two parts. The stamping die with structures that form the sealing aids and the counter die. The plain PTFE lip is inserted into the counter die and the stamping die is pressed onto the PTFE lip. The lips are stamped at 250 °C and 43 MPa pressure. The PTFE flows into the structures of the stamping die and so the sealing aids are formed. In Fig. 10 sections of counter die, stamping die and a plain and a stamped PTFE lip are shown.

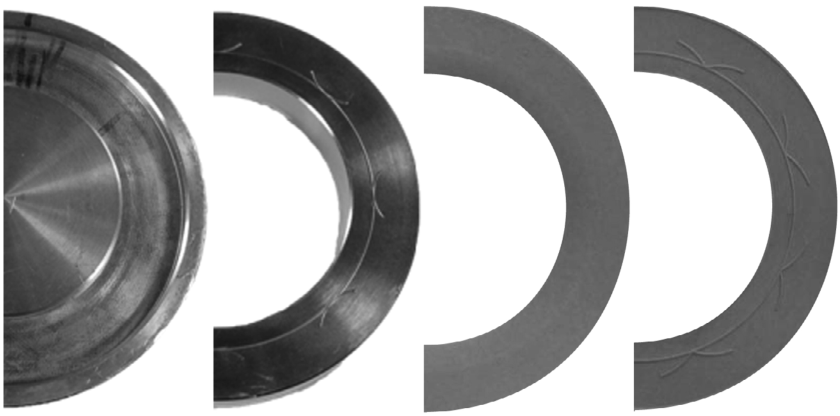


Fig. 10: Counter die, stamping die, plain and stamped PTFE lip (left to right)

The fluid flow under the seal lip and around the sealing aids is observed by using a flow analysis test rig with a glass hollow shaft, illustrated in Fig. 11. The seal is mounted between two elastomeric radial lip-seals. So oil can be filled to both sides of the test seal. The image from the seal lip is deflected by a prism and recorded with a camera. The prism is rotatable, so all positions of the seal lip can be observed. The oil is mixed with fine brass particles. These particles are dragged with the oil, making the oil-flow visible.

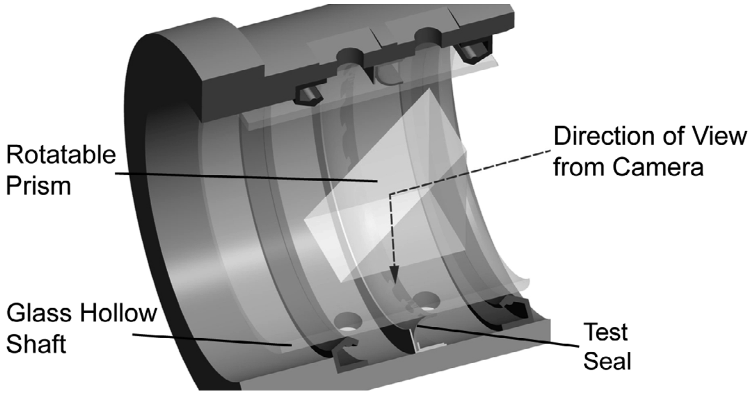


Fig. 11: Flow analysis test rig

To determine the static leakage the P-Method adapted by Bauer [3] for PTFE lip-seals with spiral groove was used. The leakage is rated from Point 8 (P8) to Point 1 (P1). P8 means that no leakage is visible. P1 means that drops of oil are flowing down the housing of the seal. For these investigations oil level is filled to the middle of the shaft. The time till each point is reached is measured. Fig. 12 shows the location of some points at the seal-lip.

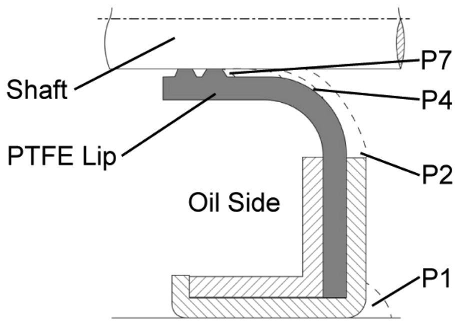


Fig. 12: Rating of static leakage

The test runs and the pumping rate tests were run at the test rig shown in Fig. 13. For all tests the oil chamber was filled with oil to the middle of the shaft. For the test runs the seals were mounted with the oil side facing into the oil chamber, see Fig. 13 right side. For the pumping rate tests the seals were mounted reversed so the sealing aids face the oil.

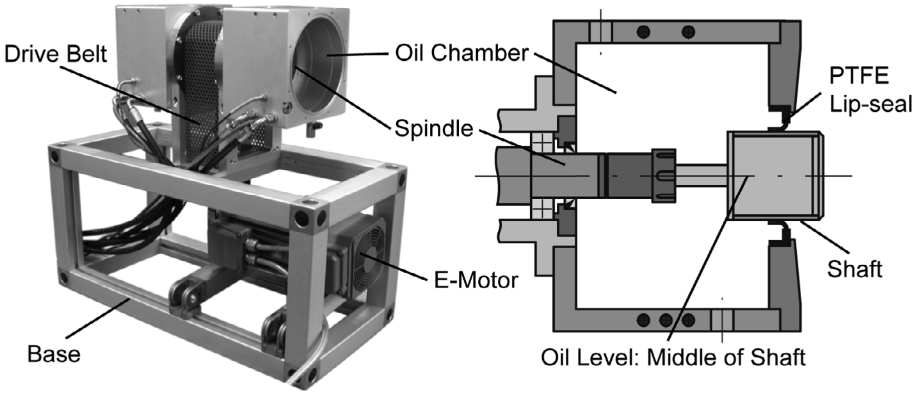


Fig. 13: Test rig for test runs and the pumping rate tests

3.1 First Investigations

In the first series of investigations a compound best suitable for stamping was determined. The first stamping dies were constructed and PTFE lips were stamped. In these investigations the foundation for all further research was laid and problems were identified. Additionally the PTFE lip-seals were comprehensively tested to have a benchmark to determine the improvement of future lip-seals.

The results from the first investigations are the following:

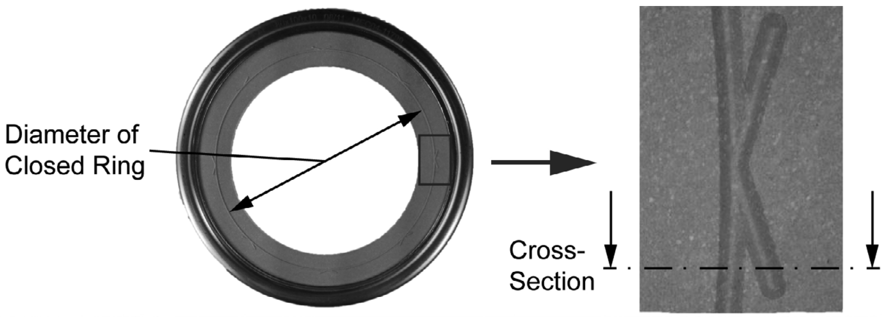
- From 5 different PTFE-Compounds the one with the best stamping ability and wear resistance was chosen; It was used for all further investigations
- Too narrow and deep structures on the stamping die lead to incorrect impression on the PTFE lips
- Thermal expansion of the PTFE-Compound is a problem due to high stamping temperatures
- The contact with the shaft is not optimal and the pumping rates between seals differ strongly; The stamping dies need to be improved
- The leak-tightness in both static and dynamic condition has to be improved

A more detailed report of the first experimental investigations is presented in [7] and [8].

3.2 *Investigations with improved lip-seals*

For these investigations the results from the first investigations were taken into account. The new lip-seals were stamped and analyzed.

Stamping dies with three different positions of the closed ring and sealing aids were constructed. The position of the sealing aids changes with the position of the closed ring since they are connected. The diameters of the closed ring on the stamping dies are 74, 76 and 78 mm. The diameters given refer to the stamping dies since the diameters on the PTFE lips change because of thermal expansion. Cross-sections from each PTFE lip-seal were taken. The form and size of the sealing aids was identical for each seal. The diameter of the closed ring of 74 mm is shown in Fig. 14. Both, the closed ring and the sealing aids do contact the shaft. The PTFE lip is too long so the closed ring does not contact the shaft well. This will cause problems for the static leak-tightness. The diameter of the closed ring of 76 mm is shown in Fig. 15. The contact of the closed ring and the sealing aids with the shaft is good. The diameter of the closed ring of 78 mm is shown in Fig. 16. Here the position of the sealing aids is too far to the outside of the PTFE lip. While the closed ring has good contact with the shaft, the sealing aids do not. This leads to good leak-tightness in static condition but the sealing aids cannot pump back oil and therefore will not work.



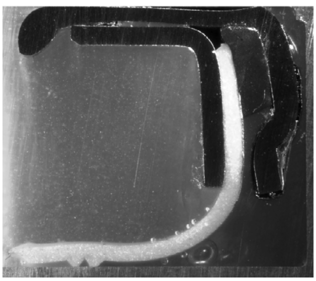


Fig. 14: 74 mm

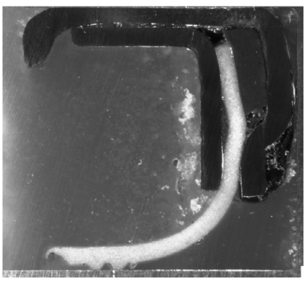


Fig. 15: 76 mm

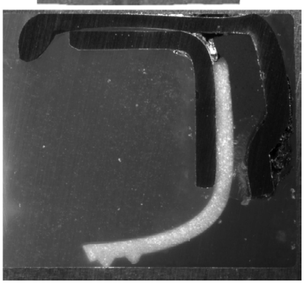


Fig. 16: 78 mm

The analysis of the leak-tightness in static condition shows that the assumptions based on the cross-sections above were correct. The results are shown in Fig. 17. The seals with a diameter of 74 mm (blue lines) showed heavy leakage after a relatively short time. The seals with diameters of 76 mm (red) and 78 mm (green) were, except for one seal, longer leak-tight. These results correlate to the cross-sections seen before.

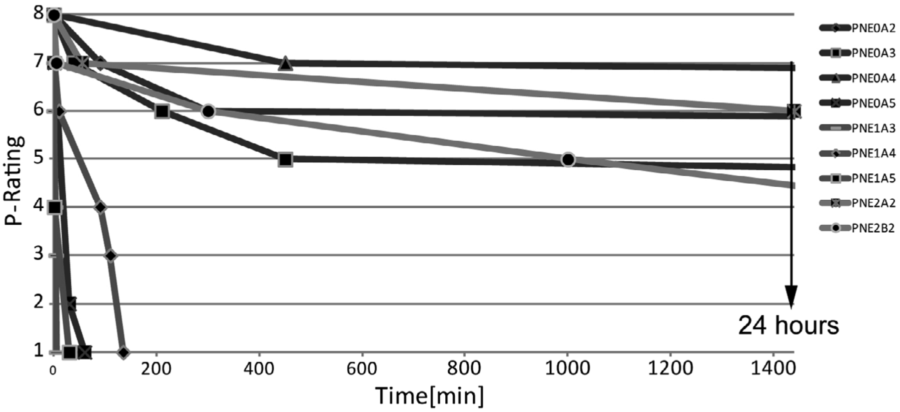


Fig. 17: Leak-tightness in static condition for the three diameters

In addition to the measurement of the static leakage the fluid flow was analyzed on the flow analysis test rig. As said before the static leak-tightness depends on the contact of the closed ring and the shaft. If the pressure of the closed ring to the shaft is not sufficient oil can leak immediately, see Fig. 18. If the pressure is sufficient the oil is held back, see Fig. 19.

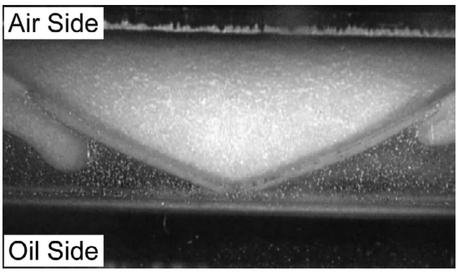


Fig. 18: Oil passes closed ring

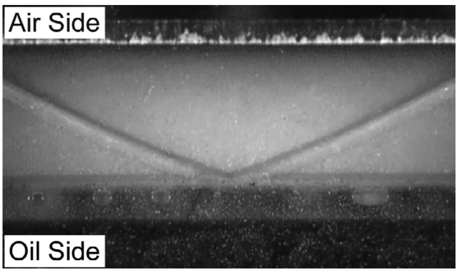


Fig. 19: No oil passes closed ring

The flow of the leakage is illustrated in Fig. 20. The position of the seal lip in the picture is above oil level. But due to capillary forces the oil flows along the closed ring and passes it in the area of the sealing aids. On the air side it follows the sealing aids and fills the area between the sealing aids and the closed ring. When more fluid passes the closed ring drops begin to form.

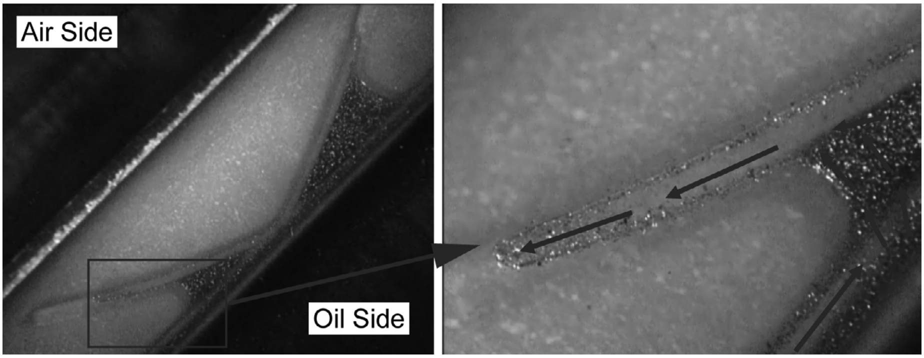


Fig. 20: Fluid flow in static condition

The seals were also analyzed in dynamic condition. There were test runs and the pumping rate of the seals was measured. Fig. 21 shows the pumping rate of a lip-seal at different shaft speeds. Up to 1000 rpm relatively little oil is pumped. With increasing speed the pumping rate also increases. The pumping rates were not identical for clockwise and counter-clockwise shaft rotation but the values measured were in the same range. This behaviour was observed for nearly all analyzed seals.

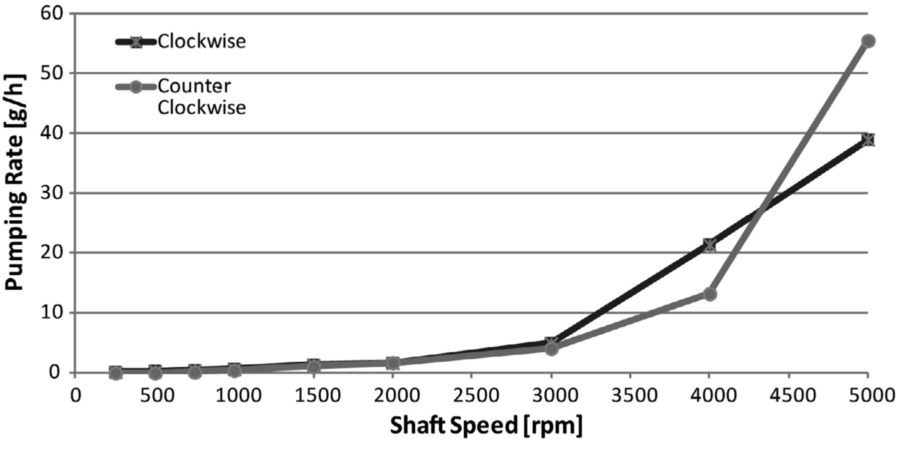


Fig. 21: Pumping rate at different speeds

A high pumping rate does not necessarily mean a seal will have no leakage in dynamic condition as shown in Fig. 22. The diagram displays the leak rates and pumping rates at 120°C and 3000 rpm. For a better clarity the average value of the measurement in both directions of rotation is used. The pumping rates are noted as negative, the leak rates positive. The leak rates were between 0.5 and 2 g/h. The pumping rates varied much more.

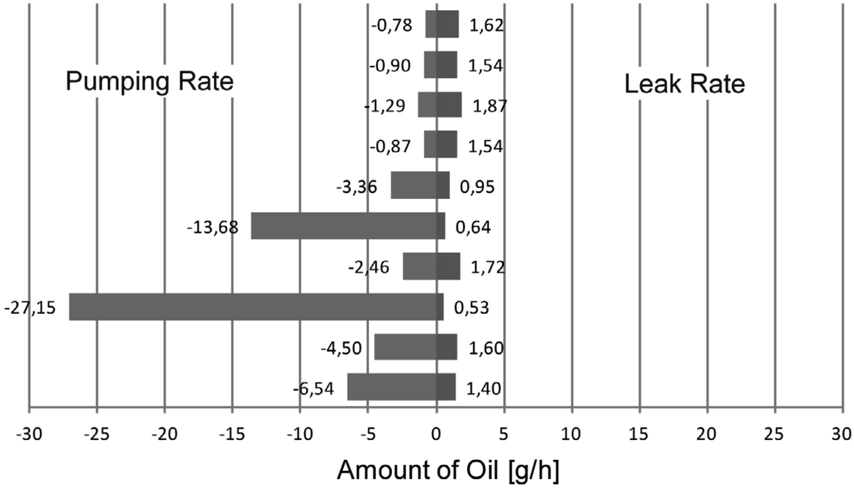


Fig. 22: Pumping rate compared to leak rate

Regardless of the pumping rate all seals showed leakage. A reason for this is that for the measurement of the pumping rate the sealing aids are flooded with oil. Mounted in normal position (as for the test runs/ leak rate measurements) the oil is on the other side of the closed ring and only the leaking oil reaches the sealing aids. If the hydrodynamic pressure built up at the sealing aids is not high enough to pump back oil under the closed ring it flows down the seal lip. Once the oil is past the sealing aids it cannot be collected and thus not pumped back.

To prove that the sealing aids can pump back oil channels were cut into the closed ring near the sealing aids. Three different alignments of channels were compared:

- No channels in the closed ring
- Channels in one direction of rotation (Direction: counter-clockwise)
- Channels in both directions of rotation

With channels in both directions of rotation naturally there is always one in flow direction and one in opposite flow direction. When the rotational direction changes the other channel is in flow direction. The position of the channels is shown in Fig. 23.

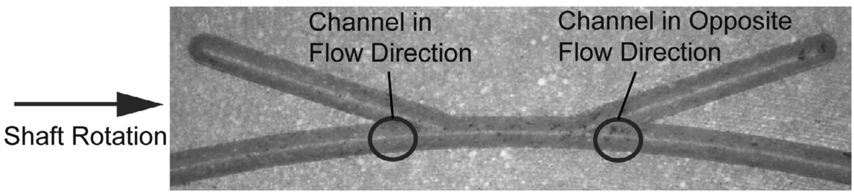


Fig. 23: Position of channels

The channels in the closed ring reduce the hydrodynamic pressure necessary to pump back fluid. In Fig. 24 the mechanism is explained.

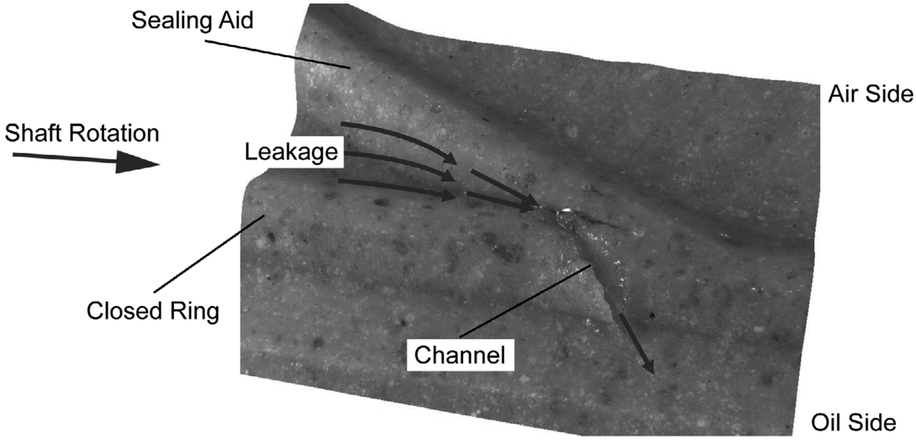


Fig. 24: Closed ring with channel; channel in flow direction

The results of the leak rate measurements for both directions of shaft rotation are shown in Fig. 25. The lip-seals without channels showed leak rates of 2-4 g/h. For the lip-seal with channels in one direction the leak rate was reduced when the channels were in flow direction (CCW). When the channels were in opposite flow direction the leak rate significantly increased. With channels in both directions the leak rate was higher than without channels but still much lower than with 1 channel in opposite flow direction.

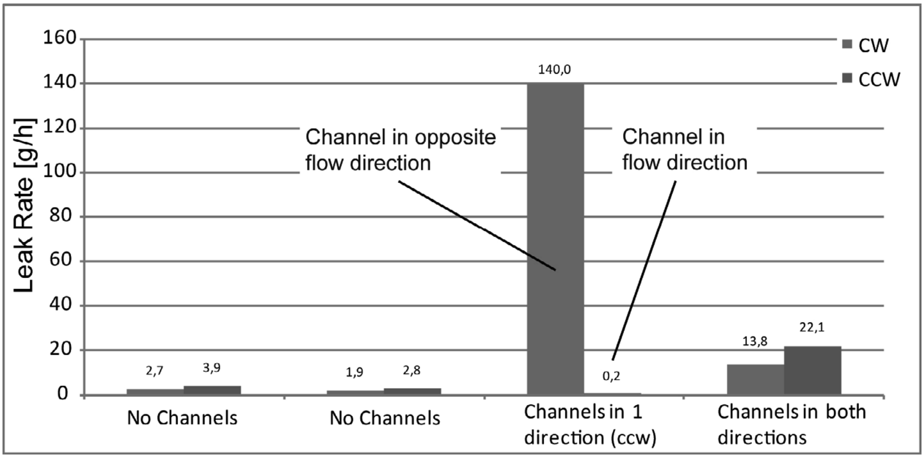


Fig. 25: Leak Rate with different alignments of channels

The results of the measurement with channels in 1 direction show that PTFE lip-seals with bi-directional sealing aids can be leak-tight in dynamic condition. If the hydrodynamic pressure is high enough to lift the closed ring or form channels locally leakage is pumped back. In opposite flow direction the closed ring has to contact the shaft enough to prevent the forming of channels or lifting of the closed ring.

3.3 Conclusions – Experimental Analysis

In the experimental analysis the optimal position of the closed ring and sealing aids was determined. It was proved that bi-directional sealing aids work if they form local channels in the closed ring in flow direction. In static condition these channels have to be closed to prevent static leakage.

4 Comparison – Simulation and Experiment

The results of the experimental investigation with channels match with simulation results. Fig. 26 shows the simulation of a channel in flow direction. Fig. 27 shows the simulation of a channel in opposite flow direction. The channel in flow direction pumps fluid to the oil side. When the channel is arrayed in opposite flow direction oil is pumped from the oil side to the air side. This explains the huge difference in the leak rate for the lip-seal with channels in one direction, Fig. 25.

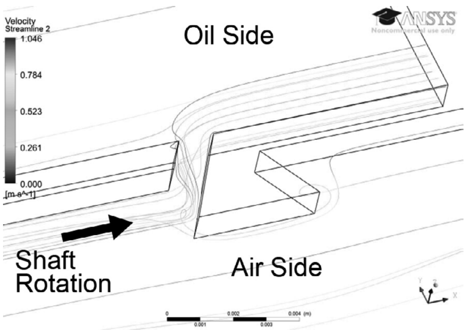


Fig. 26: Channel in flow direction

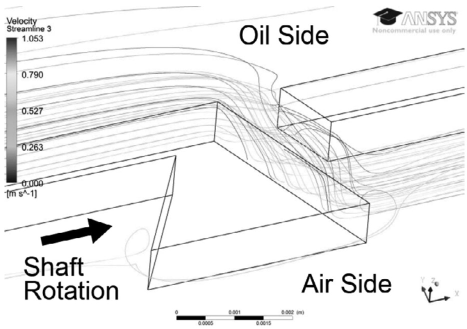


Fig. 27: Channel opposite flow direction

In Fig. 28 a simulation with channels in both directions is shown. Most of the oil that is pumped to the oil side flows back to the air side at the second channel. This explains why the leak rate with channels in both directions is higher than with one channel in flow direction or without channels, Fig. 25.

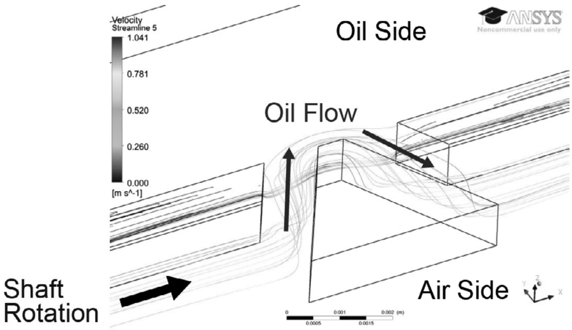


Fig. 28: Channels in both directions

5 Acknowledgements

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6 References / Literaturhinweise

[1] Deutsches Patent- und Markenamt: Offenlegungsschrift DE 101 54 788 A1. Wellendichtring. Deutschland, 2003

[2] Deutsches Patent- und Markenamt: Offenlegungsschrift DE 101 54 789 A1. Wellendichtring. Deutschland, 2003

[3] Bauer, F.: PTFE-Manschettendichtungen mit Spiralrille – Analyse, Funktionsweise und Erweiterung der Einsatzgrenzen, Dissertation Universität Stuttgart, 2008

[4] Haiser, H. B.: PTFE-Compounds im dynamischen Dichtkontakt bei druckbelastbaren Radial-Wellendichtungen, Dissertation Universität Stuttgart, 2001

[5] Jenisch, B.: Abdichten mit Radial-Wellendichtringen aus Elastomer und Polytetrafluorethylen, Dissertation Universität Stuttgart, 1991

[6] Hoffmann, C.: Untersuchungen an PTFE-Wellendichtungen, Dissertation Universität Stuttgart, 1995

[7] Bauer, F; Haas, W.: PTFE Lip-Seals for Bi-directional Rotation, ECOTRIB, Vienna, Austria, June7-9, 2011

[8] Gölz, J.; Bauer, F.; Haas, W.: PTFE Lip-Seals with Bi-directional Sealing Aids, 18th International Colloquium Tribology, Ostfildern-Nellingen, 10.-12. Januar 2012, Book of Synopses 2012, W. J. Bartz (Hrsg.) - TAE Esslingen, S. 157, ISBN 3-924813-97-3