VDMA
Fluid Power Association

22nd ISC

International Sealing Conference

Stuttgart, Germany
October 01 - 02, 2024

Sealing Technology –
Challenges accepted!

IMA UNI STUTTGART

ISBN 978-3-8163-0768-6

Fachverband Fluidtechnik im VDMA e. V. Lyoner Str. 18
50628 Frankfurt am Main
Germany

Phone +49 69 6603-1513
E-Mail maximilian.baxmann@vdma.org
Internet www.vdma.org/fluid

Prediction of RSS Sealing Performance by Fully Coupled EHL Simulation

Fabian Kaiser, Andreas Gropp, Daniele Savio, Daniel Frölich, Christian Wilbs and Dirk Möhring

The pumping and leakage behavior of radial shaft seals (RSS) has challenged researchers for more than half a century. While the main phenomena contributing to reverse pumping have been identified some decades ago, to this day there is no conclusive proof nor a comprehensive model for the exact sealing mechanisms, making it impossible to derive precise guidelines for radial shaft seal design. Therefore, Freudenberg has developed its own simulation tool based on elasto-hydrodynamic lubrication (EHL) to gain insight in the pumping mechanism and the sealing performance of RSS. It includes all steps required to simulate the pumping rate, from Finite Element Analysis to the EHL-modelling of the sealing contact, including the circumferential deformation of the sealing lip roughness. Using this tool, the pump rate of RSS can be simulated accurately and new insights in the sealing mechanism are possible.

1 Introduction

Radial shaft seals are mostly made of elastomer since synthetic nitrile rubber, compatible with mineral oils, became available in the 1930s. Through decades of ongoing research and improvements in materials and design of these seals, some fundamental principles to ensure tightness were established:

- The sealing lip must have an asymmetric shape with a significantly larger oil side angle α than air side angle β (Fig. 1).
- The region of the sealing lip in contact with the shaft requires a certain roughness that must persists even when the seal wears. This is determined by the exact rubber compound.
- The shaft should be hardened to minimize wear, and plunge-ground to remove surface features that might transport oil to the air side.

If these conditions are met, the seal will actively pump fluid from the air side to the oil side, creating an active dynamic sealing mechanism [1]. Several models have been proposed to explain this mechanism, the most popular of which being the asperity deformation model by Kammüller [2]: After installation, an asymmetric contact pressure distribution between the shaft and the seal is formed due to the difference between the angles α and β with the maximum being closer to the oil side. When the shaft is rotated, the elastomer surface is deformed in circumferential direction due to the frictional shear stress with the maximum deformation in the same location as the contact pressure peak. The stretched micro-asperities in the contact create a pumping effect towards the location of maximum pressure. As most of the asperities are located on the air side, a net reverse pumping flow ensues, see Figure 1.

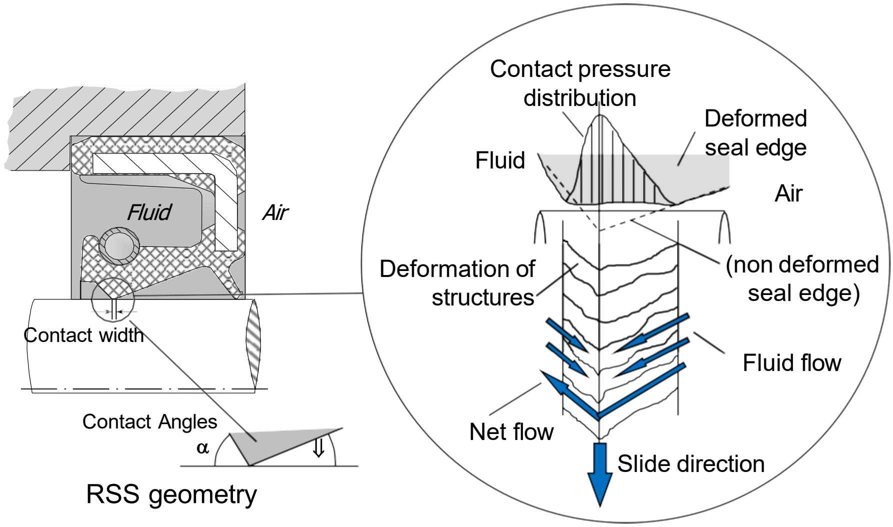

Figure 1: Radial shaft seal and explanation of pumping mechanism according to Kammüller

2 Current State of Research

The asperity deformation model of Kammüller has been verified experimentally to some degree, but local asperity behavior is not fully understood yet. Experimental studies with hollow, transparent shafts were conducted [6] and the pumping effect was demonstrated using e.g., fluorescent particles in the oil. However, the fine details at asperity level are hardly accessible due to the confined space and optical resolution. Additionally, changing the shaft from ground steel to optically smooth glass or sapphire will influence the pumping rate.

Typically, numerical simulation can provide new insights on the underlying effects. For instance, it is standard practice to calculate the contact pressure distribution of a radial shaft seal using finite element methods, but up to now it is impossible to measure contact pressures with µm-resolution.

Still, a comprehensive reverse pumping simulation requires the coupling of several numerical models: 3D elasto-hydrodynamic (EHL) film thickness and flow calculations including cavitation, simulation of the circumferential deformation due to the shear stresses and the contact mechanics between the asperities and the shaft. One of the first to present a full model was Salant [3]. More recently, Thielen developed a model that includes thermal effects and can simulate the friction torque of radial shaft seals accurately [4]. These two models, however, lack an accurate representation of the seal surface roughness, as asperities are assumed to be sinusoidal.

The latest attempt to simulate the pump rate of shaft seals was published by Grün et al. [5]: They use a high-resolution FEA simulation of the deformed surface asperities as input for a transient multiphase computational fluid dynamics (CFD) model.

Thus film thickness and asperity deformation are not coupled to the flow, but calculated from empirical formulas. It is shown that the pump-rate can vary along the circumference of a seal, even leading to local leakage, the average value agreeing well with experiments. This provides new clues as to how a seal can stay well lubricated without leaking.

In summary, no fully coupled model that takes all effects into account exists today. Nevertheless, the existing methods provide very useful insights into the function of radial shaft seals.

3 Materials and Methods

3.1 Seals and Shafts

In this study, standard radial shaft seals with 80 mm shaft diameter were used, made by Freudenberg Sealing Technologies (FST) from a FKM and a NBR material, both having 75 Shore A hardness. The shafts were made from hardened 90MnCrV8 steel (1.2842) and plunge ground to a roughness of Rz 1.5 µm. Figure 2 shows exemplary roughness measurement results of the sealing lip after the tests and the shaft, obtained using confocal microscopy and white light interferometry (WLI) respectively. As expected, the sealing lip is significantly rougher than the shaft surface.

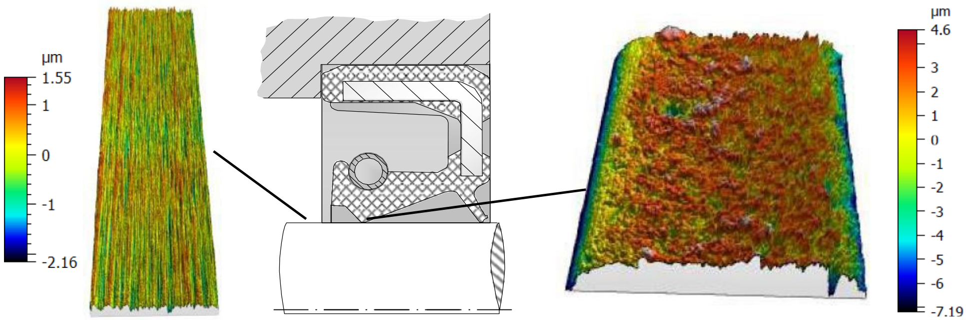

Figure 2: Roughness of ground shaft and sealing edge (shape removed)

The oil used was a commercial mineral oil with 220 mm²/s as nominal kinematic viscosity at 40 °C. At the operating conditions of 60 °C, the dynamic viscosity is 0.069 Pa s.

3.2 Simulation Model

The model presented here aims to couple all relevant effects in a single model.

At first, the contact pressure distribution and deformed geometry of the radial shaft seal is calculated using a FEA method. The rubber material is modelled as hyperelastic at an operating temperature of 60 °C. The resulting pressures and deformed geometries for the two variants can be seen in Figure 3. The NBR seal has the narrower sealing contact compared to the FKM.

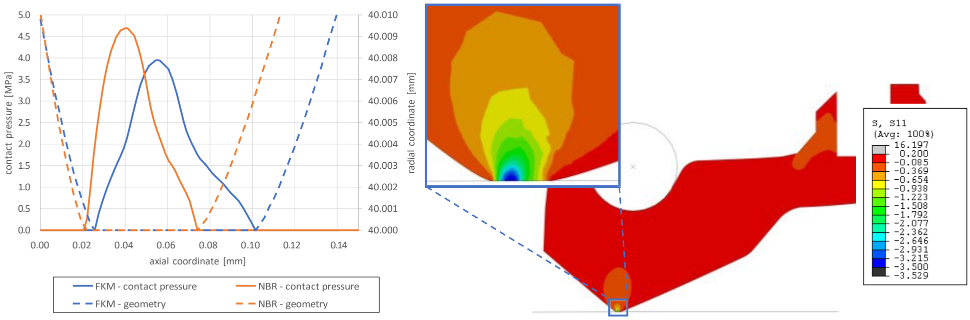

Figure 3: Deformed geometry and contact pressure distributions of analyzed seals.

In order to obtain the local film thicknesses and fluid pressures, the 2D Reynolds equation is solved iteratively, including mass conserving cavitation [10]. For the elastic deformation calculation of the sealing lip asperities, a computationally efficient FFT algorithm is used [11]. From the results, the shear stress acting on the seal can be calculated, being then used as input to compute the circumferential deformation. Following this step, the film thickness is updated, and the process is iteratively repeated until convergence is achieved. For computational reasons, the number of nodes is currently limited to 16,384 (e.g., 128x128). For a contact width of 0.1 mm, this would be equal to a resolution of 0.7 µm, but for other seal types like pressure seals with wider contact bands up to 1 mm, the resolution could be restricted to 8 µm.

Two additional methods are employed to correctly account for asperity contact and fluid flow at scales smaller than the aforementioned resolution limit. First, the contact mechanics theory by Persson and coworkers was used to calculate the asperity load and average separation of the surfaces as a function of the nominal contact pressure [7].

In a similar way, flow factors account for the effect of the roughness on the flow between two surfaces based on Almquist's homogenization method [9]. As stated above, the deformation according to shear stresses is a very important factor in the contact of radial shaft seals. This means that the roughness structures are distorted by a certain angle. Therefore, the existing flow factor calculation is extended to also be a function of this distortion angle, allowing an efficient implementation in the EHL model.

Finally, the 2D-Reynolds equations for steady state condition reads:

$$\nabla\left(-\underline{\underline{A}}_h \nabla p_{fl} - \underline{\underline{B}}_h(\underline{v}_a - \underline{v}_b) + \frac{h_{fl}}{2}(\underline{v}_a + \underline{v}_b)\right) = 0 \tag{1}$$

$\underline{\underline{A}}_h$ and $\underline{\underline{B}}_h$ are the flow factor matrices that are a function of the local distortion angle γ; $\underline{v}_a$ and $\underline{v}_b$ are the velocities of shaft and seal surface.

For the results presented in here, all surface features with a wavelength bigger than 20 µm were fully discretized using the EHL model, while accounting for the smaller features using flow factors and contact mechanics.

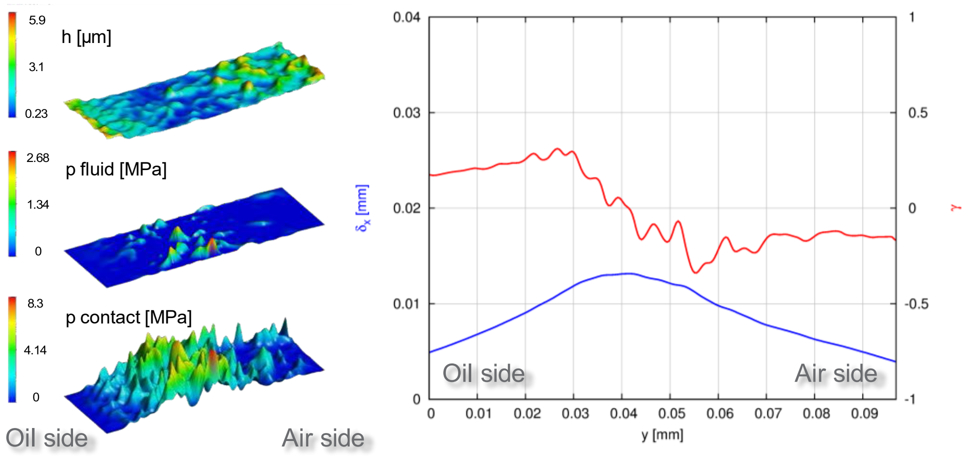

Figure 4: Simulation results for 2 m/s shaft speed: Film thickness distribution h, fluid pressure, contact pressure, circumferential deformation δ_x and distortion angle γ

Figure 4 shows typical results of a pump rate calculation: At a shaft speed of 2 m/s, the tangential deformation is maximal at the location of the contact pressure maximum with a value of 0.012 mm, in line with [source]. The fluid pressure graphs show that despite the full lubrication, most or the load is still supported by asperity contact. The simulated pump rate for this segment is 2.1 ml/h towards the oil side.

Currently, thermal heating of the contact and the changes in viscosity and elastomer stiffness are not considered, as well as starved lubrication, i.e., dry air side. However, the model is built with these extensions in mind, they will be added in the near future.

3.3 Pump Rate Experiments

To validate the simulation results, pump rate experiments were done at different speeds. The pump rate was measured by installing the seal invertedly (oil on air side) and collecting the leakage over 48 hours. The results are shown in Figure 5. The measured pump rates represent the averages of four tests, each test was rotated in clockwise direction and counter-clockwise.

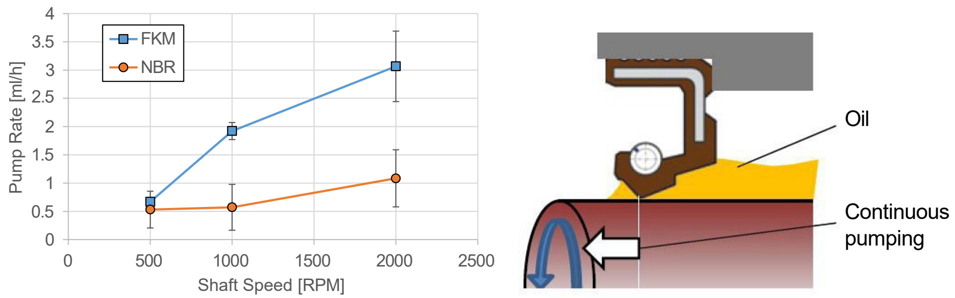

Figure 5: Configuration for pump rate measurement [8] and results

4 Discussion

The simulation of multiple segments of radial shaft seals quickly shows similar results as Grün et al. found [5]: Individual surface patches vary significantly regarding their pumping rate with some even showing leakage. With many surfaces, the average value shows good back-pumping behavior (negative pump rate, Figure 6), in good qualitative agreement with the experiments. It is observed that the pump rate of a segment can change from pumping to leaking depending on the shaft speed.

The discrepancy regarding the absolute values of the pump rate can be explained with the oil viscosity in the sealing gap: Currently, the sump temperature of 60 °C is used to calculate the viscosity for the simulation, but as Remppis has shown [12], the temperature in the contact can be more than 20 °C above the sump temperature. Such an increase would approx. halve the viscosity of the oil used here, reducing the pump rate significantly. This will be considered in a future version of the simulation model. An additional parameter which could explain the discrepancy is the elastomer-steel boundary friction coefficient – currently set to $\mu_{boundary} = 0.2$ as obtained from tribological experiments. Although we consider this a good estimation, the actual value could vary in the radial shaft seal due to the complex contact conditions.

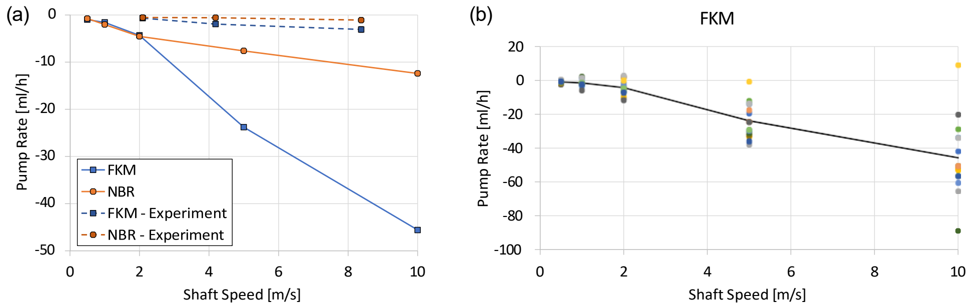

Figure 6: Simulated pump rate vs. measurement: a) Comparison of mean values for simulation and experiment; b) Mean simulated pump rate and values of individual surfaces

The simulations allow a detailed analysis of the flow in the sealing contact. The flow patterns and, crucial for the sealing function, the movement of a virtual particle from the air side to the oil side can be analyzed, see Figure 7. It is apparent, that the main movement of the fluid is in the direction of rotation, with a smaller component perpendicular to it. It is, however, very difficult to identify features on the seal surface that cause the pumping. On average, it takes about 20 mm of circumferential travel to traverse the 0.08 mm wide sealing contact (FKM). Note that these 20 mm are not across several patches but, due to the symmetry boundary condition used in the model, around 200 times across the same patch, entering a bit further towards one side from crossing to crossing due to the pumping. Following a particle across many segments would be very interesting but is computationally unfeasible.

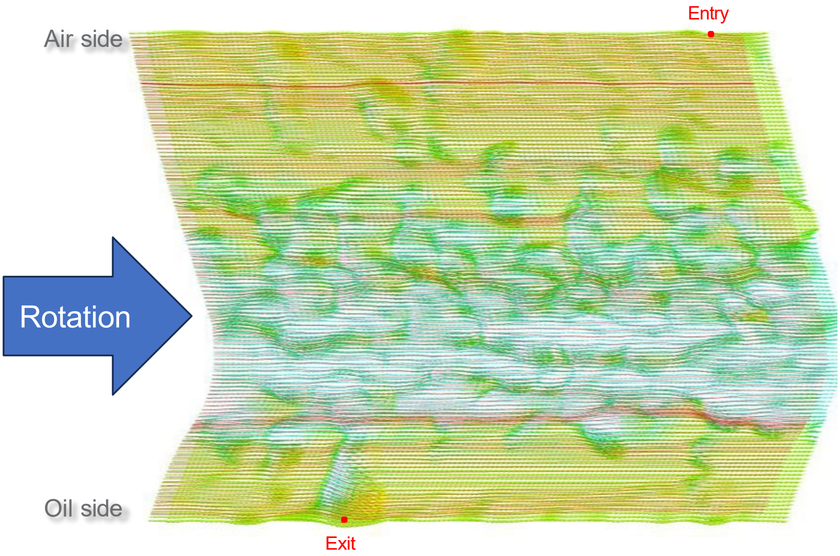

Figure 7: Transversal of virtual particle through the sealing contact

From the findings using the present model and [5], the lubrication and sealing balance of rotary seals needs to be re-thought: Instead of a fixed pumping rate, varying only from part to part, one single seal seems to have many different local pump rates along its circumference, i.e., the seal pump rate variability SPRV. Understanding and being able to control these variations will create opportunities to improve the performance of radial shaft seals: The SPRV may need to be big enough to ensure proper lubrication even at high speeds while still being small enough to avoid leakage.

5 Summary and Conclusion

The model presented here can calculate the film thickness and pump rate of radial shaft seals with good agreement to experimental results and literature. As such, it is one of the first models to comprehensively consider all relevant parameters in a fully coupled way, at least in its planned final configuration. Already now, the findings are leading towards new research aspects to improve the performance of rotary seals, particularly in challenging conditions.

This is very important for Freudenberg to remain a leading manufacturer of high performance products and to offer the best products possible to its customers.

6 Acknowledgements

The authors thank Dr.-Ing. Gerhard Stelzer, MTS, RPTU Kaiserslautern-Landau for providing seal surface measurement data.

7 Nomenclature

Variable	*Description*	*Unit*
p_{fl}	Fluid Pressure	[MPa]
p_{sol}	Solid Body Pressure	[MPa]
h_{fl}	Fluid Film Thickness	[µm]
v	Velocity	[m/s]
A_h, B_h	Influence Coefficient Matrices	[-]
γ	Distortion Angle	[rad]

8 References

[1] Müller, H. K., Nau, B. S.: *Fluid Sealing Technology*, Marcel Dekker, New York, 1998, ISBN 978-0-8247-9969-4

[2] Kammüller, M.: *Zur Abdichtwirkung von Radial-Wellendichtringen*, Dissertation, Stuttgart, Institut für Maschinenelemente, 1986

[3] Salant, R. F.: *Modelling Rotary Lip Seals*, In: Wear, 1997, https://doi.org/10.1016/S0043-1648(96)07463-7

[4] Thielen, S.: *Entwicklung eines TEHD-Tribosimulationsmodells für Radialwellendichtringe*, Dissertation, Kaiserslautern, Lehrstuhl für Maschinenelemente, Getriebe und Tribologie, 2019

[5] Grün, J., Feldmeth, S., Bauer, F.: *Multiphase Computational Fluid Dynamics of Rotary Shaft Seals*, In: Lubricants, 2022, https://doi.org/10.3390/lubricants10120347

[6] Mizuta, H., Furuyama, H., Sato, Y.: *Measurement Method of Pumping Ability of Radial Lip Seals for Small Shaft Diameters*, SAE Technical Paper 2007-01-1521, 2007, https://doi.org/10.4271/2007-01-1521

[7] Yang, C., Persson, B.N.J.: *Contact mechanics: contact area and interfacial separation from small contact to full contact*, J. Phys.: Condens. Matter, 2008 (20)

[8] Remppis, M.: *Untersuchungen zum Förderverhalten von Dichtsystemen mit Radial-Wellendichtringen aus Elastomer*, Dissertation, Stuttgart, Institut für Maschinenelemente, 2016

[9] Almqvist, A.: *On the effects of surface roughness in lubrication*, PhD-Thesis, Luleå University of Technology, 2006

[10] Elrod, H. G.: *A Cavitation Algorithm*, J. of Lubrication Tech. Jul 1981, 103(3): 350-354, https://doi.org/10.1115/1.3251669

[11] Wang, Q. J., Sun, L., Zhang, X., Liu, S., Zhu, D.: *FFT-Based Methods for Computational Contact Mechanics*, Front. Mech. Eng., 2020, https://doi.org/10.3389/fmech.2020.00061

[12] Remppis, M.: *Untersuchungen zum Förderverhalten von Dichtsystemen mit Radial-Wellendichtringen aus Elastomer*, Dissertation, Stuttgart, Institut für Maschinenelemente, 2016

9 Authors

Freudenberg Technology Innovation SE & Co. KG

Höhnerweg 2-4, 69469 Weinheim, Germany:

Dr.-Ing. Fabian Kaiser, ORCID 0009-0003-7525-7062, fabian.kaiser@freudenberg.com

Dipl.-Ing. Andreas Gropp, andreas.gropp@freudenberg.com

Dr.-Ing. Daniele Savio, ORCID 0000-0003-1908-2379, daniele.savio@freudenberg.com

Freudenberg FST GmbH

Höhnerweg 2-4, 69469 Weinheim, Germany:

Dr.-Ing. Daniel Frölich, ORCID 0000-0002-2707-948X, daniel.froelich@fst.com

Christian Wilbs, M.Sc., christian.wilbs@fst.com

Dipl.-Ing. (FH) Dirk Möhring, dirk.moehring@fst.com

https://doi.org/10.61319/2OAHH0DH