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Reducing Friction and Leakage by Means of Microstructured Sealing Surfaces – Example Mechanical Face Seal

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Abstract

By defined structuring of sliding surfaces at dynamic contact seals friction and leakage can be reduced. Compared to macro-structures, micro-structures have the advantage of a quasi-homogeneous influence on the fluid behavior in the sealing gap. The development of suitable microstructures based on prototypes, whose properties are studied on the test bench, is very expensive and time-consuming due to the challenging manufacturing process and measuring technologies, which are necessary to investigate the complex rheological behavior within the sealing gap. A simulation-based development of microstructured sealing surfaces offers a cost- and time-saving alternative. This paper presents a method for simulative design and optimization of microstructured sealing surfaces at the example of a microstructured mechanical face seal.

KEYWORDS: Surface Microstructuring, Surface Texturing, Mechanical Face Seals

1. Introduction

For about one decade, functionalization of surfaces by means of micro-structures has been subject of intensive research /1/, /2/, /3/. The positive effect of improving the hydro- and thermohydrodynamic behaviour of a technical system by influencing the fluid in the proximity of walls microscopically has been demonstrated in many cases /4/. New microproduction techniques like laser structuring with ultra-short-pulse lasers in combination with multi-beam processing establish themselves in the market and enable an economic microstructuring of hard metallic surfaces without the need of additional post-processing /5/. The big challenge today is the development of most suitable microstructures for specific applications. Several structures have already been published /6/ and the hydrodynamic influences have been investigated /7/. Methods to design structures for specific applications, however, are very limited and only a few numbers of scientific work exist /8/, /9/. Furthermore, multiphysical simulation methods considering the influence of the dynamics and deformation of the contact partners on the hydrodynamic conditions in the lubrication gap are not available.

Mechanical face seals, a special type of axial face seals, are typically used as primary seals between relatively rotating parts in heavily contaminated environments. Typical applications are wheel drives of earthmoving machinery like wheel loaders and dumpers. Mechanical face seals consist of two metal rings which are mounted on a rotating shaft and inside the bore of a stationary housing. The sliding surfaces of both rings are lapped and preloaded by O-rings which additionally function as secondary seals between the metal rings and the hub or housing (**Figure 1**).

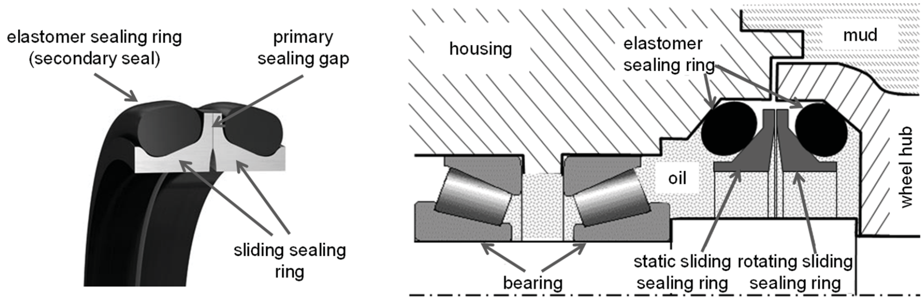

Figure 1: Left: Mechanical face seal; Right: Wheel hub with mechanical face seal

Mechanical face seals are commonly made of hard cast iron and require a lubrication of the sliding surfaces. When lubricated by oil, the sliding speed for reliable applications is

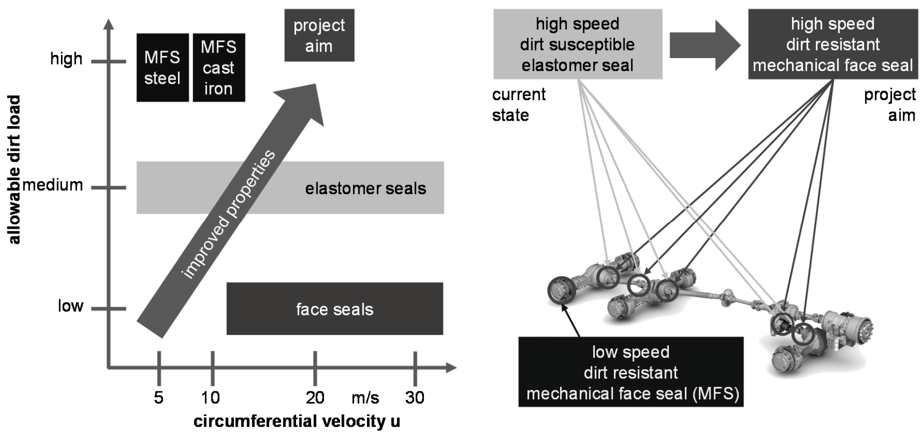

Figure 2: Seals in the area of conflict between circumferential velocity and dirt load

limited to about 3 m/s. For short times, sliding velocities of up to 10 m/s can be reached. Hence, the field of operation of mechanical face seals is limited to slowly rotating applications like wheel drives (**Figure 2**). For high-speed components, elastomeric radial shaft seals are used nowadays. These seals, however, do not reach the desired lifetime due to the heavily contaminated environment. Therefore, a new high-speed mechanical face seal for solid particle contaminated environments has been developed in a public funded (BMWi) research project.

2. Approach

The increased sliding speed requires a significant friction reduction within the lubrication gap. Conventional mechanical face seals operate in solid or mixed lubrication with a high solid contact ratio (**Figure 3**). In order to decrease the friction within the lubrication gap, a concept for the functionalization of the sealing surfaces has been developed. This concept is based on three different functional surfaces. The first functional surface (a) reduces the friction. Well-defined microstructures on the sliding surface acts as oil reservoir and generate a hydrodynamic lubrication film, so that even for low sliding speeds, the friction regime is shifted to the more advantageous fluid friction.

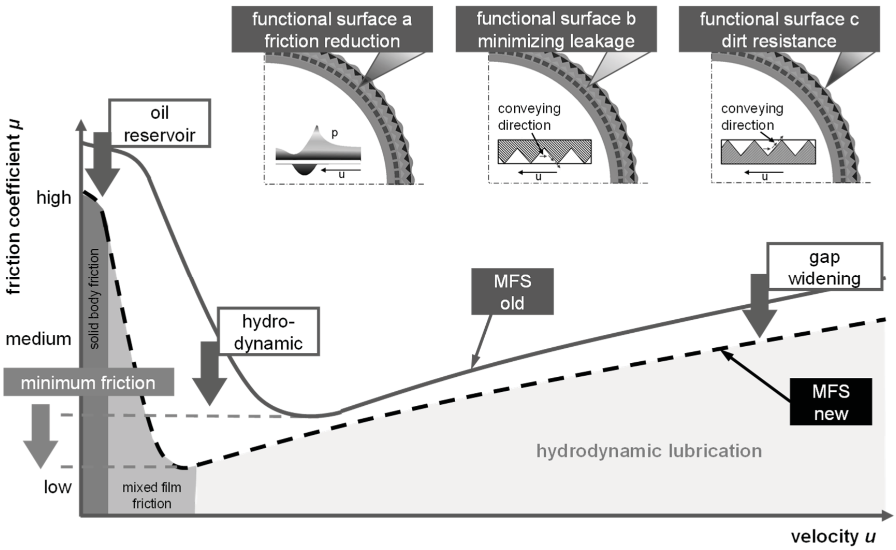

Figure 3: Coefficient of friction μ depending on the sliding velocity u (Stribeck-curve)

As a matter of principle, the lubrication gap expands. Therefore, a second functional surface (b) is used, which intends minimizing the occurring leakage. For this, conveying microstructures are applied, which influence the fluid flow within the lubrication gap.

Finally, the third functional surface (c) increases the dirt resistance by transporting solid particles out of the sealing.

In this contribution, only the functional surfaces a and b will be described. For information about the design of the functional surface c, which increases the dirt resistance, see /10/.

3. Method

For designing the functional surfaces a and b, a simulation method has been development to investigate the influence of the sliding surface topography on the friction and leakage (**Figure 4**). Based on an iterative procedure, the sliding surface topography is optimized with respect to low friction in the first step (**Figure 4**A). This friction-optimized topography is then used as an input for the second step, in which the topography is optimized for leakage (**Figure 4**B). The new topography is then given back to the friction optimization and a new iteration is started.

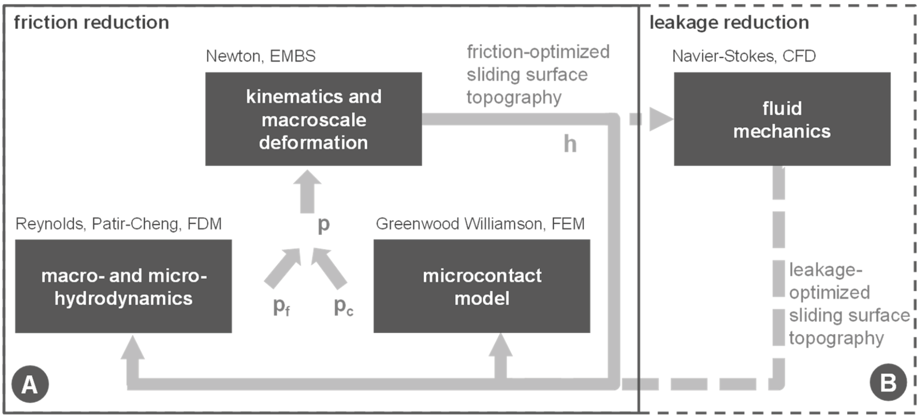

Figure 4: Simulation method for designing and optimizing hydrodynamically effective, microstructured sealing surfaces /11/

4. Functional surface „a“ – friction reducing

In addition to the topography of the sliding surface and the hydrodynamics within the lubrication gap, the friction is significantly determined by the kinematics of the sealing components and their deformation. This deformation can be in the same range as the lubrication gap height. Hence, a coupled simulation of the structural dynamics, structural mechanics and hydrodynamics becomes necessary. Due to this, the mechanical face seal has been modeled as an elastic multibody system (EMBS) with coupling to the microelastohydrodynamics inside the sealing contact. Compared to classical multibody systems, in which only rigid bodies are investigated, the EMBS considers the deformation of bodies.

In principle, the modeling approach in EMBS can be divided into two separate branches. In the first branch, the technical system is described as a MBS based on kinematic, kinetic and mechanical properties. In the second branch, the elastic solids are modeled as continuous bodies and discretized into small (finite) elements, whose spatial positions are described by nodes (**Figure 5**).

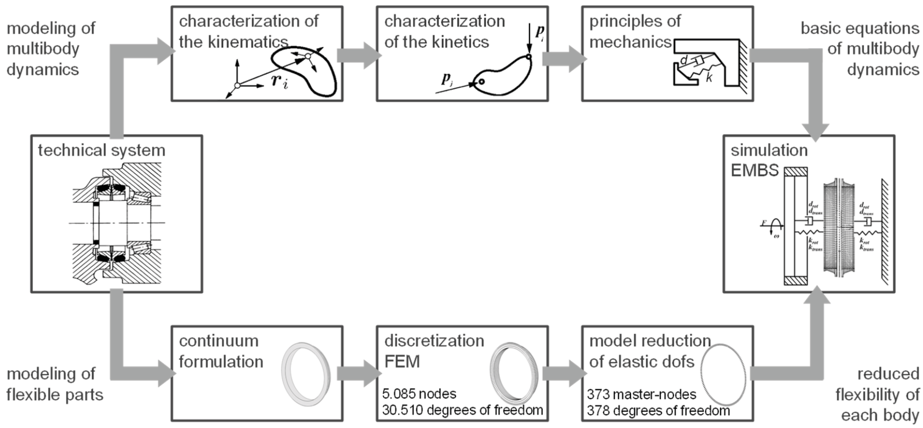

Figure 5: Modeling of a technical system as EMBS /cf. 12/

For the rigid body, the equation of motion consists of three translational and three rotational degrees of freedom (DOF). For the FE-model of the flexible body, these six DOF must be considered for each node, which leads to a considerable computational effort. Therefore, the number of DOF of the flexible body is usually decreased before it is used in an EMBS simulation. For this purpose, relevant DOF are defined as so called master DOF and the remaining slave DOF are reduced. The deformation, which was previously defined by local shape functions, is then calculated by global shape functions. Hence, the equation of motion (1) becomes /13/:

$$\boldsymbol{M}^{red}\ddot{s} + \boldsymbol{D}^{red}\dot{s} + \boldsymbol{K}^{red}s = f^{red} \tag{1}$$

The reduced flexible bodies have markers on which external forces can be applied /13/. For pressure forces, the sealing surfaces are discretized and the contact pressure p is calculated. This pressure consists of the hydrodynamic pressure p_f under mixed lubrication and the additional solid body contact pressure p_c (2):

$$p = p_f + p_c \tag{2}$$

The integral contact pressure acting on the body is transferred to the master node, so that the overall equation of motion (3) can be written as:

$$\boldsymbol{M}^{red}\ddot{s} + \boldsymbol{D}^{red}\dot{s} + \boldsymbol{K}^{red}s = f_{ext}^{red} + f_p^{red} \tag{3}$$

In this equation $\boldsymbol{M}$ represents the mass matrix, $\boldsymbol{D}$ the damping matrix, $\boldsymbol{K}$ the stiffness matrix, s is the vector containing the degrees of freedom, f_{ext}^{red} is the external force vector and f_p^{red} represents the contact pressure vector. For a detailed derivation of this equation, see /11/ and /14/.

The solid contact pressures acting on the sealing rings are determined by the micro contact model of Greenwood and Williamson /15/. In this model, the surface roughness profiles of two contacting bodies are approximated by spherical caps and transferred to one deformable surface, while the other surface is assumed to be rigid and perfectly smooth. By using the Hertzian theory, the solid contact pressure p_c^{lokal} can then be determined depending on the lubrication gap height. Based on this, the solid contact ratio can be determined and the friction torque (4) due to solid contact can be calculated as follows:

$$M_{Reib,c}^{1,2} = \mu \int_{\Omega} p_c r dA \tag{4}$$

For the hydrodynamic pressure, the Reynolds equation is solved. In order to avoid an expensive discretization of the microstructured sliding surface, the microstructural influence is decoupled from the macro hydrodynamics. In this indirect coupling approach based on the theory of Patir and Cheng /16/, the influence of microstructures on the micro hydrodynamics inside the lubrication gap is determined only on a small representative part of the surface by means of flow factors. In this way, the micro hydrodynamic influence on the fluid flow can be considered in a macro hydrodynamic simulation. For this, the Reynolds equation (5) is modified to include the pressure flow factor Φ^p and the shear flow factor Φ^s:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{h^3}{12\eta}\Phi_r^p\frac{\partial p}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \varphi}\left(\frac{h^3}{12\eta}\Phi_\varphi^p\frac{\partial p}{\partial \varphi}\right) = \frac{\partial h}{\partial t} - \frac{1}{2}\omega\frac{\partial}{\partial \varphi}\left(h + \Phi_\varphi^s\right) \tag{5}$$

The hydrodynamic shear stresses (6) can then be determined by:

$$\tau_{f\varphi}^{1,2} = -\omega\frac{\eta}{h}\left(\Phi_f \pm \Phi_\varphi^{fs}\right) \pm \frac{h}{2}\frac{\partial p}{\partial \varphi}\Phi_\varphi^{fp} \tag{6}$$

with the shear stress factors Φ_{φ}^{fs} and Φ_{φ}^{fp} and the correction factor Φ_f (see /17/). The positive sign represents the stationary and the negative sign the sliding sealing ring. Hence, the hydrodynamic friction torque (7) becomes:

$$M_{Reib,f}^{1,2} = \int_{\Omega} \tau_{f\varphi}^{1,2} r d\varphi dr \tag{7}$$

For mixed lubrication, the overall friction torque (8) can then be calculated by a simple summation of the solid friction and the hydrodynamic friction torque:

$$M_{Reib,m}^{1,2} = M_{Reib,f}^{1,2} + M_{Reib,c}^{1,2} \tag{8}$$

As a result, this part of the simulation method (Figure 4A) is able to optimize the sliding surface topography with respect to friction.

5. Functional surface „b“ – leakage reduction

In addition to the requirement of low friction, the demand for minimal leakage has to be fulfilled. As a matter of principle, hydrodynamic microstructures increase the lubrication gap. Therefore, an additional functional surface is used, which transports radially occurring leakage back into the lubrication gap. Such recirculating micro structures are state of the art for radial shaft seals. For mechanical face seals, only a few number of macroscopic structures have been published /18/, /19/, /20, /21/. These structures, however, produce high local pressure gradients and cannot offer a homogenous recirculation due to its macroscopic nature. Therefore, new microscopic microstructures have been developed.

Due to the assumptions ($\frac{\partial p}{\partial z} = 0$, $\frac{\partial w}{\partial z} = 0$) made for the derivation of the Reynolds equation /17/, equation (4) is not able to simulate occurring leakage accurately enough. Contrary to the simplifications assumption, three dimensional flow and pressure fields exist /22/, which also influence the amount of leaked fluid /23/. With respect to the intended manipulation of the fluid flow, a three dimensional hydrodynamic simulation becomes necessary.

Therefore, a CFD simulation model has been build up, which represents the microstructures inside the lubrication gap. In **Figure 6**, two exemplary structures are shown. The rear boundary of the fluid space is formed by the microstructured sealing ring. It moves in negative x-direction. The front boundary is formed by the unstructured sealing ring. It is fixed. In Figure 6, the computation grid of discretized fluid space can be seen. The recirculating feed structures shown are recesses. Under the structures additional x-z-planes are placed. These planes illustrate the velocity fields in radial direction y. At first, the fluid flows from bottom left radially inside the structures (positive y-direction). It is then diverted and finally, it flows to the bottom right back into the sealing gap (negative y-direction).

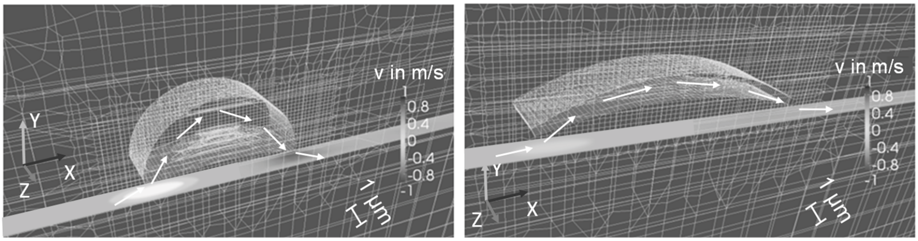

Figure 6: Velocity field of CFD simulation of two recirculating microstructures with convex/ concave edge geometry and different lengths (l.: 100 µm, r.: 300µm)

The influence of the geometry, the size and the depth of the structures have been investigated and a sensitivity analysis has been performed. As an example, the influence of the structural height is illustrated in **Figure 7**.

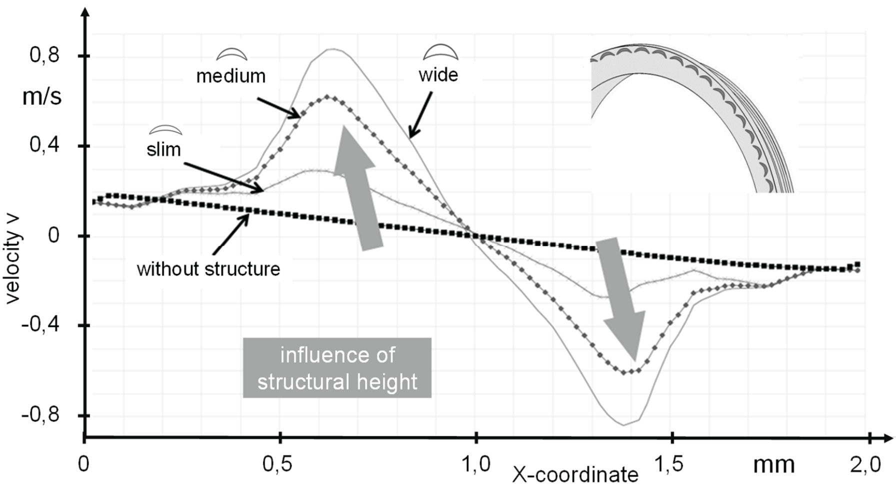

Figure 7: Influence of structural height to the fluid flow

In addition, the entire lubrication gap including all microstructures has been optimized with respect to minimal leakage in further CFD-simulation. This simulation yields the leakage-optimized sliding surface topography, which then serves as an input for the simulation of friction reduction (Figure 4A). By applying this iterative procedure, the bidirectional influence of the friction-reducing and leakage-reducing microstructures can be investigated and at the end, the optimized sliding surface topography under consideration of critical operation points can be found.

6. Results

In **Figure 8**, a prototype of the developed mechanical face seal is shown. The structures have been manufactured by ultra-short pulsed laser. Compared to classical laser techniques, ultra-short pulsed lasers can produce high quality surfaces without melt residue, so that expensive post-processing can be avoided. Due to the high resolution of this technique, the numerically developed microstructures could be inserted into the mechanical face seals very precisely.

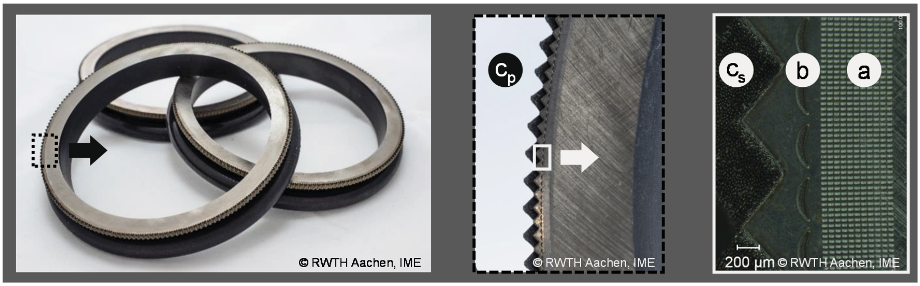

Figure 8: Sliding sealing ring with microstructured functional surfaces

The right picture in Figure 8 shows the microstructured sliding surface. The functional surface a is provided with oval dimples (L = 25 µm, W = 50 µm, D = 5 µm). The functional surface b is provided with sickle-shaped grooves (L = 400 µm, W = 50 µm, D = 20 µm). In order to convey into the sealing gap penetrating dirt back to the outside, the seal has a primary dirt conveying structure for large particles (c_p) and a secondary structure (c_s) for small particles.

For validation, the prototypes have been tested on a specific MFS-test bench (**Figure 9**).

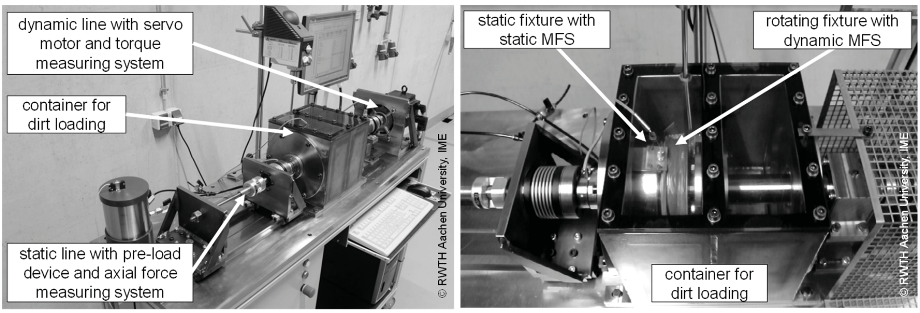

Figure 9: Testing of mechanical face seals on test bench with dirt-loading

In first trials, the friction could be reduced by up to 25 % without any leakage. The left of **Figure 10** shows the friction torques of a conventional and a microstructured mechanical face Seal at sliding velocities up to 20 m/s. The conventional MFS already failed at about 3.5 m/s through warm scuffing. This reflected in the friction torque peaks. Due to the breaking away opens the sealing gap which massive leakage. In the gap penetrates fresh oil and lubricants for a short time, so that the friction torque falls down until the next contact occurs. The conventional MFS failed finally at 16 m/s through flying sparks. The right of Figure 10 shows the scuffing marks at the sealing sliding surface. Because of the higher lubricant thickness and a better heat removal from the sealing gap, the microstructured MFS achieved without damage 20 m/s. Additionally acting the structures as a particle trap for wear particles.

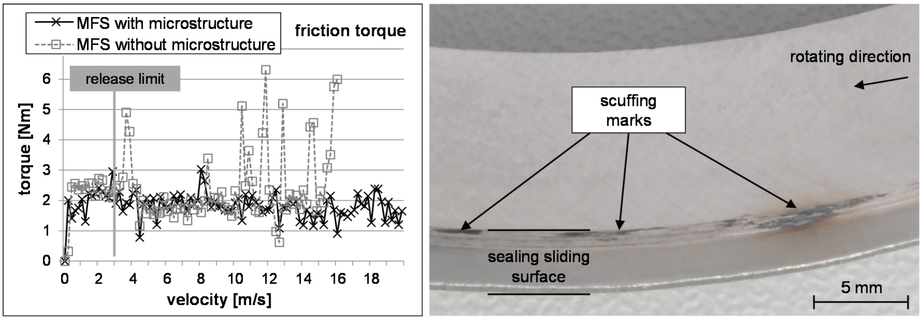

Figure 10: Left: Friction torque with microstructure and without microstructure; Right: Failed unstructured MFS with scuffing marks as result of warm scuffing

The microstructured seals have not yet been studied in longer-term tests hitherto, so that statements for wear and life time not yet available.

An actually disadvantage are still the high production costs. For economical production, they must be significantly reduced. Multi beam scanner offers here excellent opportunities.

7. Summary and Outlook

Hydrodynamic microstructures show a big potential to decrease energetic losses in high-friction applications. Although this topic has been addressed in many research projects, only a small number of methods exist, which assist the developers of microstructural surfaces to transfer the fundamental knowledge to specific applications.

In this contribution, a simulation method has been proposed, which allows the design of microstructured mechanical face seals with respect to friction and leakage reduction. The optimized properties of such mechanical face seals have been demonstrated in first trials on a test bench. Regarding the huge amount of oil-lubricated, high-friction applications, a transfer of the proposed simulation method to further dynamic seals is imaginable.

8. Acknowledgment

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10. Nomenclature

A	Area	m²
$\boldsymbol{D}^{red}$	Reduced damping matrix	N s/m
f	Force	N
f^{red}	Reduced contact pressure load vector	N
h	Clearance Height	m
$\boldsymbol{K}^{red}$	Reduced stiffness matrix	N/m
M_{Reib}	Friction torque	Nm
$\boldsymbol{M}^{red}$	Reduced mass matrix	kg
p	Pressure	Pa
p_f	Fluid pressure	Pa
p_c	Surface pressure	Pa
r	Radius	m

s	degree of freedom vector in reduced system	m
u	Velocity (x-direction, circumferential direction)	m/s
v	Velocity (y-direction)	m/s
w	Velocity (z-direction)	m/s
η	Dynamic viscosity	Pa s
μ	Friction coefficient	-
τ	shear stress	Pa
φ	Angle	°
Φ^p	Pressure flow factor	-
Φ^s	Shear flow factor	-
Φ^{fp}	Shear stress factor of the pressure flow	-
Φ^{sf}	Shear stress factor of the shear flow	-
Φ^f	Correction factor	-
ω	Angular velocity	sec^{-1}