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Simulation of the Fluid Flow in the Sealing Gap of Radial Lip Seals

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1 Introduction

Radial lip seals are used to retain fluid or grease in machines with rotating shafts and to prevent dirt ingress into the machine [1], [2]. The radial lip seal, the surface of the shaft and the fluid which has to be sealed (usually oil) form a sealing system, which is significantly affected by the environment and the operating conditions.

1.1 Radial lip seals

Elastomeric radial lip seals are standardized in national and international standards [3]. They consist of a metal insert to which a sealing lip is attached, Figure 1. During assembly, the sealing lip and a garter spring are widened. The sealing lip is pressed against the shaft surface, so that a slim contact area is formed. The width of the contact area is $b \approx$ 0.1 to 0.2 mm. The average contact pressure in the contact area is $p_m = F_r/\pi b d \approx$ 1 MPa.

During operation, a thin lubrication film develops in the contact area between the sealing lip and the shaft surface. This lubrication film separates the surfaces and forms the so-called sealing gap. A microscopic pumping mechanism of the elastomer in the contact area prevents leakage.

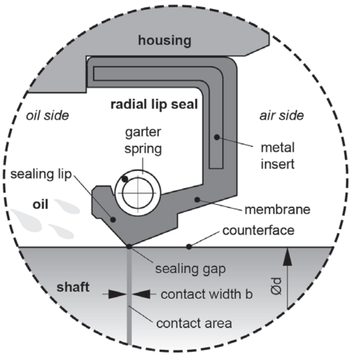

Figure 1: Sealing system consisting of radial lip seal, shaft and fluid

Frictional heat and contact temperature

During operation, frictional heat is generated in the contact area of the sealing system. The more frictional heat is generated and the poorer this frictional heat is dissipated from the contact area, the hotter the contact area becomes. High temperatures in the contact area are very harmful to the sealing system. They accelerate the aging of the elastomer of the radial lip seal and therefore they reduce the lifespan of the sealing system. In order to achieve a long lifespan, moderate temperatures in the sealing system – and especially in the contact area – must be intended during the development of machines.

However, a reliable and economic design of the sealing system is only possible, if the temperature in the contact area is exactly known. One possibility to determine the contact temperature of a sealing system is the measurement during test runs. This requires a test rig with measuring equipment and is both expensive and time consuming. The second possibility to predict the contact temperature is the numerical simulation of the heat generation and heat dissipation. This method allows an accurate prediction of the contact temperature without expensive and time consuming test runs.

At the Institute of Machine Components at the University of Stuttgart, a simulation model for predicting the temperature in radial lip seal systems was developed and is continually refined and expanded [4], [5], [6]. This simulation model is described in the following chapter.

Simulation model for temperature prediction

To predict the contact temperature correctly, the simulation model must cover all relevant mechanism of heat generation and heat dissipation. This is achieved by a multi scale approach. The multi scale model consists of two models on different length scales: a micro model and a macro model. Both models are described below.

1.2 Macro model

The macro model covers the whole sealing system as well as parts of its environment and simulates the heat transfer from the contact area to the environment, Figure 2. In the macro model, the concept of conjugate heat transfer (CHT) is used, which combines the simulation of heat transfer in fluids by computational fluid dynamics (CFD) simulation and the simulation of heat transfer in the adjacent solids. Within this method, the convective heat transfer in the fluids around the sealing system is simulated using the continuity equation, the Navier-Stokes equations and the energy equation. The conductive heat transfer in the solids of the sealing system and its environment is simulated using the heat equation including Fourier's law. ANSYS CFX is used as simulation software for the macro model.

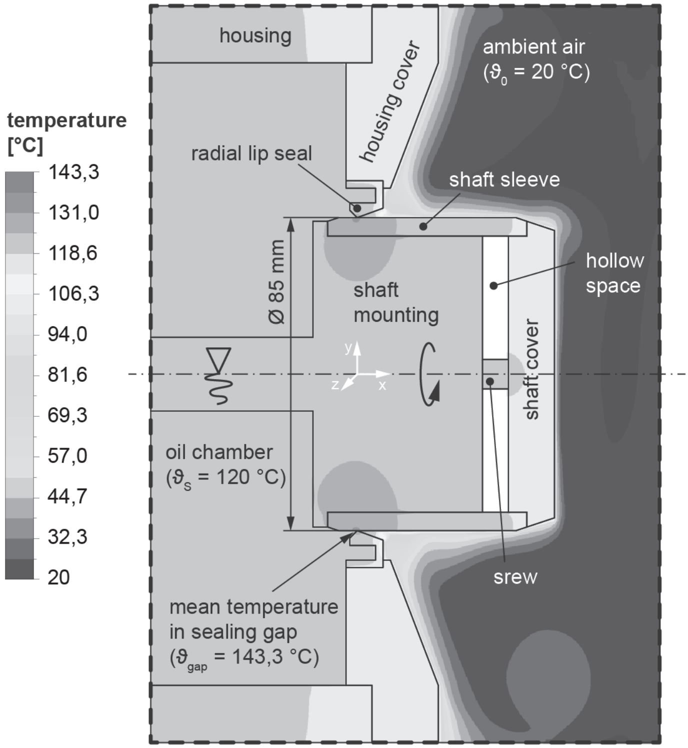

Figure 2: Simulated temperature in the center plane of a sealing system [6]

1.3 Micro model

The micro model solely covers the sealing gap and is used to estimate the frictional losses, which result mainly from the shearing of the oil film in the sealing gap. In the past, different approaches were developed to determine these frictional losses:

- Measuring the friction torque at a test rig for every single operating point and thus calculating the frictional losses [4].
- Using an empirical relation derived from previous friction torque measurements of similar sealing systems and interpolating the friction torque [6].
- Simulating the fluid flow in the contact area between the sealing lip and the shaft surface with calculation of the shear stress acting on the shaft surface.

Both, the first and the second approach require expensive and time consuming test runs. As an alternative the third approach is currently being developed at the Institute of Machine Components (IMA) at the University of Stuttgart and is described in detail in chapter 3. This development is done within the research project “Multi-Scale Simulation Model for the Temperature Prediction in Radial Lip Seals and their Environment” (Multiskalen-Simulationsmodell zur Temperaturvorhersage im Dichtsystem

Radial-Wellendichtung, HA 2251/27-1) funded by the German Research Foundation (DFG, Deutsche Forschungsgemeinschaft).

1.4 Coupling of macro and micro model

The processes of heat generation and heat dissipation are strongly coupled by the viscosity temperature relation of the fluid that has to be sealed. Due to frictional heat, the contact temperature increases. This temperature rise strongly affects the fluid viscosity in the sealing gap as well as the generation of frictional heat. To resolve this coupling of heat generation and heat dissipation, an iterative solution is chosen. The iteration steps of the macro and the micro model are performed alternately. Each simulation model uses the results, generated in the previous step by the other model as a boundary condition, Figure 3.

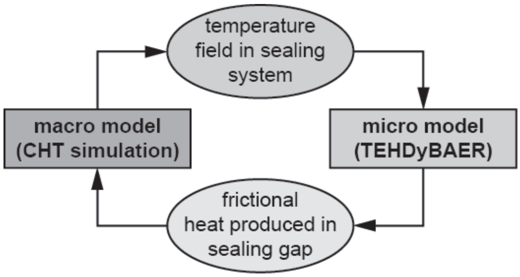

Figure 3: Coupling of macro and micro model

This paper focuses on the micro model, which is described in detail in the following chapter.

2 Micro model for the fluid flow in the sealing gap

The micro model, simulating the fluid flow in the sealing gap is written in FORTRAN and is called TEHDyBAER (German abbreviation for “Thermo-Elasto-Hydro-Dynamischer Berechnungs-Ansatz für die Energiedissipation in Radial-Wellendichtungen” meaning “thermo-elasto-hydrodynamic calculation of the heat generation in radial lip seals”). Although the program name suggests a thermal calculation, currently the program assumes isothermal conditions (and therefore also incompressible fluids).

TEHDyBAER is based on the multigrid method described by Venner and Lubrecht [7] to accelerate the convergence speed of the iterative process. This is achieved by using multiple grids with different fine meshes. These grids are used alternately to solve the discretized equations, with the results transferred from one grid to the next by interpolation or restriction. On each grid the high frequency errors are reduced quite fast, resulting in a good overall convergence, even for fine meshes. The systems of linear equations are solved using routines provided by the linear algebra package LAPACK.

The definition of the computational domain used by the micro model TEHDyBAER is shown in Figure 4.

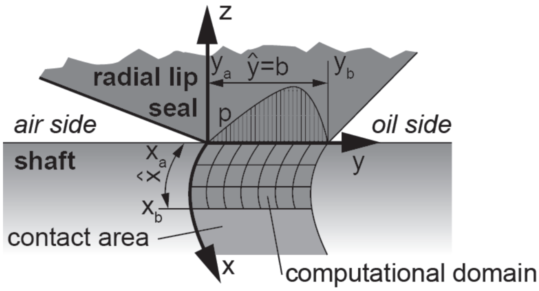

Figure 4: Definition of the computational domain used by the micro model

The simulation model TEHDyBAER is based on several equations which describe all relevant processes occurring in the contact area of radial lip seal and shaft surface. These equations will be presented in the following.

2.1 Governing equations

The fluid flow in the sealing gap is described by the Reynolds equation which can be seen as a simplification of the Navier Stokes equation for laminar flow in narrow gaps. The Reynolds equation is a differential equation containing the gap height h, the fluid pressure p, the fluid density ρ (which is assumed as constant) and the fluid viscosity η as well as the relative velocity of the shaft u_s and the coordinates x and y (for circumferential and radial direction),

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{6\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{6\eta}\frac{\partial p}{\partial y}\right)-u_s\frac{\partial(\rho h)}{\partial x}=0\,.$$

The gap height is defined as the sum the following four parts, Figure 5:

- the average distance h_0 between the sealing lip and the shaft surface
- the shape of the shaft including the superimposed surface roughness $h_1(x,y)$
- the shape of the sealing lip including the superimposed surface roughness $h_2(x,y)$ and
- the deformation (mainly of the sealing lip) $w(x,y)$.

The deformation $w(x,y)$ is calculated using the theory of the elastic half-space and depends on the reduced elasticity E' and the pressure in the whole contact area (boundaries x_a and x_b in x direction, boundaries y_a and y_b in y direction),

$$h(x,y) = h_0 + h_1(x,y) + h_2(x,y) + \frac{2}{\pi E'} \int_{y_a}^{y_b} \int_{x_a}^{x_b} \frac{p(x',y')dx'dy'}{\sqrt{(x-x')^2 + (y-y')^2}}.$$

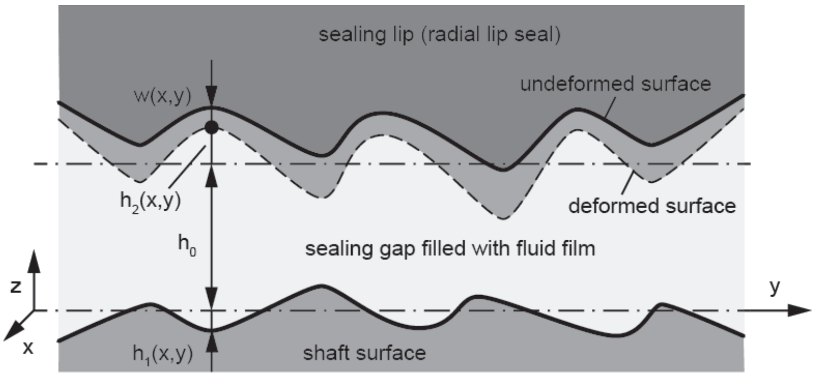

Figure 5: Definition of the gap height

Assuming linear elastic behavior of both solid materials (shaft and sealing lip), the reduced elasticity E' can be calculated with the moduli E and the Poisson ratios ν of the shaft material (index 1) and the elastomer of the sealing lip (index 2),

$$\frac{2}{E'} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}.$$

Since the shaft material is usually steel (or another kind of metal), the modulus E_1 is about five orders of magnitude higher than the modulus E_2 of the elastomeric sealing lip. Additionally, elastomers can be assumed as incompressible ($\nu_2 = 0.5$). Using these assumptions the reduced elastic modulus simplifies to

$$\frac{2}{E'} \approx \frac{3/4}{E_2}.$$

The viscosity of many fluids which have to be sealed (e.g. oil) is strongly affected by the temperature. The viscosity temperature relation of oils can be represented by the Ubbelohde-Walther equation wherein the kinematic viscosity ν [mm²/s] is a function of the absolute temperature T [K], a constant $c \approx 0.8$ and two fluid specific parameters K_ν and m,

$$\log_{10}(\log_{10}\nu + c) = K_\nu - m \cdot \log_{10}(T).$$

These two parameters are dimensionless and must be calculated using two measured values of the viscosity at two different temperatures.

In order to satisfy the equilibrium of forces in radial direction, the fluid pressure, integrated over the whole contact area, must correspond to the radial force F_r of the radial lip seal. If the computational domain does not cover the whole contact area but only a part in circumferential direction (from x_a to x_b), the ratio of the areas must be taken into account,

$$F_r = \frac{\pi d}{x_b - x_a} \int_{y_a}^{y_b} \int_{x_a}^{x_b} p(x, y)\, dx\, dy\,.$$

2.2 Dimensionless variables and characteristic quantities

In order to reduce the number of parameters in the governing equations, dimensionless variables are introduced. The dimensionless variables (labeled with an overbar) are defined using characteristic quantities (labeled with a hat) as

$$\bar{h} = \frac{h}{\hat{h}}\,, \quad \bar{x} = \frac{x}{\hat{x}}\,, \quad \bar{y} = \frac{y}{\hat{y}}\,, \quad \bar{\eta} = \frac{\eta}{\hat{\eta}}\,, \quad \bar{p} = \frac{p}{\hat{p}} = p\frac{\pi d \hat{y}}{\hat{F}}\,.$$

All characteristic quantities are chosen in such a way, that the dimensional quantities are in the order of one. In further work it is planned to define the characteristic quantities using “real” reference values depending on the particular operating conditions, e.g. the viscosity at oil sump temperature, surface roughness parameters as characteristic lengths, etc. However, currently constant characteristic quantities are chosen for the development of the simulation model and will be described below.

Since the exact contact width is a priori unknown, the characteristic length in axial direction is chosen as

$$\hat{y} = 0.1\, mm \ \approx b\,.$$

For numerical reasons, a squared computational domain is preferred. Therefore the characteristic length in circumferential direction is chosen equal to the characteristic length in axial direction,

$$\hat{x} = 0.1\, mm\ (= \hat{y})\,.$$

The height of the fluid film is in the order of the surface roughnesses of the shaft and the sealing lip. Since the exact film height is a priori unknown, the characteristic quantity is chosen as

$$\hat{h} = 1\, \mu m\,.$$

As reference viscosity $\hat{\eta}$ the dynamic viscosity η at 40°C is chosen which can easily be calculated using the kinematic viscosity ν at 40°C and the density ρ which both should be listed in the data sheet of every oil,

$$\hat{\eta} = \eta(\vartheta = 40°C) = \nu(\vartheta = 40°C) \cdot \rho\,.$$

Several ratios can be defined using the characteristic quantities. The ratio K_y represents the aspect ratio of the computational domain and equals often one,

$$K_y = \frac{\hat{y}}{\hat{x}}\,.$$

The ratio K_d represents the ratio of the shaft diameter d to the characteristic length $\hat{x}$ in circumferential direction,

$$K_d = \frac{d}{\hat{x}}\,.$$

The ratio K_h represents the ratio of the characteristic film thickness $\hat{h}$ to the characteristically length $\hat{x}$ in circumferential direction,

$$K_h = \frac{\hat{h}}{\hat{x}}\,.$$

Using the characteristic quantities and the ratios defined above, the governing equations can be transformed to dimensionless equations, to reduce the number of input parameters [8].

2.3 Dimensionless Reynolds equation

The dimensionless Reynolds equation is

$$\frac{\hat{\rho}\hat{h}^3\hat{p}}{\hat{x}^2\hat{\eta}} \cdot \frac{\partial}{\partial \bar{x}}\left(\frac{\bar{\rho}\bar{h}^3}{6\bar{\eta}}\frac{\partial \bar{p}}{\partial \bar{x}}\right) + \frac{\hat{\rho}\hat{h}^3\hat{p}}{\hat{y}^2\hat{\eta}} \cdot \frac{\partial}{\partial \bar{y}}\left(\frac{\bar{\rho}\bar{h}^3}{6\bar{\eta}}\frac{\partial \bar{p}}{\partial \bar{y}}\right) - u_s\frac{\hat{\rho}\hat{h}}{\hat{x}} \cdot \frac{\partial(\bar{\rho}\bar{h})}{\partial \bar{x}} = 0\,.$$

The characteristic quantities and input parameters (such as radial force and rotational speed) can be summarized assuming an incompressible fluid and using $u_s = \omega \cdot d/2$,

$$\frac{1}{3} \cdot \underbrace{\frac{\hat{h}^2}{\hat{x}^2}}_{=K_h^2} \cdot \underbrace{\frac{\hat{x}}{d}}_{=\frac{1}{K_d}} \cdot \underbrace{\frac{\mathrm{F_r}}{\omega\hat{\eta} \cdot \pi d\hat{y}}}_{=\frac{1}{G}} \cdot \frac{\partial}{\partial \bar{x}}\left(\frac{\bar{\rho}\bar{h}^3}{\bar{\eta}}\frac{\partial \bar{p}}{\partial \bar{x}}\right) +$$

$$\underbrace{\frac{\hat{x}^2}{\hat{y}^2}}_{=\frac{1}{K_y^2}} \cdot \frac{1}{3} \cdot \underbrace{\frac{\hat{h}^2}{\hat{x}^2}}_{=K_h^2} \cdot \underbrace{\frac{\hat{x}}{d}}_{=\frac{1}{K_d}} \cdot \underbrace{\frac{\mathrm{F_r}}{\omega\hat{\eta} \cdot \pi d\hat{y}}}_{=\frac{1}{G}} \cdot \frac{\partial}{\partial \bar{y}}\left(\frac{\bar{\rho}\bar{h}^3}{\bar{\eta}}\frac{\partial \bar{p}}{\partial \bar{y}}\right) - \frac{\partial(\bar{\rho}\bar{h})}{\partial \bar{x}} = 0\,.$$

Using the dimensionless ratios K_y and K_h defined above, the so called Guembel number G appears in the dimensionless Reynolds equation,

$$G = \frac{\eta\hat{\omega}}{\frac{F_r}{\pi d\hat{y}}}\,.$$

The Guembel number G is a dimensionless number used to characterize the lubrication and friction conditions in the contact area of radial lip seals [6].

The dimensionless ratios and the original Guembel number G can be summarized to a modified Guembel number G^* which is also dimensionless,

$$G^* = 3 \cdot \frac{K_d}{K_h^2} \cdot G\,.$$

This summary simplifies the dimensionless Reynolds equation to its final form, which is discretized for the numerical solution in TEHDyBAER,

$$\frac{1}{G^*} \cdot \frac{\partial}{\partial \bar{x}} \left(\frac{\bar{\rho}\bar{h}^3}{\bar{\eta}} \frac{\partial \bar{p}}{\partial \bar{x}} \right) + \frac{1}{K_{\bar{y}}^2} \cdot \frac{1}{G^*} \cdot \frac{\partial}{\partial \bar{y}} \left(\frac{\bar{\rho}\bar{h}^3}{\bar{\eta}} \frac{\partial \bar{p}}{\partial \bar{y}} \right) - \frac{\partial \left(\bar{\rho}\bar{h} \right)}{\partial \bar{x}} = 0\,.$$

The dimensionless Guembel equation shows, that the Guembel number (in the original as well as in the modified definition) is the relevant parameter for the fluid flow in the sealing gap. The Guembel number can be interpreted as a measure for the ability of the sealing system to build up hydrodynamic pressure.

2.4 Dimensionless elasticity equation

Just like the Reynolds equation, the elasticity equation can be nondimensionalized,

$$\bar{h}(\bar{x},\bar{y}) = \bar{h}_0 + \bar{h}_1(\bar{x},\bar{y}) + \bar{h}_2(\bar{x},\bar{y}) + \frac{2}{\pi} \cdot \frac{F_r}{E'\hat{x}\hat{h}} \int_{\bar{y}_a}^{\bar{y}_b} \int_{\bar{x}_a}^{\bar{x}_b} \frac{\bar{p}(\bar{x}',\bar{y}')d\bar{x}'d\bar{y}'}{\sqrt{(\bar{x}-\bar{x}')^2 + K_{\bar{y}}^2(\bar{y}-\bar{y}')^2}}\,.$$

Summarizing the characteristic quantities and input parameters as for the Reynolds equation, one obtains

$$\bar{h}(\bar{x},\bar{y}) = \bar{h}_0 + \bar{h}_1(\bar{x},\bar{y}) + \bar{h}_2(\bar{x},\bar{y}) + \underbrace{\frac{2}{\pi} \cdot \frac{K_y}{K_h} \cdot \frac{\frac{F_r}{\pi d\hat{y}}}{E'}}_{=\frac{1}{S}} \int_{\bar{y}_a}^{\bar{y}_b} \int_{\bar{x}_a}^{\bar{x}_b} \frac{\bar{p}(\bar{x}',\bar{y}')d\bar{x}'d\bar{y}'}{\sqrt{(\bar{x}-\bar{x}')^2 + K_{\bar{y}}^2(\bar{y}-\bar{y}')^2}}\,.$$

The term in front of the deformation integral is summarized to a dimensionless number S, called stiffness number, representing the mechanical gap stiffness,

$$S = \frac{\pi}{2} \cdot \frac{K_h}{K_y} \cdot \frac{E'}{\frac{F_r}{\pi d\hat{y}}}\,.$$

The reciprocal of S represents the opening tendency of the sealing gap formed by the sealing lip and the shaft surface. For high values of S, the sealing gap increases its height only slightly with increasing hydrodynamic pressure. For low values of S, the sealing gap increases its height easily with increasing hydrodynamic pressung.

2.5 Dimensionless force balance equation

The force balance equation can be also simplified to a dimensionless form which does not contain any dimensionless numbers but only dimensionless variables,

$$\int_{\bar{y}_a}^{\bar{y}_b} \int_{\bar{x}_a}^{\bar{x}_b} \bar{p}(\bar{x},\bar{y})\, d\bar{x}\, d\bar{y} = (\bar{x}_b - \bar{x}_a)\,.$$

Since the parameters K_v and m, used in the Ubblohde-Walther equation, are already dimensionless, this equation needs not to be nondimensionalized.

3 Simulation results

Since the simulation model is in an early development stage, only preliminary results are presented showing the potential of the simulation model.

The simulated results, such as pressure distribution and local gap height, are exported from THEDyBAER into ASCII text files. Based on this text files, several kinds of plots can be created using gnuplot.

Figure 6 shows an example of a pressure distribution simulated for synthetically generated surfaces containing a sinusoidal roughness. As expected, each asperity generates a pressure spike.

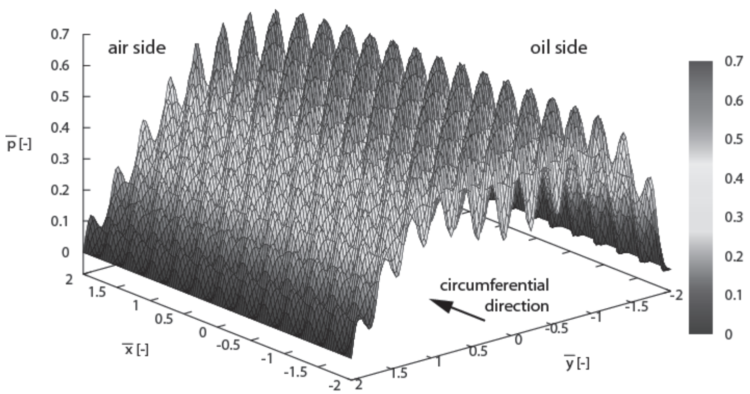

Figure 6: Pressure distribution in the contact area

4 Conclusion

A multi scale simulation model is developed at the Institute of Machine Components (IMA) at the University of Stuttgart for predicting the temperature in radial lip seals and their environment. This multi scale model consists of two models on different scales which are coupled. The macro model covers the whole sealing system as well as its environment and simulates the heat dissipation by means of a conjugate heat transfer simulation. The micro model, that is called TEHDyBAER, solely covers the sealing gap and simulates the heat generation by means of a thermo-elasto-hydrodynamic simulation.

The governing equations of the micro model are

- the Reynolds equation, describing the fluid flow in the sealing gap,
- the elasticity equation describing the deformation and the film thickness,
- the Ubbelohde Walther equation, describing the viscosity temperature relation of the fluid and
- the force balance equation, describing the equilibrium of forces in radial direction.

Nondimensionalizing these equations, dimensionless numbers can be derived which characterize the behavior of the sealing system. These parameters are:

- the modified Guembel number G^*, characterizing the ability of the sealing system to build up hydrodynamic pressure,
- the stiffness number S, characterizing the mechanical stiffness of the gap, and
- the fluid parameter m, representing the slope in the Ubbelohde-Walther chart and thus characterizing the change of fluid viscosity with variations in temperature.

The dimensionless equations are used in the micro model which is based on a multi grid approach and delivers first results.

In the further work the micro model will be expanded to carry out parameter studies analyzing the influence of these parameters.

5 Acknowledgement

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