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A MIXED THERMOELASTOHYRODYNAMIC LUBRICATION ANALYSIS OF MECHANICAL FACE SEALS BY A MULTISCALE APPROACH

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Abstract

This paper presents an improvement of a previous multiscale approach used to model the mixed lubrication in a mechanical face seal. The physical mechanisms considered by the improved model are surface roughness effects on the fluid film lubrication, thermal deformations and heat transfer in the seal ring due to the viscous and dry frictions. The developed numerical model determines the pressure distribution by taking into account the effect of cavitation and contact asperity between the surfaces. Heat dissipation, heat transfer and deformations are computed from the heat dissipated at the seal interface by a finite element technique. The multiscale model significantly reduces computation time whilst maintaining the accuracy of the results. Results obtained through a parametric study show that there are different operating zones where the lubricating film thickness is controlled by the roughness height or so by the thermal effect.

Introduction

For over twenty years, the problem of heat transfer in mechanical seals has been the subject of several research works at the University of Poitiers. An initial study by Tournerie et al [1] evaluated the thermal effects at the face seal interface. A measurement technique based on the infrared thermography was used to determine the temperature field at the interface. The heat transfer between the mechanical seal and the surrounding environment can be achieved by convection with the fluid or air, by conduction with the rings, the shaft or the housing. Lebeck [2] summarized the various transfer mechanisms around the seal. In 1994, Knoll et al [3] developed a numerical model based on a finite element program, which allows calculating thermal and mechanical characteristic (temperature distribution, thermal deformation etc.). In the work of Person et al [4], the thermohydrodynamic effects in mechanical seals are modeled. A 3D numerical model with a floating stator and misaligned faces is developed. The different equations used are discretized by the finite difference method.

In all of the studies presented thereby, the thermal problem has been solved by considering that the surfaces are perfectly smooth or grooved. However, the analysis of thermal effects in the mechanical seals remains essential to predict its performance such as leakage rate, which requires an accurate estimation of the film thickness between the faces and thus their deformations.

In the present paper, a previous multiscale approach [5] for mechanical seals operating in the mixed lubrication regime is improved by including the thermal distortions and heat transfer in the seal rings due to the viscous and dry frictions. This study is different, as it integrates the surface roughness, which is numerically generated. The developed model is focused towards a comparison between three different models: isothermal hydrodynamic (HD), non isothermal thermo-hydrodynamic (THD) and themo-elasto-hydrodynamic, which takes into account the thermoelastic deformation (TEHD).

1. Theoretical Considerations

The theoretical multiscale model has been developed and described in a previous study [5]. The principle of the model is to express pressure on a macro-mesh by using a mass-conservative law, whose coefficients are computed on a micro-scale mesh.

1.1 Energy dissipated in the film

The problem of the energy dissipation is solved by assuming that the flux dissipated by viscous and dry frictions at the interface is entirely transmitted by conduction to the rings. In addition, the temperature is supposed to constant across the film thickness, that implies T(r) =T_1(r) =T_2(r). This can be resumed by the following equation:

$$\frac{Cf_{macr\_el}}{S_{macr\_el}}+\left[+K_1\frac{\partial T(r)}{\partial z}\bigg|_{H_1^+}-K_2\frac{\partial T(r)}{\partial z}\bigg|_{H_2^-}\right]=0 \qquad (1.\ 1)$$

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1.2 Heat equation in the rings

The rings are made of two different materials (silicon carbide for the stator and carbon for the rotor, in the following examples) However, both rings share the same interface, where heat is generated. The temperature fields in the two rings are determined simultaneously by solving the heat equation for both solids. For an axisymmetric and steady configuration, the equation is:

$$K_i\left(\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial r^2}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}\right)=0 \qquad (1.2)$$

1.3 Thermal deformation

The displacement fields in the solids are obtained by solving the Lame-Navier equation. It is expressed as:

$$\frac{1}{2(1+\nu)}\overrightarrow{grad}\,div\,\vec{u}+\frac{1-2\nu}{2(1+\nu)}\Delta\vec{u}-\lambda\,\overrightarrow{grad}\,T=0 \quad (1.3)$$

where ν, λ are respectively the Poisson's ratio and the coefficient of linear expansion of the material.

1.4 Boundary conditions

The thermal boundary conditions are shown in figure 1. There is no exchange around the rings, except on the outer side that is wetted by the sealed fluid.

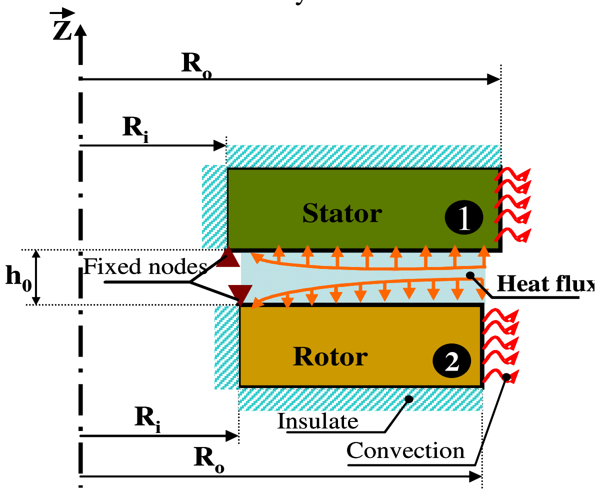

Figure 1: Thermal and mechanical boundary conditions

The outer surface is cooled by convection with the sealed fluid. The convection coefficient is calculated using the Becker's empirical formula:

$$h_c=0.133\frac{k_f}{2R_e}\left(\frac{2\rho\omega R_e}{\mu}\right)^{2/3}\left(\frac{C_p\,\mu}{k_f}\right)^{1/3} \qquad (1.4)$$

For calculating the deformations, the necessary condition is to block the first nodes on the inner boundary along the z direction.

1.5 Numerical resolution

The heat flux, the temperature, the faces distortion and the forces are determined in each macro-cell. The friction coefficient is evaluated in the micro-scale model, where the micro-pressure and micro-flow rate are computed. Since the fiction coefficient is known, the heat flux can be computed in the macro-cells. This is followed by the determination of the temperature, which is obtained from the energy dissipated at the seal interface. After that, the viscosity is updated. The thermal deformation in each macro-cell is then calculated with equation 1.3. The different equations presented above are solved by using the finite element technique. It allows an accurate consideration of the form of solids with the associated boundary conditions. When the temperature, deformations and viscosity are calculated, the multi-scale model is used to determine new pressure, film thickness and friction distributions. The entire process is repeated until convergence is reached.

2. Results

The study has been performed with the mechanical seal described in table 1.

Table 1: Operating and design parameters of the mechanical seal.

Inner radius $R_{i\,(Stator)}$	0.029 m
Outer radius $R_{o\,(Stator)}$	0.033 m
Inner radius $R_{i\,(Rotor)}$	0.0028 m
Outer radius $R_{o(Rotor)}$	0.0034 m
Height of rotor and stator	0.005 m
Balance ratio	0.75
Rotation speed ω	10 – 900 rad/s
Outer pressure p_o	1 Mpa
Inner pressure	0
Fluid viscosity μ	10^{-3} Pa.s
Fluid density ρ_0	1000 kg/m^3
Cavitation pressure p_{cav}	-0.01 MPa
Dry Contact friction coeff.	0.2

The surfaces used in this study are numerically generated. The characteristics are the same as the ones used in the previous paper [5]. The efficiency of the multiscale approach to solve de thermal problem in mechanical seals is performed by comparing the following HD, THD and TEHD models.
The results are presented as a function of the duty parameter G, which characterises the magnitude of the hydrodynamic effect in the fluid film:

$$G=\frac{\mu\,\omega\left(R^2_{o(St)}-R^2_{i\,(St)}\right)}{2\,F_{clos}} \qquad (1.5)$$

Figure 2 shows the development of the hydrodynamic lift and contact forces as a function of the G parameter for different types of behavior.

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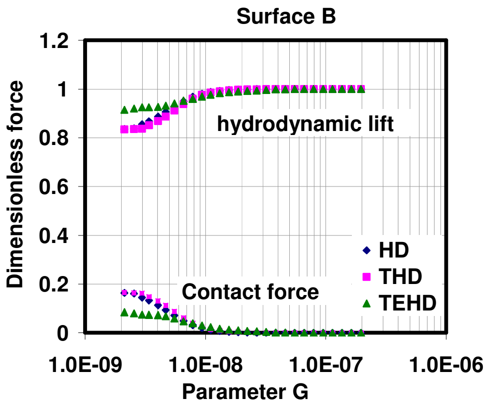

Figure 2: Comparison of contact force and hydrodynamic lift

The result of the TEHD model differs from others in the area of mixed lubrication, where there is an increase of the hydrodynamic lift. This is partly due to the deformation of the faces leading to an evolution of the conicity that enhances the hydrostatic lift.

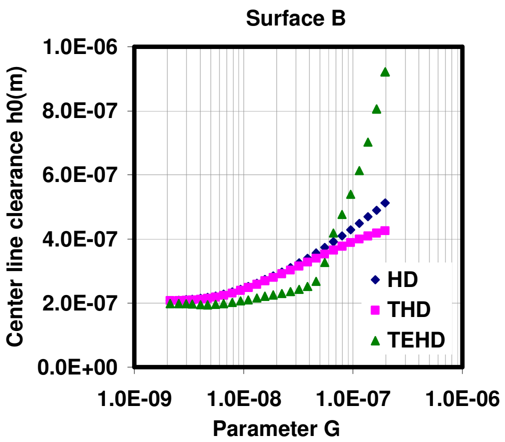

Figure 3: Film thickness variation

The distance between the seal faces is evaluated after faces equilibrium is reached. It can vary from one type of model to another. Curves obtained with the HD model and the THD model in figure 3 are almost superimposed and the film thickness increases with the parameter G up to a certain value (G $>10^{-8}$). After this value, the film thickness of the THD model begins to deviate from that obtained with the HD model. This behavior is due to the increase of temperature (hence the lower viscosity), leading to a smaller hydrodynamic force and thus a lower film thickness.
For the TEHD model, the behavior is very different. The film thickness increases progressively with the G parameter, and after a certain value, it increases abruptly. This result is due to the thermal deformations which tend to increase the faces coning angle, resulting in ae high value of the film thickness at equilibrium.
Figure 4 shows the evolution of the friction coefficient as a function of the G parameter. There is a superposition of curves in the area of mixed lubrication of HD and THD models. The difference between them appears in the area of hydrodynamic lubrication. The gap observed is due to an increased temperature in the film (model THD), decreasing thereby the viscous shear and consequently the coefficient of friction.
The TEHD model differs significantly from other models with a rather peculiar evolution. Indeed, because of the thermal gradients and the associated deformations contributing to the increase in faces conicity, the seal behaves as a hydrostatic seal whose film thickness is thermally controlled. This behavior increases the film thickness and leakage, while decreasing the friction.

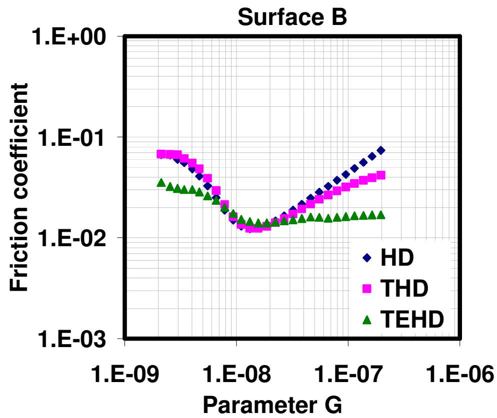

Figure 4: Stribeck's curves variation

3. Conclusion

The important and innovative part of this work is the consideration of the THD and TEHD models in the study of mixed lubrification, when one surface is rough. The analysis of these models was made by a comparative study (HD, THD and TEHD models). It allowed examining the role of thermal effects and the deformations in the mechanical seal.
It has been possible to identify a transition between a roughness controlled film thickness, at low duty parameter values, and a thermally controlled film thickness for higher G values.

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