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Experimental and numerical study of the lubrication regimes of a liquid mechanical seal

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ABSTRACT

The paper presents an experimental and numerical study of the different lubrication regimes occurring in the sealing gap of a mechanical seal with water as sealed fluid. For this purpose, an industrial seal has been tested in a wide range of rotation speeds and fluid pressure values. Friction torque, temperature and leakage were measured and compared to numerical results. These results were obtained on a multi-scale mixed lubrication model considering heat transfer in the solids and seal rings deflections. A satisfactory correlation was obtained demonstrating the relevance of the model. The model is then used to analyze the lubrication regimes.

1. Introduction

Mechanical face seals are dynamic sealing devices used in almost every sector of industrial pumping applications: the aeronautical industries, drive mechanisms for vehicles, drives in conveyers, mixers, machinery and equipment for chemical, food processing and mining. These seals are composed of a rotating ring pressed on a static ring and used to separate a pressurized process fluid from the atmosphere. Good working conditions are reached when a thin lubricating liquid film (a fraction of a micrometer) separates the two faces, providing a high wear resistance as well as a low leakage rate. The identification of the lubrication regime is very important for the design of mechanical face seals because it controls the durability of the seal and must prevent the seal from overheating and excessive wear.

Denny [1] was among the first to show the existence of a fluid film between the sealing parallel surfaces counter to classical lubrication theory. Indeed, it is theoretically impossible to generate a hydrodynamic pressure when two smooth parallel surfaces are sliding in the presence of a liquid. His experiments were carried out with a plane parallel thrust surface with an applied pressure differential across the face as in radial face seals. He showed by using a capacitive method that the film thickness varies between 0.5 and 2.5 μm depending on speed and load.

The same year, Summers-Smith [2] presented measurements of friction torque, leakage and wear on mechanical seals made of different materials and operating with different sealed fluids. He obtained typical Stribeck curves showing the existence of two regimes of lubrication: mixed lubrication at low duty parameter G values and full fluid film for higher values. The transition between these two regimes was observed at a value of about $G=5\cdot 10^{-8}$. Beyond this threshold, wear rates are acceptably low, confirming that the mechanical face seal operates in a non-contact regime typical of a hydrodynamic lubrication.

In 1967, Nau [3] reviewed in detail the literature of fundamental and experimental studies focused on mechanical face seals operating in a hydrodynamic regime. The seal waviness is, according to the author, the parameter which promotes the hydrodynamic pressure generation. He demonstrated that the friction coefficient f is a power function of G in the full film regime with an exponent equal to 0.5.

However, the experiments and theoretical analysis of Pape [4] lead to a different relation where the exponent could vary between 2/3 and 1. According to Pape, the macroroughness of the sealing faces is the prime factor causing hydrodynamic lubrication. He also highlighted the effect of the faces' radial taper in fluid film generation.

In two papers, Lebeck [5,6] analyzed the mechanisms of load generation between two parallel surfaces. It could be due to several combined effects such as roughness, waviness, and thermal wedge.

In 1992, Flitney and Nau [7] tried to obtain Stribeck curves for different seal types and different operating conditions. They

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Nomenclature

B_h	Balance ratio
C_f	Friction torque (N m)
E,F	Elliptic integrals
E'	Equivalent elasticity modulus (Pa)
fs	Dry friction coefficient
f	Friction coefficient
F_c	Contact force (N)
F_s	Spring force (N)
F_{cl}	Closing force (N)
G	Duty parameter $G=\mu\omega\left(R_0^2-R_i^2\right)/2F_{cl}$
h_r	Height of the rough surface (m)
k_f	Thermal conductivity coefficient (W/m K)
k	Ellipticity ratio
Ku	Kurtosis
L	Lubrication number $L=G\frac{S_f}{(R_e-R_i)R_t}$
Lr	Rotor length (m)
Ls	Stator length (m)
l	Centerline clearance (m)
$\dot{m}_i$	Mass flow rate at macro-cell (kg/s)
p_i	Pressure at the macro-cell (Pa)
P_i	Atmosphere pressure (Pa)
P_0	Sealed fluid pressure (Pa)
r_{eq}	Equivalent curvature radius (m)
R_o	Outer seal radius (m)
R_i	Inner seal radius (m)
R_{os}	Outer stator radius (m)
R_{is}	Inner stator radius (m)
R_{or}	Outer rotor radius (m)
R_{ir}	Inner rotor radius (m)
Sk	Skewness
Sq	Roughness height (m)
T	Temperature rise (°C)
T_f	Fluid temperature (°C)
W_h	Fluid film force (N)
W_c	Total contact force (N)
β_f	Thermal viscosity coefficient (K^{-1})
δ	Interference of the contact (m)
ΔT	Temperature rise (K)
λ	Thermal expansion coefficient (K^{-1})
μ_0	Fluid dynamic viscosity (Pa s)
ν	Poisson ratio
ω	Rotational speed (rad/s)

indicated that there is an important scatter on friction, whereas the faces' temperature or seal wear are a more convenient way to identify the transition from mixed to hydrodynamic lubrication regimes. Gu [8] demonstrated that the duty parameter is not sufficient to determine the friction states of a mechanical face seal. He proposed other parameters such as the fluid film load carrying ratio and the specific film thickness. In 1992, Schipper et al. [9] introduced a lubrication number which is better at determining the mode of lubrication of a seal. This number is an improved duty parameter also including the surface roughness. Vezajk and Vizintin [10] confirmed the role of the lubrication number in characterizing the lubrication regime transition. For that, they performed short interval P–V tests with different face material combinations. They found that the transition occurs at $L=10^{-2}$ whatever the material. Based on his experimental and numerical work, Lubbinge [11] stated that the transition from mixed to hydrodynamic lubrication regimes is influenced by the number and amplitude of the waves on the seal ring, the taper angle and the surface roughness, but also by the viscosity of the fluid, the level of loading and the elasticity modulus of materials.

By using a mathematical generation of surface roughness and a deterministic mixed lubrication model, Minet et al. [12] theoretically demonstrated the role of surface roughness in hydrodynamic pressure generation. In a second paper [13], they showed the role of several parameters on the friction regime. Surface taper appeared to be an important parameter affecting the transition to the hydrodynamic regime.

The seal faces generally become tapered because of thermal gradients in the seal rings resulting from interfacial friction [14]. Thus, to propose a relevant seal model, Ruan et al. [15] considered mixed lubrication, heat transfer and seal rings deflection due to mechanical and thermal loadings. However, the stochastic lubrication models proposed by Ruan et al. cannot reproduce the roughness generated pressure. Lebeck [16] solved this problem by using an empirical law based on experimental results.

To offer a more realistic model, Nyemeck et al. developed a multiscale deterministic mixed lubrication model [17] that they coupled with a heat transfer and deformation model for the seal rings [18]. They were able to simulate the transition from a mixed to a hydrodynamic lubrication model and highlighted the role of thermal effects. However, they neglected the mechanical deformation of the rings, which could be significant in real situations.

The objective of the present paper is to study a mechanical seal working in different lubrication regimes.

Several experiments were performed while varying the rotation speed and the pressure of the sealed fluid to measure the friction torque, thermal gradients and leakage of the mechanical seal. The numerical model of Nyemeck et al. [18] is improved to consider the real shape of the rings and mechanical deformation. The numerical and experimental results are compared. The transition from mixed to full film regime is discussed.

2. Experimental apparatus

2.1. Test rig

The test rig is composed of a horizontal precision spindle with a special test cell enclosing the seals to be tested (Fig. 1). The spindle is driven by a synchronous motor by means of a flexible safety shaft. It allows filtering of torsional vibrations coming from the motor and protecting the system in case of high torque.

A cross-sectional view of the test rig is shown in Fig. 2. A first double hydrostatic conical bearing is located between the shaft and the frame, which ensures an accurate rotating motion of the shaft, and another double hydrostatic conical bearing is located between the stator of the experimental cell and the frame, which gives a rotational degree of freedom to the cell.

The experimental cell is designed to receive simultaneously two industrial seals to eliminate the axial load due to the sealant pressure. The test cell is fed in water by a pressurized hydraulic loop.

2.2. Measurements

A force sensor (Fig. 1) is used to measure the torque applied to the stator of the experimental cell. However, it is necessary to remove the torque due to the feeding flexible pipes and due to churning around the seal from the measurement to obtain the friction torque of the two seals.