Validation of a 1-D Elastohydrodynamic lubrication model
Validation d'un modèle 1-D de lubrification élastohydrodynamique

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Abstract

The sealing performance of a hydraulic cylinder depends on the characteristics of three essential elements: the rod, the seal and the fluid. To predict the behavior of a hydraulic seal, including the friction force and leakage rate, a series of theoretical and experimental studies have been carried out. In this article, a one-dimensional elastohydrodynamic model of the U-cup hydraulic rod seal is developed taking into account: the roughness of the shaft and lips. The numerical results are validated by experiments previously published.

Résumé

Les performances d'étanchéité d'un vérin hydraulique dépendent des caractéristiques de trois éléments essentiels : la tige, le joint et le fluide. Pour prédire le comportement d'un joint hydraulique, à savoir la force de frottement et le taux de fuite, plusieurs travaux théoriques et expérimentaux ont été réalisés. Dans le présent article, un modèle unidimensionnel élastohydrodynamique du joint hydraulique en U est réalisé en prenant en compte: la rugosité de l'arbre et des lèvres. Les résultats numériques sont validés par des expérimentations publiées précédemment.

Mots clefs : Joint hydraulique ; Surface texturée ; Rugosité ; Force de frottement ; Fuite.

Keywords: Rod seal, Surface-textured, Roughness, Friction force, Leakage.

1. Introduction

The hydraulic U-rod seal is the most frequently used machine component to prevent leakages with minimal wear effect. Since the 1960s, great importance has been assigned to the modeling of hydraulic joints. The behavior complexity of this device is due to several physical phenomena that could interfere. Indeed, the thermal effect of friction affects simultaneously the elasticity of the elastomeric seal and the rheological law of the lubricant film.

Previously, Lawrie and O'Donoghue [1] have proved experimentally the presence of a thin film throughout the lubricated contact. They also demonstrated that in the outstroke rod motion, the film is thicker and the shape of the lip has a significant effect on the U-cup hydraulic rod seal performances.
Earlier, Vissher and Kanters [2] have investigated the rod roughness effect on the performance of rectangular polyurethane seals. Thus, above a critical arithmetic value of roughness, the seal may leak. However, by using a grooved shaft, the friction force decreases substantially as described by Elgadari et al [3]. Therefore, a judicious pattern could improve the lifetime of such a device.
To model the film thickness behavior, several approaches have been used. In this article, only two numerical methods have been reported:

- Inverse hydrodynamic lubrication (IHL): Based on the assumption that the hydrodynamic pressure is equal to the static pressure calculated on structural computational software (FEM), the Reynolds equation is solved by taking the film thickness as the unknown parameter, as carefully detailed in Crudu thesis [4]. It was underlined that the numerical results are slightly agreed with the measurements.
- Elastohydrodynamic lubrication (EHL): Elgadari et al [3] have recently used this approach by solving the Reynolds equation and taking into account the elasticity of the seal, and the roughness of the lip and the shaft. It was proved that asymmetric grooves of the shaft, can improve significantly the sealing performances.

The objective of this work is to perform a parametric study by considering a one-dimensional elasto-hydrodynamic model that takes into account the elasticity of the lip and the roughness of the shaft.
After validating the current model, numerical simulations were performed and compared with experimental results.

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2. Theoretical approach

2.1 Assumptions

Fig. 1 illustrates a hydraulic rod seal, and Fig. 2 represents the model of the sealing zone, by assuming:

- The seal operates at a steady velocity in both directions of motion: Instroke and Outstroke cases,
- The dynamic viscosity of the lubricant is only dependent on the oil pressure by the piezo-viscosity property given by the equation:

$\mu = \mu_0.\exp(\alpha p_s)$ where $\alpha=[34.95+9.65\log_{10}(\mu_0)]10^{-9}$GPa

- The lubricant side of the seal is submerged entirely in the lubricant with a sealed pressure p_s.
- The average film thickness is uniform in the axial direction, according to previous numerical and experimental results Crudu [4].
- The dry contact is not considered in this theoretical model.
- The circumferential shear deformation is considered.

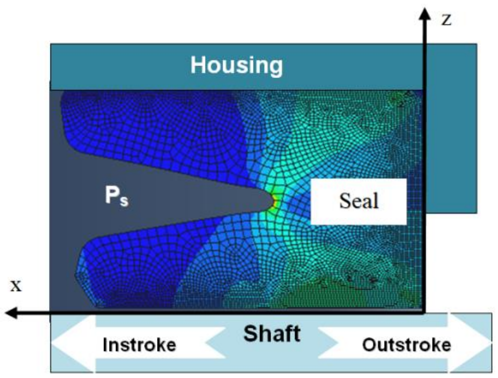

Fig.1: U-Cup reciprocating rod seal

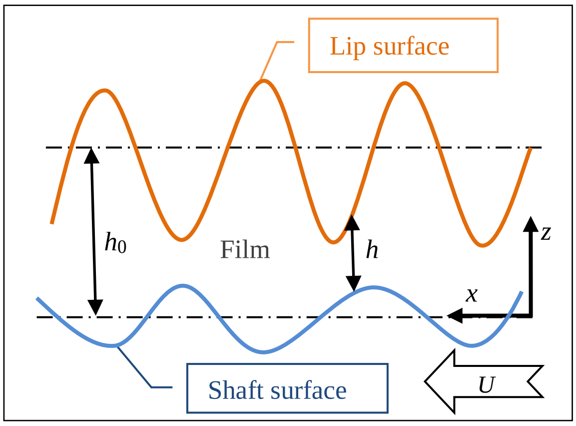

Fig.2: Schematic diagram of the sealing zone

2.2 Governing equations

In order to take into account the cavitation effect, the modified Reynolds equation is adopted:

$$\left\{\frac{\partial}{\partial x}\left(h^3\frac{\partial D}{\partial x}\right)\right\}F = 6\mu U\frac{\partial h}{\partial x} + 12\mu\frac{\partial h}{\partial t} + 6\mu(1-F)\left\{U\frac{\partial D}{\partial x} + 2\frac{\partial D}{\partial t}\right\} \quad (1)$$

Where:

$D = p$ and $F = 1$, when $D > 0$

$D = r - h,\ r = \frac{\rho}{\rho_0}h$, and $F = 0$ when $D \leq 0$

ρ and ρ_0 are respectively the densities of the lubricant-gas mixture and the lubricant.

D at $x = 0$ is equal to p_{air} and at $x = L$, D is equal to p_s.

A preliminary structural analysis of commercial software was carried out to determine:

- Contact pressure field p_c and contact width L due to the mounting of the seal on the rod.
- The radial and tangential compliance matrices C_z and C_x respectively, based on the study by Elgadari et al [3]. So, the radial and tangential displacements are given by:

$$(\delta_z) = \sum_{j=1}^{N_x}(C_z)_{i,j}(p_j - p_{cj}) \quad (2)$$

$$(\delta_x) = \sum_{j=1}^{N_x}(C_x)_{i,j}\tau_{xzj} \quad (3)$$

Where p_j is the nodal film pressure, p_{cj} is the nodal contact static pressure, N_x is the number of nodes, and τ_{xzj} is the nodal shear stress calculated with:

$$\tau_{xz} = F\left[\frac{1}{2}\frac{\partial p}{\partial x}h + \mu\frac{U}{h}\right] - (1-F)\mu\frac{rU}{h^2} \quad (4)$$

2. 3 Film thickness

Figure 2 shows that the thickness of the thin film h is written as follows:

$$h(x,t) = h_2(x-\delta_x) - h_1(x-U.t) + h_0 + \delta_z(x,t) \quad (5)$$

With h_1 the rod roughness, h_2 the lip roughness, h_0 the average film thickness, δ_x and δ_z the axial and normal lip displacement respectively by equations (2) and (3).

To validate the current model, the work of Crudu [4] and Elgadari [3] is reproduced by simulating 4 cases of sealed pressure: 4.5 MPa, 9.5 MPa, 12.5 MPa, and 19.5 MPa and taking into account surface roughness with simple analytical functions similar to the one previously published.

The lip roughness is assumed sinusoidal and given by:

$$h_2(x-\delta_x) = A_2\sin\left(\frac{2\pi}{\lambda_2}(x-\delta_x)\right) \quad (6)$$

And the rod roughness is given by:

$$h_1(x-Ut) = A_1\sin\left(\frac{2\pi}{\lambda_1}(x-Ut)\right) \quad (7)$$

With λ_1 and λ_2 are the axial periodicities of the rod and seal roughness respectively.

3. Validation

In order to compare the friction force F_f with the experimental results achieved by Crudu [4], this parameter is calculated by :

$$F_f = 2\pi R \int_0^L \tau_{xz}.dx \qquad (8)$$

With R is the radius of the rod and L is the width of contact.

Table 1 summarizes the different functional parameters considered in the numerical simulations.

Parameter	Numerical Value
Dynamic viscosity	$\mu = 1.26\ 10^{-7}$MPa.s
Lubricant density (Eq.[4])	$\rho = 974$Kg/m^3
Rod velocity	U=+/- 80mm/s
Lip asperity number	N_2=10
Lip wavelength roughness	$\lambda_2 = L/N_2$
Lip roughness fluctuation	A_2=1,54 micron
Rod asperity number	N_1=10
Rod wavelength roughness	$\lambda_1 = L/N_1$
Rod roughness fluctuation	$A_1 = A_2/10$

Tab.1: Parameters adopted for the parametric study

Figure 3 shows a significant correlation between the current model and the experiment.

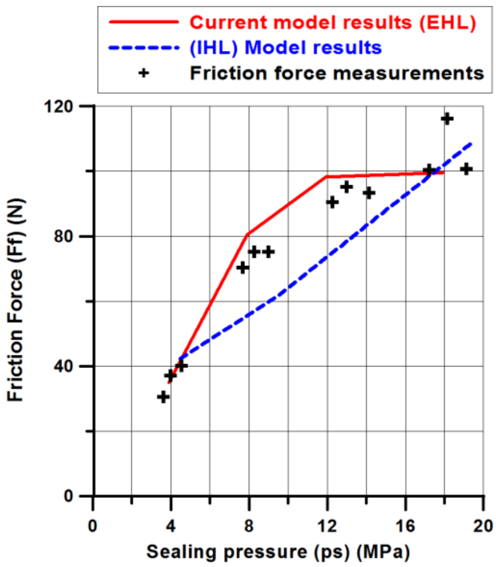

Figure 3. Friction force versus different sealed pressure

4. Conclusions and perspectives

In the present study, we have developed an elasto-hydrodynamic model (EHL). A structural analysis was performed to determine the width and pressure of contact. These were used as input parameters for resolving the modified Reynolds equation. A comparison between the friction force obtained by the simulation and the other experimental results confirms the validation of the actual model. This work has introduced new ways to explore, by changing the surface states and characteristics of the lubricant.

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