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# Shaft roughness effect on elasto-hydrodynamic lubrication of rotary lip seals: Experimentation and numerical simulation

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ABSTRACT

Numerical analyses of the isothermal elasto-hydrodynamic lubrication (EHL) have made considerable advances in order to identify the most important features in the successful operation of rotary lip seal, and the results have shown a good agreement with experiments.

Most of the models previously published are capable of predicting the combined effects of thin film through deformed lip and rotating shaft, but they assume a smooth surface of the shaft. Although this assumption is only verified for shaft roughness much smaller than that of the seal lip, it is the best solution to avoid a transient model.

First, the present study describes an experimental work that provides a basis upon which a numerical EHL model of rotary lip seal is constructed by taking into account both the shaft and lip roughness. After confirming the validity of the current model by comparing experimental with numerical results, simulations have been performed and have underlined the effect of shaft roughness amplitude and profile on the rotary lip seal performance. It is shown that for shaft roughness beyond half of the lip roughness, the seal may leak.



## 1. Introduction

The rotary lip seal is the most common type of rotary shaft seals (Fig. 1). It is used to withstand differences in pressure, to contain lubricant and to exclude contaminants such as air, water and dust particles.

The functioning of radial lip seal is based on the formation of an oil film between the lip and the shaft. As described by Kammüller [1], elastomer lip seals operate by adopting a viscous inverse pumping regime, generated by the micro-geometry of the lip. It is shown that the pumping flow strongly depends on lip roughness [2] or micro-undulations [3–5]. In installed conditions, the lip geometry of a successful radial lip seal leads to a contact approach angle at the oil side significantly larger than the contact approach angle at the air side. Therefore, the resulting contact pressure profile has a maximum near the oil side of the contact. The frictional shear stress generated by the shaft rotation leads to a maximum tangential displacement that coincides with the maximum radial contact pressure. Therefore, the skewed asperities act like micro pumps that could pump oil into the contact from either side, but with a higher potential pumping capacity at the air side. In steady state non-leaking conditions, a hydrodynamic equilibrium is established and a meniscus separates the liquid from the air side of the seal [6].

Starting with Jagger's work [7,8], the shaft is assumed smooth compared to lip roughness. Typically, the arithmetic average roughness $R_a$ of the shaft is 10 times less than the lip surface. However, practical experiences reflected in the Rubber manufactures Association (RMA) standard on upper and lower limits on the shaft surface roughness indicate that the shaft roughness influences significantly the rotary lip seal performances. Referring to Horve book [9]: "if the shaft is too rough or too smooth, the seal will leak". Recently, experimentations have shown that not only the shaft roughness amplitude (measured by $R_a$) affects the lip seal performance, but also that the profile is important [10–12].

The first numerical study that takes into account both lip and shaft roughness has been presented by Salant and Shen [8]. The authors first presented a hydrodynamic model to determine the effects of the shaft surface micro geometry without considering material deformation on the lip surface. Later, they incorporated the elastic deformations of the lip into the model [13] and then they added an asperity contact model [4]. The model has been successfully used to predict the performance characteristics of a lip seal, such as load support, contact area ratio, cavitation area ratio, reverse pumping and average film thickness.

In the present study, a numerical isothermal full film lubrication model is developed and validated by experimentations so as to investigate the shaft roughness effect on rotary lip seal behavior.

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**Nomenclature**

| | |
|---|---|
| $A_1$ | shaft roughness amplitude [μm] |
| $A_2$ | lip roughness amplitude [μm] |
| $b$ | width contact [mm] |
| $C_1$, $C_2$ | compliance matrix (radial and circumferential respectively), [$mm^3$/N] |
| $C_a$ | instantaneous friction torque [N m] |
| $\overline{C}_a$ | friction torque averaged [N m] |
| $C_{10}$, $C_{01}$ | Mooney Rivlin coefficients [MPa] |
| $D$ | Universal function, describing the pressure "$p$" in active zone, and the difference between replenishment and film thickness "$r_f$–$h$" in non active zone |
| $\boldsymbol{D}$ | vector of nodal values of the universal variable |
| $D_a$ | shaft diameter [mm] |
| $F$ | cavitation index ($F$=1 when $p>0$) else ($F$=0) |
| $h$ | film thickness [mm] |
| $h_0(=h_{avg})$ | average film thickness [mm] |
| $h_1$ | shaft sinusoidal roughness [mm] |
| $h_2$ | lip sinusoidal roughness [mm] |
| $N_x$ | peak number according to circumferential direction |
| $N_y$ | peak number according to leakage direction |
| $NE_x$ | finite element number according to circumferential direction |
| $NE_y$ | finite element number according to leakage direction |
| nep | number of gauss points |
| $N_{*k}$ | interpolation function relative to $k$ node |
| NN | total nodes number |
| $p$ | film pressure [MPa] |
| $p_s$ | contact static pressure [MPa] |
| $R$ | shaft radius [mm] |
| $\boldsymbol{R}$ | residual vector of the discretized modified Reynolds equation |
| $R_a$ | arithmetic roughness [mm] |
| Rsk | skewness measurement |
| Sku | kurtosis measurement |
| $\boldsymbol{S}$ | right hand side (RHS) vector member of modified Reynolds equation |
| $t$ | time [s] |
| $T$ | calculation cycle (where $T=\lambda/U$) [s] |
| $U$ | shaft velocity [mm/s] |
| $x$, $y$, $z$ | cartesian coordinates [mm] |
| $\lambda(=\lambda_x)$ | wavelength according to circumferential direction [μm] |
| $\lambda_y$ | wavelength according to leakage direction [μm] |
| $\mu$ | lubricant dynamic viscosity [Pa s] |
| $\rho$ | lubricant density [Kg/$m^3$] |
| $\rho_0$ | air density [Kg/$m^3$] |
| $\tau$ | shearing according to circumferential direction [MPa] |
| $\delta$ | interference between shaft and lip [mm] |
| $\delta_x$ | lip circumferential displacement [mm] |
| $\delta_z$ | lip radial displacement [mm] |

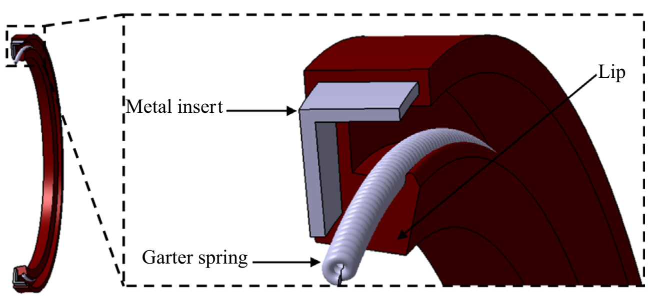


**Fig. 1.** Rotary lip seal.

The model is an alternative of the one proposed by Salant and Shen. The differences come from a different numerical treatment of the Reynolds equation and also from the solutions used to obtain the elastic deformations generated by the hydrodynamic pressure. Salant and Shen used finite volumes method to discretize the Reynolds equation, while the present model is based on the finite elements method. The influence coefficient (or compliance) matrix is obtained here by considering a tridimensional behavior of the lip close to the contact, while Salant and Shen considered a two-dimensional axisymmetric model. It must be mentioned that the present model does not take into account the asperity contact, but the studied functioning conditions lead to full film lubrication conditions, and consequently, there is no contact observed between the lip and shaft roughness.

## 2. Experimental part

### 2.1. Test ring description

Fig. 2 shows the experimental device used to test the seals. A two-phase asynchronous motor of 1.1 kW is used to control the rotational speed of the spindle. A Plexiglas tank contains the liquid to seal. The tank is mounted on a needle bearing and is blocked in rotation by a torque-meter. This way, the friction force generated in the sealing zone is directly measured by the torque-meter. The Plexiglas tank is installed on a micrometric mobile support in order to obtain a very precise adjustment of the seal.

The calibration of the torque meter is carried out by using different suspended masses and varying their attachment position according to the torque meter axis. From measurements, the accuracy of torque meter, which is 0.036 N m, can be deduced.

The temperature measurements are carried out by type K thermocouples. A first thermocouple is placed inside the oil tank. Two additional thermocouples are used to measure the temperature of the seal near the contact zone. Sub-millimeter holes are made on the seal and the thermocouple is positioned as near as possible to the contact surface. The mean value measured by these two thermocouples will be referred to as the film temperature.

The oil used is TOTAL ACTIVA 5000 15 petrol. Fig. 3 shows the variation of the oil dynamic viscosity with the temperature. The oil density is 816 kg/$m^3$ at the ambient temperature (20 °C).

### 2.2. Seal description

Experiments are performed on classical lip seals (*material: fluorocarbon elastomer, Paulstra-Hutchinson, reference 702619*), designed to be mounted on an 85 mm shaft. The radial force is measured for two different seals with and without the elastic spring. The measurements are shown in Table 1 and have been carried out on a laboratory test device according to DIN 3761-9.

Shaft and lip roughness have been measured by an optical interferometry device. The results presented in Fig. 4 show that the shaft and lip topography do not have a Gaussian distribution (Sku > 3), with a profile that has more cavities for both surfaces (Rsk < 0). It must be noted that the shaft roughness amplitude (measured by $R_a$) is here 10 times smaller than the lip roughness amplitude, which is in accordance with the seal manufacture specifications.