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The Effect of Wear on the Performance of a Rotary Lip Seal

Theoretical analysis, combined with experimental verification, is used to study the effect of wear on the performance of a rotary lip seal as characterized by the pumping rate and friction torque. The performance of a rotary lip seal is determined by the sealing lip surface microscopic characteristics and contact characteristics at the sealing zone. The variation of the contact characteristics with wear has been established based on the variation of the sealing lip profiles measured by using a trilinear coordinate measuring instrument. The impression method is used to copy the lip surface microtopography at different wear times and then an optical interferometer is used to measure the impression surface microtopography to obtain the variation of roughness with wear. The variations of the roughness, contact characteristics, and approximate contact temperature with wear are inserted into a mixed lubrication model to obtain the variations of the pumping rate and friction torque. A comparison of the simulated results with those from experimental measurement verifies the validity of the theoretical analysis. [DOI: 10.1115/1.4027623]

Keywords: lip seal, wear, pumping rate, friction torque

1 Introduction

Elastomeric rotary lip seals are widely used in industry to retain lubricant and exclude contamination in rotating machinery and bearing applications. A rotary lip seal is composed of an elastomeric ring bonded to a steel frame and is designed to have an interference fit with the shaft to produce a preload. A garter spring is often used to provide additional force, which compensates for the load and flexibility loss over time. Figure 1 shows the schematic of an installed lip seal and its components for this study, where the x direction represents the circumferential direction, and the y direction represents the axial direction.

Well-designed rotary lip seals do not leak significantly, unlike most other types of rotary seals such as mechanical seals and labyrinth seals. While a micron scale film of fluid lubricates the lip-shaft interface in the sealing zone [1–4], leakage is prevented by a pumping action from the air-side toward the liquid-side of the seal [5–7]. The pumping rate is the most important performance indicator for evaluating whether a rotary lip seal is successful. If the pumping rate is too small, the seal will leak. This pumping action is produced by the combined action of lip surface asperities, shear deformation, and the contact characteristics of the lip-shaft interface [8]. Under dynamic conditions, the asperities on the lip surface are distorted into vanelike shapes due to the shear deformation of the elastomeric lip in the circumferential direction. As liquid is dragged over each vanelike asperity by the shaft rotation, flow is induced in the axial direction. Under the action of the interference and the spring force, the elastic deformation of the elastomeric ring generates a contact pressure distribution with an unsymmetrical trianglelike profile, with the peak closer to the liquid side than to the air-side in the case of a successful lip seal. It has been found that this indicates that the axial location of maximum shear deformation is closer to the liquid-side than to

the air-side, resulting in a net pumping of fluid from the air-side toward the liquid-side [5–7].

However, the loss of material due to wear, which is an inevitable phenomenon, will simultaneously change the lip surface microtopography, the sealing lip profile, and the interference between the elastomeric ring and shaft. The change of microtopography will directly affect the pumping action of the lip seal; the change of the sealing lip profile and the interference will change the contact temperature and contact characteristic of the lip-shaft interface including the static contact pressure and contact width, which also affects the pumping action. In addition, it is known that the wear extent is correlated with the friction torque [9]. Therefore, the friction torque is another important indicator for evaluating the sealing performance.

The ultimate purpose of studying the influence of wear on the sealing performance, as characterized by the pumping rate and friction torque, is to predict the service life of a rotary lip seal. The prediction of seal wear and service life is of great interest to users and manufacturers of lip seals and is a needed research focus in the seal field [10]. This prediction has generally been made on the basis of extensive testing, which is time-consuming and expensive. Thus far, there is a lack of published literature regarding the theoretical prediction of seal wear, in order to reduce or eliminate extensive testing. However, over the past twenty years substantial progress has been made in the numerical simulation of elastomeric seal performance, leading to an increased understanding of the basic physics governing seal behavior [11]. If the physical changes resulting from wear can be taken into consideration in the numerical simulation, it is possible to extend such a simulation to the prediction of seal wear.

The preliminary research effort has been finished, laying a foundation for this study. A mixed lubrication model of a rotary lip seal using flow factors has been established, which is used to calculate the pumping rate and friction torque [12]. The effect of the lip surface asperities is taken into consideration in the model by using a statistical approach.

From Ref. [12], the static contact pressure and contact width are input parameters for the previously described mixed

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Fig. 1 Schematic of the lip seal

lubrication model. By using a trilinear coordinate measuring instrument, the variation of the sealing lip profile with wear time is obtained. Based on the measurement results, the variations of the contact characteristics are then obtained from the finite element analysis (FEA), utilizing a commercial package, ANSYS. A measurement of the contact width is used to validate the FEA results.

Roughness is a comprehensive indicator of the lip surface asperities characteristic [13,14]. The impression method is used to copy the lip surface microtopography at different wear times and an optical interferometer is used to measure the copied surface microtopography to obtain the variation of roughness with wear. In addition, the contact temperature changes with wear and, in turn, affects the wear process. As discussed in Sec. 2, the variation of contact temperature with wear time is experimentally determined and accounted for in the mixed lubrication model. The variations of the roughness, sealing lip profile, static contact characteristic, and contact temperature are then inserted into the mixed lubrication model, in order to obtain the variation of the pumping rate and friction torque with the wear time. Finally, a bench test is used to validate the numerical simulation results.

2 Experimental Approach

The goals of the experiments are to provide the values of the relevant parameters required for the theoretical model and to validate the theoretical model.

2.1 Bench Test. The bench test is used to measure the temperature in the sealing zone and to provide lip seal samples for the measurement of the lip surface roughness, lip profiles, contact width, and radial force. It is also used to measure the reverse pumping rate and friction torque to validate the numerical simulation results.

Figure 2 shows a schematic diagram of the experimental apparatus. The seal is installed backwards, with the original air-side of the seal now facing the oil chamber, becoming the oil-side. Consequently, the leakage rate with this configuration is equivalent to the reverse pumping rate of the seal. This configuration is consistent with the flooded boundary conditions of the mixed lubrication model [12], although it deviates from the actual operating conditions when starvation occurs. The leakage from the oil chamber is collected by the oil-collecting cup and measured using a JJ series electronic scale with a resolution of 0.01 g. A JN338F static torque sensor is connected to the chamber to measure the friction torque.

The test lubricant is hydraulic oil ISO VG-32 with a viscosity of 0.0279 Pa s at 40 °C. The oil level is at the shaft centerline. From Refs. [15,16], the viscosity changes with rotation speed, due to the change in the interface temperature, leading to a change in the sealing performance. Therefore, in this study the rotation speed is set at 2000 rpm in order to exclude the influence of speed. However, the viscosity will also change with wear, due to the change of the interface temperature in the sealing zone. Its change

Fig. 2 Schematic of the bench test rig

affects the film thickness, resulting in changes in the pumping rate and friction torque. By measuring the contact (interface) temperature at different wear times and combining it with the viscosity-temperature relationship, the relationship between the viscosity and wear time can be obtained. In this study, a thermal imaging camera is used to measure the shaft surface temperature adjacent to the sealing zone, as shown in Fig. 2, in order to approximate the contact temperature.

The material of the elastomeric ring (as shown in Fig. 1) used in this study is nitrile butadiene rubber (NBR). The nominal diameter of the lip seals is 100 mm. Ten hours (10 h), 25 h, 50 h, 100 h, 200 h, 300 h, 400 h, and 500 h are chosen as the bench test time nodes to study the change of the surface roughness and other contact characteristics with wear time. Note that in order to exclude the difference between different lip seals and to decrease the test time, the same lip seal is used for all of the time nodes. The debris is removed with each cleaning between measurements. As a result, the effect of the debris on the wear process is neglected. The test series is repeated three times (with three different seals) and the results are averaged.

2.2 Measurement of the Sealing Lip Profile. The profile of the sealing lip is the most important intermediate variable. Its change, resulting from wear, will decrease the interference of the lip/shaft and change the contact conditions. This will lead to changes in the contact width and static contact pressure distribution, consequently affecting the pumping action of the seal. It is therefore necessary to continuously monitor the changes in the sealing lip profile. Figure 3 shows the device used to measure the lip profile, a trilinear coordinate measuring instrument. The resolution of the instrument is 0.5 μm . It consists of a CCD camera connected to a computer, a microscope sensor, a lip seal support with a 70 deg slope surface, a reflection light source, a transmission light source, an X-Y two-dimensional platform, and a Z one-dimensional platform. The coordinates of the discrete points of the sealing lip are obtained from the instrument and then a smooth curve is drawn through the discrete points to obtain the profile of the sealing lip.

In order to analyze the change in the profile of the sealing lip resulting from wear, it is necessary to place a reference mark on an unworn part of the lip and three other reference marks on the back abutment surface and on the slope surface of the seal support. It is also necessary to place three reference marks on the surface of the X-Y platform. Consequently, the six degrees of freedom

Fig. 3 Schematic of the sealing lip profile measurement

of the sealing lip profile are restricted, which guarantees the same location of the sealing lip profile for each measurement. In order to obtain a smooth measurement curve, at least 50 measurement points are needed. Three specimens are measured for each operating condition and the mean value is used to decrease the experimental error. Note that the lip profiles are measured in a free state; the sealing lip is not subject to any force during measurement.

2.3 Measurement of the Contact Width. Contact width is one of the important parameters that affect the pumping action of lip seals; it also changes with wear. Figure 4 shows the device used to measure the contact width. It consists of an SLR camera connected to a computer, a microscope, a transparent sleeve, a two-dimensional platform, a powerful light source, and a reflecting mirror, which is fixed in the sleeve at 45 deg. The transparent sleeve has the same diameter as the shaft. After the seal is mounted on the sleeve for 15 h to decrease the effect of rubber viscoelasticity, the contact width can be measured by the device. A photograph is taken and the contact width is obtained through the computer post-processing software. In addition, in order to eliminate the measurement error caused by the glass refraction, a grating scale with a fixed width of 0.15 mm (measured by the trilinear coordinates measuring instrument described in Sec. 2.2) is aligned with the seal to calibrate the measured length.

2.4 Measurement of the Radial Force. The radial force also plays a vital role in determining the success of a rotary lip seal. If it is too small to allow the seal surface to follow the vibrations, wear, or eccentricity, the seal will leak. If it is too large, the durability could be diminished due to the high friction torque and consequent increased wear and aging of the elastomer ring. The radial force F_r is equal to the integral of the static contact pressure over the contact area; it can be calculated from

Fig. 4 Schematic of the contact width measurement

Fig. 5 Schematic of the radial force test apparatus

$$F_r = \int_0^{\pi D_i} \int_0^{L_y} p_{sc} dx dy \quad (1)$$

where D_i and L_y represent the shaft diameter and axial contact width, respectively.

Measurement of the radial force is usually carried out on a jig that has a short length of shaft, split in half lengthwise. One-half is rigidly fixed and the other is connected to a force transducer. Figure 5 presents the schematic diagram of the test apparatus for measuring the radial force. The lip seal is mounted as shown in the figure.

The measurement is carried out at room temperature. After the seal is mounted for 15 h, the seal is assumed quasi-stationary and the force value can be read. Each measurement is repeated three times in order to obtain an average, each time rotated by about 120 deg about the vertical direction.

2.5 Roughness Measurement. As previously mentioned, the lip surface microtopography directly affects the performance of lip seals. During operation, material loss resulting from wear inevitably induces a change in the lip surface microtopography, leading to a change in roughness. To measure this change in roughness by optical interferometry, it is necessary to use the impression method, due to the poor reflectivity of the NBR.

The impression method involves attaching a plastic material with good reflectivity to the measured surface. The measured surface will be copied onto the impression material. By measuring the roughness of the impression material, the roughness of the measured surface can be evaluated. Polyurethane, initially in the liquid state, is used as the impression material, since it has good reflectivity [17].

The polyurethane liquid is mixed with a curing agent in equal proportions and evenly stirred. The cleaned lip seal is immersed in the liquid mixture and then placed in a vacuum chamber to remove any remaining air in the liquid. After 30 min, the seal is removed from the vacuum chamber and the polyurethane is allowed to solidify for 48 h in the ultraclean room. The final state is shown in Fig. 6.

After the preceding steps, the polyurethane impression is peeled off of the sealing lip and the impression surface is wiped with

Fig. 6 Physical picture of the impression method

alcohol. The 3D optical interferometer is used to measure the roughness of the polyurethane impression.

3 Numerical Analysis Using a Mixed Lubrication Model

Numerical analysis methods have often been used to study the sealing behavior of lip seals. A mixed lubrication model, previously developed by the present authors, is used to predict the reverse pumping rate and friction torque in this study [12]. The model consists of a coupled hydrodynamic lubrication analysis, asperity contact analysis, and deformation analysis, with an iterative computational procedure. Since the details of the model are given in Ref. [12], only a summary of the model is given in the following text.

The hydrodynamic lubrication analysis in the sealing zone is governed by the Reynolds equation. The flow factors ϕ_{xx} , ϕ_{xy} , ϕ_{yx} , ϕ_{yy} , $\phi_{c,c}$, $\phi_{s,c,x}$, and $\phi_{s,c,y}$ are used to account for the influence of the lip surface roughness, while the shaft surface is assumed to be perfectly smooth. The cavitation index F and universal variable Φ are used to take into account the effect of cavitation. The form of the Reynolds equation is

$$\begin{aligned} & \frac{\partial}{\partial \hat{x}} \left[\hat{h}^3 \left(\phi_{xx} \frac{\partial(F\Phi)}{\partial \hat{x}} + K \phi_{xy} \frac{\partial(F\Phi)}{\partial \hat{y}} \right) \right] \\ & + K \frac{\partial}{\partial \hat{y}} \left[\hat{h}^3 \left(\phi_{yx} \frac{\partial(F\Phi)}{\partial \hat{x}} + K \phi_{yy} \frac{\partial(F\Phi)}{\partial \hat{y}} \right) \right] \\ & = 6\xi \frac{\partial}{\partial \hat{x}} \left\{ [1 + (1-F)\Phi] (\hat{h}_T - F\phi_{c,c}) \right\} \\ & + 6\xi F \left(\frac{\partial \phi_{s,c,x}}{\partial \hat{x}} + K \frac{\partial \phi_{s,c,y}}{\partial \hat{y}} \right) \end{aligned} \quad (2)$$

A finite volume scheme is used to discretize the preceding equation to obtain a system of linear algebraic equations, which is solved for Φ and F by using the alternating direction implicit method. Once Φ is obtained, the dimensionless volumetric pumping rate can be found from

$$\hat{Q}_y = \int_0^1 \hat{q}_y d\hat{x} \quad (3)$$

where

$$\hat{q}_y = -\hat{h}^3 \left[\phi_{yx} \frac{\partial(F\Phi)}{\partial \hat{x}} + K \phi_{yy} \frac{\partial(F\Phi)}{\partial \hat{y}} \right] + 6\xi \phi_{s,c,y} \quad (4)$$

The viscous shear stress on the lip can be computed from

$$\hat{\tau}_{avg} = \frac{\tau_{avg}}{E} = \frac{\mu U}{hE} (\phi_f + \phi_{fss}) + \phi_{fpp} \frac{h}{2E} \frac{\partial(F\Phi)p_{ref}}{\partial x} \quad (5)$$

The flow factors in Eq. (2) are evaluated as described in Refs. [12,14] and take account of the asperity orientation variation in the axial direction, also described in detail in Refs. [12,14].

The asperity contact pressure is obtained from the Greenwood–Williamson surface contact model (G–W model). Assuming a Gaussian distribution of asperities, the contact pressure p_c is given by Eq. (6)

$$p_c = \frac{4}{3} \eta \frac{E}{(1-\nu^2)} \sigma^{3/2} R^{1/2} \frac{1}{\sqrt{2\pi}} \int_{h/\sigma}^{\infty} (u - h/\sigma)^{3/2} e^{-u^2/2} du \quad (6)$$

The friction shear stress produced by the contacting asperities is given by Eq. (7)

$$\tau_f = -f \cdot p_c \quad (7)$$

Since the deformation of the seal is very small after mounting and pressurization, a linear deformation analysis may be used [18,19]. In the present study, the influence coefficient method is used. The normal and circumferential deformations of the lip surface at the i th node can be expressed as

$$(\delta_n)_i = \sum_{k=1}^m (I_n)_{ik} [p_f + p_c - p_{sc}]_k \quad (8)$$

$$(\delta_s)_i = \sum_{k=1}^m (I_s)_{ik} (\tau_{avg} + \tau_f)_k \quad (9)$$

The influence coefficients are obtained from an offline FEA, utilizing a commercial package, ANSYS, as is the static contact pressure distribution.

Since the previously discussed equations are strongly coupled, it is necessary to use an iterative computational procedure, as illustrated in Fig. 7. Convergence is reached when the relative error of the film thickness is less than the given error limit of 0.001% (10^{-5}). Prior to convergence, the film thickness is relaxed with a coefficient of 0.21. After all of the computations are finished, auxiliary calculations are performed for such characteristics as the reverse pumping rate Q and friction torque T , which can be calculated from Eqs. (10) and (11)

$$Q = \hat{Q}_y \frac{p_{ref} \sigma^3}{12\mu} \rho_f \quad (10)$$

$$T = \pi D_i L_y (\tau_{avg} + \tau_f) \frac{D_i}{2} \quad (11)$$

4 Results and Discussion

4.1 Variation of Contact Characteristics With Wear.

Figure 8 shows the measured sealing lip profiles at different wear times obtained from the trilinear coordinate measuring instrument described in Sec. 2.2. It can be seen that there are two stages of wear: during the initial stage of 0–50 h, wear is rapid and then

Fig. 7 Computational procedure

Fig. 8 Sealing lip profiles versus wear time

Fig. 9 Static contact pressure distributions versus wear time

becomes slower during the steady wear stage of 50–500 h. Additionally, it can be seen that wear of the lip mainly focuses on the air side; the profile near the oil side changes little. In the figure, the horizontal coordinate s represents the distance to the back abutment surface (as shown in Fig. 1) and the vertical coordinate r represents the distance to the shaft center line.

For the sealing lip profile at each wear time, the FEA is carried out to obtain the contact characteristics of a lip seal by using ANSYS, as described in Ref. [12] using the material data of Ref. [12] at ambient temperature. The computed static contact pressure distributions for different wear times are shown in Fig. 9. It is noted that the contact zone extends more toward the air side of the lip seal than toward the liquid side, indicating that wear mainly occurs on the air side, which is consistent with the lip profiles in Fig. 8. The difference of the abscissa of each pressure distribution represents the contact width, which can be measured by using the test apparatus shown in Fig. 4.

Figure 10 shows the calibration and measurement of the contact width. For the fixed width of the grating scale of 0.15 mm, it can be seen from Fig. 10(a) that 1 pixel is equal to $0.9328 \mu\text{m}$. Figure 10(b) shows the measurement result of the 0 h state; it yields a contact width of 0.117 mm. From Fig. 9, it is seen that the simulated contact width is about 0.113 mm. The relative error between the simulated and experimental values is 3.7%. The simulated and experimental contact width of each wear time can be obtained by using the same method as previously discussed, which is shown in Fig. 11. As can be seen, the contact width rapidly increases during the initial stage, corresponding to 0–50 h, and then slowly increases. The average relative error over the entire wear time range is less than 7%, which validates the FEA.

From the contact pressure distribution, the radial force can be calculated from Eq. (1). Figure 12 shows a comparison between the simulated and experimental radial force. It is believed that the

Fig. 10 Calibration and measurement of the contact width

Fig. 11 Contact width versus wear time

Fig. 12 Radial force versus wear time

small discrepancy between theory and experiment is due to aging of the rubber.

4.2 Variation of Roughness With Wear. As an example, consider the lip seal with a wear time of 500 h. Figure 13 shows the microtopography image of the polyurethane impression surface, obtained from the 3D optical interferometer. Figure 14 shows measurements of the polyurethane impression surface and the sealing lip surface itself (at two different arbitrary locations). The trace is made in the axial direction. The poor reflection of the NBR results in many missing data points in its 2D profile, as shown in Fig. 14(b). From the complete measurement profile of Fig. 14(a), it is shown that the impression method eliminates the problem of poor reflectivity (although it may introduce other measurement errors).

Fig. 13 Surface map measurements of the polyurethane impression

Fig. 14 Measurements of the polyurethane impression and lip seal itself: (a) 2D surface profile of the polyurethane impression and (b) 2D surface profile of the lip seal itself

Fig. 15 RMS roughness versus wear time

Figure 15 shows the variation of the RMS roughness of the sealing lip with wear time at the rotation speed of 2000 rpm and a sump temperature of 40 °C and also contains a formula fitted to the data. In the formula, x represents the wear time and y represents the RMS roughness. The corresponding sealing lip surface RMS roughness at any time can be obtained from the fitting formula. This roughness is used in evaluating the flow factors. Once the roughness is obtained, along with the other required parameters, the mixed lubrication model can be used to predict the pumping rate and friction torque.

4.3 The Effect of Wear on Sealing Performance. The pumping rate and friction torque are the two most important

indicators for evaluating the sealing performance. These can be obtained from the mixed lubrication model.

The following seal parameter values are used in the mixed lubrication model: $v = 0.499$, $p_s = p_a = 0.1$ MPa, $p_{cav} = 0$, $D_i = 100$ mm, $f = 0.15$, $h_s = 1.4$ μm, $\rho_f = 857.1$ kg/m³, and $\mu = [8.71 \times 10^{-2} \exp(-2.9 \times 10^{-2}T)]$ Pa s [20]; the unit of T is °C. Figure 16 shows the measured variation of the contact temperature T with wear time at the rotation speed of 2000 rpm. As the lip microtopography changes with wear, the asperity cross section changes simultaneously in the axial and circumferential directions. As a first approximation, for simplicity, it is reasonable to assume that the aspect ratio of the asperities is invariant in time. Consequently, the flow factors only change with the RMS roughness. The contact width L_y is shown in Fig. 4 and L_x is set equal to L_y .

The experimental results and the corresponding theoretical simulation results for the pumping rate and friction torque are shown in Figs. 17 and 18. From Fig. 17, as the wear time increases, the pumping rate decreases, resulting from the decrease of the asymmetry of the static contact pressure distribution shown in Fig. 9 and the decrease of the sealing lip surface roughness shown in Fig. 15. As previously mentioned, these two factors are the necessary conditions for producing the pumping rate. During the initial stage of wear, corresponding to the range from 0 h to 50 h, the rapid decrease of the asymmetry leads to the rapid decrease of the pumping rate. As the wear time continues to increase, both the asymmetry and the surface roughness slowly change so that the pumping rate slowly decreases. In addition, during the actual operation of lip seals, aging in oil is inevitable, which will also deteriorate the sealing lip microtopography and change the static contact characteristic, further leading to a decrease in the pumping rate. The influence of aging in oil is not taken into consideration in the present mixed lubrication model, thus the simulation predictions of the pumping rate are larger than the experimental results.

Fig. 16 Contact temperature versus wear time

Fig. 18 Friction torque versus wear time

Fig. 17 Pumping rate versus wear time

The influence of aging in the mixed lubrication model is the subject of ongoing research work.

From Fig. 18, it can be seen that the friction torque first rapidly increases, during the initial stage corresponding to the range from 0 h to 50 h, then gradually decreases. The experimentally measured decrease is larger than that of the simulation results. During the initial stage of wear, the contact temperature rapidly increases, as shown in Fig. 16, causing the lubricant oil viscosity to rapidly decrease, leading to a decrease in the lubricant oil film thickness. The decrease of film thickness leads to increased asperity contact, so that the friction torque increases. As the wear time continues to increase, the increase of the oil film thickness due to the increase of viscosity resulting from the decrease of contact temperature, along with the decrease of the static contact pressure and radial force, lead to the friction decrease. During the operation of lip seals, aging in oil will also lead to the decrease of the radial force, leading to a smaller radial force than that of the simulation result, as shown in Fig. 12. The influence of aging is not taken into consideration in the mixed lubrication model. Consequently, the decrease in the experimental friction torque is larger than that of the simulation result, as shown in Fig. 18.

Referring back to Fig. 16, the contact temperature is directly dependent on the friction power consumption, which is determined by the friction torque. The variation in friction torque shown in Fig. 18 leads to the initial increase and then gradual decrease of the contact temperature. Thus, Figs. 16 and 18 are consistent.

5 Conclusion

Theoretical analysis, combined with experimental verification, is used to study the effect of wear on the performance of a rotary

lip seal, as characterized by the pumping rate and friction torque. The performance of a rotary lip seal is determined by the sealing lip surface microscopic characteristic and contact characteristics at the sealing zone. The variation of the contact characteristics with wear has been established based on the variation of the sealing lip profiles measured by using a trilinear coordinate measuring instrument. This study has also measured the surface microtopography in order to obtain the variation of roughness with wear. As the wear time increases, the roughness gradually decreases. Inserting the variation of roughness and contact characteristics with wear into the mixed lubrication model allows one to obtain the variation of the pumping rate and friction torque with wear. As the wear time increases, the pumping rate decreases, first sharply and then gradually, and the friction torque first sharply increases and then gradually decreases. Comparing the simulated results with those from experimental measurements verifies the validity of the theoretical analysis. The aforementioned work lays a foundation for studying lip seal life, which is the subject of ongoing research.

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Nomenclature

- D_i = shaft diameter
- f = friction coefficient
- F = cavitation index
- F_r = radial force
- h = nominal film thickness, separation between surface means
- $\hat{h}$ = dimensionless film thickness, h/σ
- h_s = static undeformed film thickness
- I_n = influence coefficient for normal deformation
- I_s = influence coefficient for circumferential deformation
- K = aspect ratio of solution space, L_x/L_y
- L_x = length of solution domain in the x direction
- L_y = length of solution domain in the y direction (contact width)
- p_a = ambient pressure
- p_c = contact pressure
- p_{cav} = cavitation pressure
- p_{ref} = characteristic reference pressure
- p_s = sealed pressure

p_{sc} = static contact pressure
 Q_y = dimensionless pumping rate in the y direction
 $\dot{q}_y$ = dimensionless pumping rate in the y direction per unit length in the x direction
 U = linear speed
 ν = Poisson's ratio
 $\hat{x}$ = dimensionless circumferential coordinate, x/L_x
 $\hat{y}$ = dimensionless axial coordinate, y/L_y

Greek Symbols

ξ = dimensionless number, $\mu UL_x/p_{ref}\sigma^2$
 δ_n = normal deformation of lip surface
 δ_s = circumferential deformation of lip surface
 μ = lubricant dynamic viscosity
 ρ_f = density of the full film (uncavitated)
 σ = the standard deviation of surface heights
 $\hat{\tau}_{avg}$ = dimensionless average viscous shear stress in the x direction, τ_{avg}/E
 τ_f = shear stress due to contacting asperities
 Φ = variable representing pressure/average density, defined by Eq. (2)
 $\phi_{c,c}$ = dimensionless density flow factor
 $\phi_f, \phi_{fss}, \phi_{fpp}$ = dimensionless shear stress factors
 $\phi_{s,c,x}, \phi_{s,c,y}$ = dimensionless shear flow factors
 ϕ_{xx}, ϕ_{xy} = dimensionless pressure flow factors
 ϕ_{yx}, ϕ_{yy} = dimensionless pressure flow factors

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