A mixed thermal elastohydrodynamic lubrication model of rotary lip seal

Xiaokai Huang¹, Shouwen Liu¹ and Chao Zhang²,³

¹Beijing Institute of Spacecraft Environment Engineering, China Academy of Space Technology, Beijing, P.R. China
²School of Automation Science and Electrical Engineering, Beihang University, Beijing, P.R. China
³Research Institute of Frontier Science, Beihang University, Beijing, P.R. China

Abstract
Rotary lip seal is used in various applications where the rotation shaft needs to be sealed, such as hydraulic pumps, fuel pumps, camshafts, crankshafts, and so on. Many thermal elastohydrodynamic lubrication models of rotary lip seal have been introduced, and most of these models neglect the asperity contact. This article proposes a mixed thermal elastohydrodynamic lubrication model of rotary lip seal, in which the microstructure of sealing lip surface, influence of temperature on fluid viscosity, and deformation of lip surface, as well as the asperity contact, are taken into consideration. Simulation study is carried out, and the results show that the asperity contact should not be neglected for analyzing the sealing performance of the rotary lip seal. The influence of speed on the sealing performance is also analyzed based on the proposed model.

Keywords
Rotary lip seal, thermal analysis, mixed lubrication, elastohydrodynamic model, Reynolds equation

Introduction
Rotary lip seals are usually installed in rotating equipment at the end of rotary shaft, which isolate the transmission components from the output components to prevent leakage of lubricating oil. Rotary lip seal has the advantages of simple structure, small installation position, good sealing performance, long service life,
low cost, good adaptability to the vibration of the machine, and the eccentricity of
the spindle, so it is widely used in aviation industry, automobile industry, construction
machinery, mining machinery, and various other mechanical equipment.\textsuperscript{1}
When the shaft rotates, a thin layer of lubricating oil film exists between the shaft
and the seal lip, which can not only reduce the friction but also prevent leakage of
lubrication. Figure 1 shows the schematic of rotary lip seal. A rotary lip seal usu-
ally consists of oil resistant rubber, steel frame, and garter spring.

Based on the experimental results, Guo et al.\textsuperscript{2} found that under the condition of
normal operation, there is a continuous lubricating oil film between the seal lip
and shaft. Later they found that the friction torque between the seal lip and shaft
under dry friction is much larger than that under lubricated condition, and the
relationship between velocity and friction coefficient was analyzed.\textsuperscript{3,4} Fowell et al.\textsuperscript{5}
measured the thickness of the lubricant film directly on the rotary lip seal of the lip
contact area. Fatu and Hajjam\textsuperscript{6} found out that the thickness of the lubricant film
is about several microns in normal operation condition.

The theory of reverse pumping effect has been widely accepted.\textsuperscript{7} In the experi-
ment, it has been observed that all the lip surfaces of the rotary lip seal are irregu-
lar. It was assumed that the shear force in the oil film causes the corrugation
deformation. When the shaft rotates, the seal surface works like a viscous shear
pump, and the oil is pumped from air side to oil side. Jia et al.\textsuperscript{8} found that the
reverse pumping effect was related to seal lip surface roughness. Frölich et al.\textsuperscript{9,10}
tested normal and failed rotary shaft seals with hollow glass shaft, and found that
sealing performance was related to surface roughness distribution. Yang et al.\textsuperscript{11}
established a simplified analytical model, in which it was assumed that the shear
force in the lubricating oil film causes deformation of the seal, and the maximum
circumferential displacement was closer to the oil side. Salant and Flaherty\textsuperscript{12}
confirmed that there was an asymmetric tangential deformation on the seal lip surface.

Many lubrication models of rotary lip seal did not consider the thermal effect.
However, thermal effect has a great impact on the performance of rotary lip seal.\textsuperscript{13}
Nakanishi et al.\textsuperscript{14} established a finite element model, which assumed that the lip

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{diagram.png}
\caption{Schematic of a typical rotary lip seal.}
\end{figure}
heat was transferred through the shaft and the thermal conductivity of rubber was neglected. Kang et al. established a thermodynamic model, in which the oil film thickness was assumed to be constant. The temperature of the shaft was calculated using a two-dimensional axisymmetric model and the heat transferred to the seal was neglected. Fatu et al. analyzed the influence of micro fluctuation of seal lip on the performance. The combination method of finite element analysis and elastic fluid dynamics analysis was proposed, and the power consumption, reverse pumping rate and temperature of rotary lip seal were analyzed. The results showed that the power consumption of the rotary lip seal was proportional to the square of the shaft speed. The leakage of rotary lip seal was negative, which confirmed the pumping effect, and the pumping rate increased when shaft speed increased. The temperature of the rotary lip seal also increased with the increased of shaft speed. Then Hajjam and Bonneau established a heat balance model of rotary lip seal, which considered the roughness of rotary lip seal. In this research, it was assumed that the heat was produced by the friction between the shaft and the seal lip, and was lost through the shaft and the surrounding oil by heat convection. The results showed that the influence of different roughness on the oil film thickness was greater than that of the rotating speed. The heat was proportional to the speed of the shaft, and the pumping rate was proportional to the shaft speed.

Most of these thermal elastohydrodynamic (TEHD) lubrication models can calculate the lubrication status of the contact surface for rotary lip seal considering thermal effect, and can get comprehensive results. However, the abovementioned models did not take the micro contacts between lip surface and shaft surface into account. Guo et al. built a mixed elastohydrodynamic lubrication model of rotary lip seal at constant temperature, and demonstrated that the micro contacts had certain influence on the accuracy of calculation results and should not be neglected. This article builds a mixed TEHD lubrication model of rotary lip seal, in which the micro contacts between lip surface and shaft surface are taken into consideration. A rough surface simulation method based on a two-dimensional digital filter is used to simulate the rough surfaces of shaft and sealing lip. Fluid mechanics analysis, contact mechanics analysis, as well as deformation mechanics analysis are proceeded to build the mixed TEHD model for rotary lip seal.

The rest of the article is organized as follows. In Section 2, first the rough surfaces of shaft and sealing lip are simulated based on a two-dimensional digital filter, then the mixed TEHD model for rotary lip seal is built based on fluid mechanics analysis, contact mechanics analysis, as well as deformation mechanics analysis. In Section 3, simulation study is carried out, and asperity contact influence and the effect of shaft speed on sealing performance are discussed. The conclusions are in Section 4.

**Numerical model and analysis**

First, the following assumptions are made:
1. Shaft surface is perfectly smooth, and seal lip surface is rough.
2. Because the stiffness of the shaft is much larger than that of the seal, so the shaft is considered as rigid body, and the lip seal is regarded as elastic body. The shaft is rotating, and the lip seal is static.
3. The influence of oil film curvature is neglected; therefore, the translation speed is used instead of the rotation speed.
4. The shear deformation of the asperities on the rough lip surface does not affect the normal macroscopic deformation of the lip seal, so the lip seal can be considered as asymmetrically macroscopic.
5. In order to calculate the reverse pumping rate, the air side is assumed to be filled with fluid all the time.

**Numerical simulation of rough surfaces**

It had been proved that the exponential and exponential cosine autocorrelation function can express many random phenomena very precisely.\(^{12}\) The experimental results of Whitehouse and Archard showed that the height distribution of many engineering surfaces can be described by the Gauss distribution, and the surface profile has an exponential autocorrelation function.\(^ {18}\) In this article, a rough surface simulation method based on two-dimensional digital filter is used to simulate the 3D rough surfaces of shaft and sealing lip, as shown in Figure 2. The exponential autocorrelation function can be given by

\[
R(\tau_x, \tau_y) = \sigma^2 \exp \left( -2.3 \left( \frac{\tau_x}{\lambda_{avg-x}} \right)^2 + \left( \frac{\tau_y}{\lambda_{avg-y}} \right)^{1/2} \right) \tag{1}
\]

where \(\sigma\) is root mean square roughness (RMS) of the surface, \(\lambda_{avg-x}\) represents correlation length in the direction of \(x\), and \(\lambda_{avg-y}\) is correlation length in the direction of \(y\). If the surface is isotropic, \(\lambda_{avg-x} = \lambda_{avg-y}\).

**Fluid mechanics analysis**

When the hydrodynamic oil film lubricates the contact zone, a pumping action which absorbs oil from the air side to the liquid side will occur, which is called the pumping effect. Due to the combined influence of the interference and the spring

![Figure 2. Surface topography of the seal.](image-url)
force, a contact pressure distribution is generated by the elastic deformation from
the elastomeric ring. Profile of the distribution is asymmetrical, and when the shaft
rotates, the asymmetrical profile of the distribution will generate the pumping
action. The pumping rate is one of the most important performance characteristics
of the rotary lip seal, and needs to be calculated in the establishment of TEHD
model. When the lip seal is installed on the shaft with interference, because the
stiffness of the shaft is much larger than that of the seal lip surface, deformation of
the asperity on the seal lip surface will occur. Once the shaft rotates, a hydrody-
amic oil film is formed on the contact surface between the seal lip and shaft. The
dynamic pressure oil film is dynamically distributed on the contact zone, and the
elastic deformation of each point on the lip surface is continuously changing.

Therefore, this article assumes that the average distance between the two sur-
faces of the seal lip and shaft is the average film thickness when the shaft does not
rotate. The film thickness between any point on the seal lip and shaft surfaces can
be given by

\[ h(x, y) = h_{avg} + h_{seal}(x, y) \]  \hspace{1cm} (2)

where \( h_{avg} \) is lip roughness, and \( h_{seal} \) represents the seal surface height distribution.

As illustrated in Figure 1, here \( x \) is circumferential direction, and \( y \) is axial
direction.

Once the shaft is rotating, a hydrodynamic lubricant film will be generated, and
the oil film thickness can be expressed as

\[ h(x, y) = h_{avg} + h_{seal}(x, y) + d_e(x, y) \]  \hspace{1cm} (3)

where \( d_e \) is the elastic deformation of the lip surface under the oil film pressure.

Due to the existence of surface roughness, the two relatively sliding surfaces will
be separated from the micro wedge. According to the basic theory of fluid lubrica-
tion, a discrete hydrodynamic region will be generated. If the hydrodynamic pres-
sure is large enough to form hydrodynamic lubrication, the governing equations
can be expressed by the Reynolds equation at hydrodynamic lubrication zone, as
given by

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial p_f}{\partial x} \right) + \frac{\partial}{\partial y} \left( h^3 \frac{\partial p_f}{\partial y} \right) = 6\mu U \frac{\partial h}{\partial x}
\]  \hspace{1cm} (4)

where \( p_f \) is the film pressure, \( U \) indicates the speed of shaft, \( h \) is the film thickness
distribution, and \( \mu \) is the dynamic viscosity of lubrication.

The Reynolds boundary condition adopted in this article is given by

\[
\begin{align*}
\{ p(x, y_{in}) &= p_{seal}, p(x, y_{out}) = p_e \\
 p(0, y) &= p(l, y) \\
 p(x, y) &\geq 0
\end{align*}
\]  \hspace{1cm} (5)
where \( p_e \) is the environmental pressure, \( l \) indicates the calculated area width in circumference direction. In this article, the finite volume method is used to discretize the Reynolds equation. After the pressure distribution and lubrication film thickness distribution is obtained, the viscosity shear stress and pumping rate can be given by

\[
\tau_f = \frac{h}{2} \frac{\partial p}{\partial x} + \mu \frac{U}{h}
\]  

(6)

\[
q_y = -\frac{h^3}{12\mu} \frac{\partial p_f}{\partial y}
\]  

(7)

**Contact mechanics analysis**

According to the theory of mixed lubrication, when the ratio of oil film thickness over surface roughness is less than three, there will be a rough contact, and the influence of contact stress cannot be ignored. The contact pressure of asperities on seal lip surface is calculated by Greenwood-Williamson contact model. The radius of each contact asperity is regarded as \( R \), and the contact is Hertz contact in Greenwood-Williamson contact model. Furthermore, it is assumed that all rough deformation is elastic deformation in the Hertz contact. Greenwood-Williamson contact model is a statistical based model, and the contact stress and contact area can be given as

\[
A_e = \pi \eta A_n R_e \int_{h(y)/\sigma}^{\infty} \phi(z) \left( z - \frac{h_{\text{avg}}(y)}{\sigma} \right) dz
\]  

(8)

\[
p_c = \frac{F_c}{A_n} = \frac{4}{3} \eta R_e \frac{1}{\sigma} E_e \int_{h(y)/\sigma}^{\infty} \phi(z) \left( z - \frac{h_{\text{avg}}(y)}{\sigma} \right) \frac{1}{\sigma} dz
\]  

(9)

where \( p_c \) is the contact stress, \( A_e \) is the real contact area, \( R_e \) is the average radius of curvature, \( \eta \) is the asperities density, \( A_n \) is the nominal contact area, \( F_c \) is the total contact force, and \( E_e \) is the elastic modulus. In addition, \( \phi(z) \) is the distribution of surface height, which can be expressed as

\[
\phi(z) = \frac{1}{\sqrt{2\pi}\sigma} \exp \left( -\frac{(z)^2}{2\sigma^2} \right)
\]  

(10)

where \( \sigma \) is the surface roughness.

The dry friction shear stress can then be given by

\[
\tau_c(y) = -f p_c(y)
\]  

(11)

with \( f \) indicates the friction coefficient in dry friction condition.
Deformation mechanics analysis

In the deformation analysis, the normal deformation and shear deformation of lip seal can be obtained through the influence coefficient method. Then the height distribution of lubrication fluid is determined. It should be noted that when using the influence coefficient method, one needs to follow the following three assumptions. First, the stiffness of the sealing ring remains constant in the whole calculation process. Second, the deformation at any position of the sealing contact area is linear with the applied load, that is, the theory of small deformation is applied to the seal lip deformation caused by oil film. Finally, micro morphology does not affect the macroscopic deformation, so the elastic deformation of seal lip can simplify the two-dimensional problem. Therefore, the formula for calculating the normal deformation of any node on the seal lip is given by

\[
(\delta_z)_i = \sum_{k=1}^{m} (I_z)_{ik} [p_f + p_{sc}]_k
\]

(12)

The shear deformation of any node is calculated as

\[
(\delta_x)_i = \sum_{k=1}^{m} (I_x)_{ik} (\tau_c + \tau_f)
\]

(13)

where influence coefficient matrix \((I_z)_{ik}\), \((I_x)_{ik}\) and the static contact pressure \(p_{sc}\) are obtained by ANSYS finite element analysis off-line. Here plane 183 is selected as solid element, target 169 is selected as contact target element, and contact 172 is selected as contact element. \(\tau_f\) represents the fluid viscous force and \(\tau_f\) indicates dry friction. The normal influence coefficient matrix \((I_z)_{ik}\) represents the \(i\)th axial node deformation caused by \(k\)th axial node unit normal load. The shear influence coefficient matrix \((I_x)_{ik}\) represents the \(i\)th axial node deformation caused by \(k\)th axial node unit shear load. The material of lip seal is rubber, so Mooney-Rivlin model is selected as the constitutive model.

Lip surface thermal model

The friction of the sealing area will generate a lot of heat, which will cause temperature rise in the sealing surface. One of the most important parameters to characterize the lubricating oil is viscosity, which varies with operating temperature. The effect of temperature on the performance of lip seal cannot be neglected. However, as the sealing area is very small, the temperature change in this area is not large, so the analysis of the sealing lip temperature focuses on its average value. With the operation of lip seal, the temperature of sealing surface rises, as given by

\[
T - T_{ref} = \frac{\Phi}{Q_0C_p + h_c2\pi RL}
\]

(14)
where $h_c$ indicates heat exchange coefficient of oil, $R$ represents shaft radius, $L$ is the length of shaft, $T_{ref}$ represents reference temperature, that is, environmental temperature, $\Phi$ indicates the total heat production of seal surface, $Q_p$ represents leakage rate, and $C_p$ is specific heat capacity of oil.

The thermal effect of rotary lip seal is that friction generates heat and temperature of contact zone increases when the shaft rotates. The temperature increment is closely related to static contact pressure caused by pre-tightening force of spring and interference between shaft and lip seal.

In the process of theoretical analysis, the seal lip static contact load is generally used. The relationship between the axial force and static contact load of seal lip can be expressed as

$$F = \frac{G}{D}$$

(15)

where $F$ is static contact load of seal lip, $G$ is the axial force, and $D$ is diameter of shaft.

The total heating power of seal lip can be given by

$$\Phi = \frac{\pi^2 fnD^2 F}{60} = \frac{\pi^2 fnDG}{60}$$

(16)

where $f$ is friction factor of seal lip in mixed lubrication condition, and $n$ is the rotating speed of shaft.

Once the average temperature of the sealing lip is obtained, the liquid viscosity can be obtained by the relationship between the viscosity of liquid and the temperature. Roelands viscosity–temperature equation and Reynolds viscosity–temperature equation are two of the most frequently-used viscosity–temperature equations. Roelands’ viscosity–temperature equation is given by

$$\mu = \mu_0 \exp\left\{ (\ln \mu_0 + 9.67) \left[ \left( \frac{T}{T_0} - 138 \right)^{s_0} - 1 \right] \right\}$$

(17)

where $\mu_0$ is viscosity at temperature $T_0$, $\mu$ is viscosity at temperature $T$, and $s_0$ is viscosity temperature coefficient.

Reynolds viscosity–temperature equation is given by

$$\mu = \mu_0 \exp[-a(T - T_0)]$$

(18)

where $\mu_0$ is viscosity at temperature $T_0$, $\mu$ is viscosity at temperature $T$, and $a$ is viscosity temperature coefficient.

Compared with the Reynolds viscosity–temperature equation, the Roelands viscosity–temperature equation is more complex. The Reynolds viscosity–temperature can be considered as the simplification of the Roelands viscosity–temperature equation when the range of temperature is limited. Because the variation range of seal lip temperature is wide, the Reynolds viscosity–temperature equation is applied to predict the viscosity of the oil at the lubricated zone in this work.


**Computational procedure**

The solution of lip seal mixed thermal lubrication model is essentially a strong coupling solution of fluid mechanics, contact mechanics, thermal, and deformation. Figure 3 shows the flow chart of the numerical solution. It should be noted that,
although only one loop step is shown in the diagram, in fact, there is a nested loop step in solving the Reynolds equation. When the two cycles converge, the parameters such as pumping rate and friction torque can be obtained.

**Result and discussion**

The mixed TEHD lubrication model is applied to a certain type of rotary lip seal. To analyze the influence of the micro contact between seal lip and shaft and shaft speed on lip seal performance, the numerical calculation of TEHD lubrication model is conducted under different rotating speed. The Reynolds equation is solved by finite difference method. The influence of micro contact between seal lip and shaft is discussed. Average film pressure, maximum film pressure, minimum film thickness, heat productivity, average temperature of the contact area, and friction torque are calculated for different operating conditions. The parameters are first set, as shown in Table 1. In this section, the influence of the asperity contact is first discussed, and the necessity of the consideration of asperity contact is proved. Then the influence of rotation speed on the sealing performance is studied based on the proposed model.

**Asperity contact influence**

The Greenwood and Williamson contact model (G-W model) is used to analyze the contact condition between the shaft surface and sealing lip surface from a statistical point of view. Then the contact pressure and dry friction shear stress caused by contact between the two surfaces are calculated, and the contact pressure in axial direction compared with fluid pressure in axial direction are shown in Figure 4. It can be seen from Figure 4 that both contact pressure and fluid pressure experience several large fluctuations. The maximum value of fluid pressure is about 1.5 MPa, and the maximum value of contact pressure is about 0.8 MPa. The maximum value of fluid

| Parameter | Meaning | Value |
|-----------|---------|-------|
| $T_{ref}$ | Reference temperature | 30°C |
| $h_{avg}$ | Average liquid film thickness | 3 μm |
| $p_s$ | Environmental pressure | 0.1 MPa |
| $R$ | Shaft radius | 60 mm |
| $\lambda_x$ | Autocorrelation length | 1.667 μm |
| $\lambda_y$ | Autocorrelation length | 5.0 μm |
| $E$ | Elastic modulus of seal lip | 9.8 MPa |
| $\nu$ | Poisson ratio | 0.5 |
| $f$ | Friction coefficient | 0.1 |
| RMS | Roughness of seal lip | 1 μm |

RMS: root mean square.
pressure is about twice as that of contact pressure. Hence, contact between the two surfaces cannot be ignored from the view of pressure supporting.

Fluid torque in axial direction compared with contact torque in axial direction is illustrated in Figure 5, with $U = 1\, \text{m/s}$, $\sigma = 1\, \mu\text{m}$. It can be seen from Figure 5 that the plot of fluid torque is generally smooth, and only a few small fluctuations exist. However, for the plot of contact torque, several large fluctuations exist. Furthermore, although the fluid shear stress is much bigger than dry friction shear stress, as shown in Figure 5, the deviation of dry friction shear stress is not
negligible compared with fluid shear stress. Hence, the asymmetry of tangential deformation mainly depends on the friction shear stress distribution, that is, dry friction has significant impact on pumping effect. As mentioned before, pumping effect is one of the most important performance characteristics of rotary lip seal and affects the sealing performance; hence, the asperity contact cannot be neglected when analyzing the sealing performance of the rotary lip seal.

**Effect of shaft speed on sealing performance**

Figure 6 shows the relationship between the average oil film thickness of the calculation zone and the rotation speed. It can be seen that the average oil film thickness is significantly affected by the rotation speed, and the average oil film thickness increases with the increase of the rotation speed. This is because that higher rotation speed can provide higher hydrodynamic pressure, and higher oil film thickness is obtained. Hence, it can be seen that higher speed is helpful to maintain the lubrication of the sealing zone for rotary lip seal.

Figures 7 and 8 show the influence of rotation speed of shaft on the maximum and minimum hydrodynamic pressure in the sealing zone, respectively. As shown in Figures 7 and 8, the maximum film pressure will decrease when rotation speed increases and the minimum film pressure will increase when rotation speed increases. The maximum film pressure is significantly affected by rotation speed when rotation speed is low, and the maximum film pressure will reach the minimum value when rotation speed is over 2500 rpm. The minimum film pressure is linearly related to the rotation speed. It can be seen that the hydrodynamic pressure distribution of the sealing zone will become smooth with the increase of rotation speed.

Figure 9 shows the relationship between leakage rare and rotation speed. As shown in Figure 9, the leakage is negative when the sealed fluid pressure is lower
Figure 7. Relationship between maximum film pressure and rotation speed.

Figure 8. Relationship between minimum film thickness and rotation speed.

Figure 9. Relationship between leakage rate and rotation speed.
than ambient pressure, which means that the leaking fluid will be reversely pumped to the fluid side when the shaft rotates (pumping action). Furthermore, the pumping rate will increase with the increase of rotation speed. Hence, it can be seen that the high rotation speed is helpful for both the lubrication of the sealing zone and the sealing performance of the rotary lip seal.

Figures 10 and 11 show the influence of rotation speed on the heat production and average temperature of sealing zone. As shown in Figures 10 and 11, the heat productivity is approximately linearly related to the rotation speed. Similarly, the average temperature of the sealing zone will also increase when rotation speed increases, and the relationship is also approximately linear. As rotary lip seal is mostly made of rubber, aging will also be accelerated at higher temperature. When temperature increases, the compression set of rotary lip seal will increase.

Figure 10. Relationship between heat productivity and rotation speed.

Figure 11. Relationship between average temperature and rotation speed.
dramatically, and the stress-strain ratio and friction torque will also increase. The pumping rate, which is one of the most important performance characteristics of rotary lip seal, will gradually decrease when temperature increases. All these phenomena show that the aging speed is bigger at higher temperature. It can be concluded that although higher rotation speed of shaft is helpful for the sealing performance of the rotary lip seal, the higher temperature caused by higher rotation speed will increase the aging speed, and thus reduce the operation lifetime.

Figure 12 shows the relationship between the friction torque and rotation speed. As shown in Figure 12, the friction torque of the rotary lip seal increases with the increase of rotation speed. The increasing rate of the friction torque will decrease when the rotation speed increases.

**Conclusion**

A mixed TEHD lubrication model of rotary lip seal is introduced in this work. The microstructure of sealing lip surface, asperity contact, influence of temperature on fluid viscosity, and deformation of lip surface are taken into consideration in this model. Simulation results show that although the fluid shear stress is more considerable than dry friction shear stress, the deviation of dry friction shear stress is large compared with fluid shear stress, so the asymmetry of tangential deformation mainly depends on the friction shear stress distribution. The influence of shaft speed on the seal performance is also studied by the proposed model. Simulation results show that high rotation speed of shaft will improve the sealing performance of the rotary lip seal, but the temperature of sealing zone will also increase and the aging of the seal will be accelerated in this situation. Future work should focus on simulating the rough surfaces of shaft and sealing lip with more accurate models, including other types of autocorrelation functions. Meanwhile, the calculation efficiency of the proposed model should also be assessed.
Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This study was supported by the National Natural Science Foundation of China (51875015).

ORCID iD

Chao Zhang https://orcid.org/0000-0001-9054-5132

References

1. Liu D, Wang S and Zhang C. A multiscale wear simulation method for rotary lip seal under mixed lubricating conditions. Tribol Int 2018; 121: 190–203.
2. Guo F, Jia X, Suo S, et al. A mixed lubrication model of a rotary lip seal using flow factors. Tribol Int 2013; 57: 195–201.
3. Guo F, Jia X, Lv M, et al. The effect of aging in oil on the performance of a radial lip seal. Tribol Int 2014; 78: 187–194.
4. Jia X, Guo F, Huang L, et al. Parameter analysis of the radial lip seal by orthogonal array method. Tribol Int 2013; 64: 96–102.
5. Fowell MT, Myant C, Spikes HA, et al. A study of lubricant film thickness in compliant contacts of elastomeric seal materials using a laser induced fluorescence technique. Tribol Int 2014; 80: 76–89.
6. Fatu A and Hajjam M. Effect of grooved shaft on the rotary lip seal performance in transient condition: Elasto-hydrodynamic simulations. Tribol Int 2016; 93: 411–418.
7. Jau YY, Miron E, Post AB, et al. Push-pull optical pumping of pure superposition states. Phys Rev Lett 2004; 93(16): 160802.
8. Jia X, Jung S, Haas W, et al. Numerical simulation and experimental study of shaft pumping by laser structured shafts with rotary lip seals. Tribol Int 2011; 44(5): 651–659.
9. Fröhlich D, Magyar B and Sauer B. A comprehensive model of wear, friction and contact temperature in radial shaft seals. Wear 2014; 311(1–2): 71–80.
10. Fröhlich D, Magyar B, Sauer B, et al. Investigation of wear resistance of dry and cryogenic turned metastable austenitic steel shafts and dry turned and ground carburized steel shafts in the radial shaft seal ring system. Wear 2015; 328: 123–131.
11. Yang AS, Wen CY and Tseng CS. Analysis of flow field around a ribbed helix lip seal. Tribol Int 2009; 42(5): 649–656.
12. Salant RF and Flaherty AL. Elastohydrodynamic analysis of reverse pumping in rotary lip seals with microasperities. J Tribol 1995; 117(1): 53–59.
13. Guo F, Jia X, Huang L, et al. The effect of aging during storage on the performance of a radial lip seal. Polym Degrad Stabil 2013; 98(11): 2193–2200.
14. Nakanishi Y, Honda T, Kasamura K, et al. Bio-inspired seal lip for application in electric vehicle coolant pumps. Mech Eng Lett 2018; 4: 1–7.
15. Kang YS, Sadeghi F and Ai X. Debris effects on EHL contact. J Tribol 2000; 122(4): 711–720.
16. Fatu A, Hajjam M and Bonneau D. A new model of thermoelastohydrodynamic lubrication in dynamically loaded journal bearings. *J Tribol* 2006; 128(1): 85–95.

17. Hajjam M and Bonneau D. A transient finite element cavitation algorithm with application to radial lip seals. *Tribol Int* 2007; 40(8): 1258–1269.

18. Whitehouse DJ and Archard JF. The properties of random surfaces of significance in their contact. *Proc R Soc Lon Ser-a* 1970; 316(1524): 97–121.

**Author biographies**

**Xiaokai Huang** received the B.S. and Ph.D. degrees in system engineering from Beihang University, Beijing, China, in 2009 and 2014 respectively. He is currently serving as a senior engineer in Beijing Institute of Spacecraft Environment Engineering, Beijing, China. His research interests include accelerated life testing in astronautic engineering, and fault diagnosis of mechanical components.

**Shouwen Liu** received the B.S. and master degrees in thermal power engineering from Tsinghua University, Beijing, China, in 2000 and 2003 respectively, and he received his Ph.D. degree in aeronautic engineering from Beihang University, Beijing, China in 2010. He is currently serving as a research fellow in Beijing Institute of Spacecraft Environment Engineering, Beijing, China. His research interests include reliability and life testing in astronautic engineering.

**Chao Zhang** received the B.S. and Ph.D. degrees in mechanical engineering from Beihang University, Beijing, China, in 2008 and 2014 respectively. He is currently serving as an assistant professor in School of Automation Science and Electrical Engineering, Beihang University, Beijing, China. His research interests include reliability engineering, accelerated life testing, and prognostics of hydraulic components.