Numerical analysis of the lubrication performance of variable section seal rings

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Abstract. A variable section seal ring is a special type of sealing element that features a regularly changing sealing boundary. This paper proposes a numerical analysis method for calculating the elastohydrodynamic lubrication performance of the sealing surface of a variable section seal, considering factors such as the surface morphology of the seal lip, the initial compression rate of the seal, and the micro-elastic deformation of the rough surface of the seal lip. The three-dimensional distribution of the oil film thickness was obtained. The results show that the oil film thickness on the sealing lip of the variable section seal changes both circumferentially and axially, being thicker in the middle and thinner at the sides when viewed axially, and thicker at the peaks than at the troughs when viewed circumferentially. Higher compression rates of the seal lead to greater friction during operation; different compression rates significantly affect the distribution of the oil film and the oil film pressure on the variable section seal, which directly impacts the lubrication and sealing effectiveness of the seal. Therefore, choosing an appropriate initial compression rate for the seal is very important.

Keywords: Variable section seal rings, Elastohydrodynamic lubrication, Surface roughness, Numerical analysis

1. Introduction

A variable section seal ring is a unique type of sealing element characterized by its regularly changing wavy lubrication boundary. Therefore, when the shaft rotates, the action of the rotating shaft causes the lubricating oil to form a velocity component perpendicular to the boundary on the wavy boundary, thus facilitating the lubricating oil to penetrate into the sealing surface, forming a fluid dynamic pressure lubrication oil film [1]. When the internal oil film pressure of the seal exceeds the external environmental pressure, it prevents external abrasives from entering the seal [2], thereby enabling the variable section seal ring to perform its sealing function. Therefore, by adopting the design of the variable section seal ring, it is possible not only to effectively achieve fluid dynamic pressure sealing but also to avoid friction and wear caused by direct contact between the seal ring and the shaft. The shape of the variable section seal ring is shown in Figure 1:

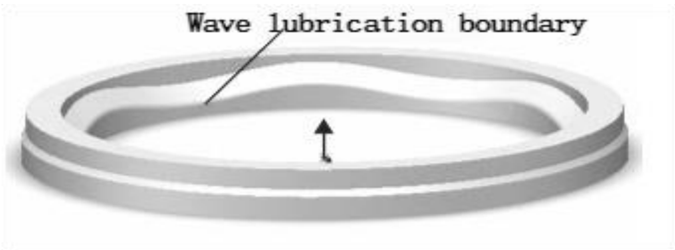

Figure 1. Variable Section Seal Ring

Domestic scholars have conducted research on the sealing mechanism of variable section seal rings and have achieved some important research results. For example, based on a simplified model of a rectangular cross-section, Chen Jiaqing [3] and others derived lubrication and deformation equations for this type of seal. Assuming that the oil film thickness remains constant, Zou Longqing [4] and others performed integral operations on the lubrication equation and derived an approximate formula for calculating the oil film thickness. Using finite element analysis software, Zhu Juan [5] established a three-dimensional model of a seal ring and wrote a numerical calculation program for elastohydrodynamic lubrication of the seal. This program produced a three-dimensional distribution of oil film thickness and oil film pressure. Using finite element analysis software, Lu Xinzhou [6] studied the impact of temperature on the sealing performance of variable section seals.

The aforementioned research works assumed that the surface of the sealing lip is absolutely smooth, without considering the surface morphology of the sealing lip. In practical production, considering the surface morphology of the sealing lip of the variable section seal ring and the surface deformation under working conditions, the range of changes in oil film thickness spans several basic lubrication states. This paper simulates the micro-morphology of the sealing lip of the variable section seal ring and considers its elastic deformation, using the Reynolds equation to construct a numerical calculation method that can calculate the elastohydrodynamic lubrication performance of the variable section seal ring's lubrication surface under different initial compression rates.

2. Modeling of Rough Peaks on the Sealing Surface of the Seal Ring

Generally, after the seal ring surface is ground and polished to achieve an appropriate roughness, the rough peaks of the seal lip are randomly distributed without regularity. Therefore, the micro-morphology of the seal ring can be regarded as a random process. To reduce computational errors, we assume that the surface roughness distribution of the seal lip conforms to a Gaussian distribution [7], with isotropic surfaces. To simplify the problem, based on the characteristics of the sealing surface of the variable section seal ring, we select two peaks and one trough, and simulate the surface roughness $1.8\mu m$, as shown in Figure 2:

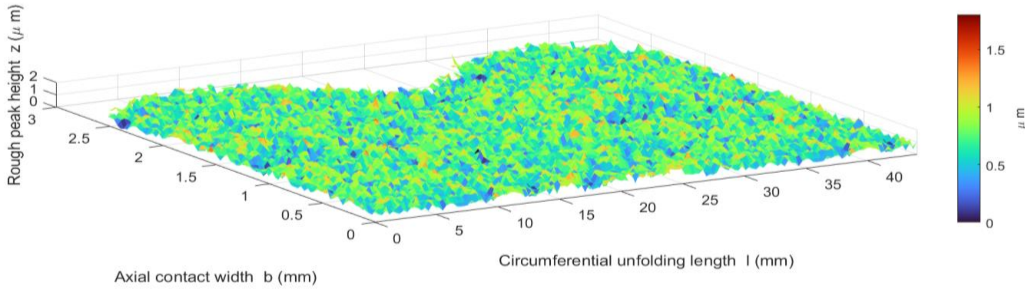

Figure 2. Rough Surface of Variable Section Seal Ring

3. Numerical Calculation Method for Lubrication Performance of Variable Section Seal Rings

3.1. Establishment of Elastohydrodynamic Lubrication Equations

In general, to achieve sealing effectiveness, the fit between the seal ring and the shaft is an interference fit. The seal ring is made of rubber material, while the shaft is generally made of alloy, so the stiffness of the shaft must be much greater than that of the seal ring. After the seal ring and the shaft are installed, macroscopically, the seal ring undergoes significant deformation; microscopically, the micro-convexities on the sealing surface of the seal ring undergo noticeable deformation. When the shaft starts to rotate, due to the wavy curve of the sealing surface of the variable section seal ring, the lubricating oil will have a velocity component perpendicular to the sealing surface, making it easier for the lubricating oil to flow into the wedge-shaped gaps between the micro-convexities of the seal ring. After the shaft speed stabilizes, a layer of lubricating oil film forms between the seal ring and the shaft. The fluid in this oil film is continuously flowing, and before the shaft stabilizes its operation, the morphology of this oil film is actually changing continuously. For convenience of calculation, in this paper, only the distribution of the oil film on the seal ring after the shaft speed stabilizes is calculated. Therefore, to address this issue, an assumption is made: before solving, it is assumed that the variable section seal ring does not rotate, and there is a certain gap between the variable section seal ring and the shaft, with the size of the gap being the average height of the rough peaks on the sealing lip of the seal ring. The thickness of the oil film at this time can be expressed as:

$$h(x,y) = h_{avg} + z_{seal}(x,y) \tag{1}$$

where h_{avg} represents the height of the gap between the variable section seal ring and the shaft.

According to the above assumption, after the shaft rotates steadily, the micro-convexities on the sealing surface of the seal ring will deform. At this time, the thickness of the oil film is:

$$h(x,y) = h_{avg} + z_{seal}(x,y) + d_z(x,y) \tag{2}$$

Where $d_z(x,y)$ represents the elastic deformation of the sealing lip surface under the oil film pressure. In actual situations, due to the regular wavy lubrication boundary of the variable section seal ring, when the shaft rotates, the lubricating oil will enter the tiny wedge-shaped gaps formed by the rough peaks of the seal ring. When the pressure is high enough, a layer of hydrodynamic oil film forms between the seal ring and the shaft. When the seal ring operates stably, the situation is consistent with our assumption. In this case, according to the basic theory of hydrodynamic lubrication, we generally use the Reynolds equation to solve such problems.

When constructing the hydrodynamic lubrication model in this paper, for convenience of calculation, the following assumptions are made:

(1) The surface of the shaft is absolutely smooth, the surface of the sealing lip is rough, the shaft is a rigid body, the seal ring is an elastic body, the shaft rotates, and the seal ring is stationary.

(2) The effect of temperature on the lubricating oil of the seal ring is not considered.

(3) The lubricating oil is considered to be an incompressible Newtonian fluid, with viscosity and density assumed to be constant.

(4) Changes in pressure along the thickness of the oil film direction are not considered, and the oil film thickness is much smaller than the radius of the shaft, neglecting the effect of oil film curvature.

(5) The flow of lubricant between the seal ring and the shaft is laminar, and the inertial effects in its flow are not considered.

(6) The roughness of the sealing lip of the variable section seal ring conforms to a Gaussian distribution.

According to the characteristics of the lubrication of variable section seal rings, the Reynolds equation can be simplified as:

$$\frac{\partial}{\partial X}\left(H^3\frac{\partial P}{\partial X}\right)+\lambda^2\frac{\partial}{\partial Y}\left(H^3\frac{\partial P}{\partial Y}\right)=\Lambda\frac{\partial H}{\partial X} \tag{3}$$

Where $X=x/l$ (l is the circumferential length of the seal region, mm), $Y=y/b$ (b is the axial contact width, mm), $H=h/h_0$ (h_0 is the estimated maximum oil film thickness, μm), $P=p/p_0$ (p_0 is the atmospheric pressure, MPa), $\Lambda=6\eta UI/(h_0^2/p_0)$($U$ is the rotational speed of the shaft, mm/s, η is the viscosity of the fluid, $Pa\cdot s$).

The Reynolds boundary conditions adopted in this paper are:

$$\begin{cases} p(x,y\_in)=p\_seal, p(x,y\_out)=p\_a \\ p(x,y)\ |x=0=p(x,y)|\_(x=l) \\ p(x,y)\geq 0 \end{cases} \tag{4}$$

where p_a represents atmospheric pressure; p_{seal} represents the sealing pressure, y_{in} and y_{out} represent the axial boundary coordinates of the sealing region.

When the shaft operates stably, the sealing lip undergoes a certain elastic deformation, and its governing equations are [7]:

$$d_z(x_k,y_l)=\frac{2(1-v^2)}{\pi E}$$
$$\iint\frac{p(x,y)}{\sqrt{(x-x_k)^2+(y-y_l)^2}}dxdy \tag{5}$$

Where E and v represent the elastic modulus and Poisson's ratio of rubber, respectively.

In the analysis of the sealing characteristics of variable section seal rings, the main focus is on the shear force of the sealing lip. The shear force of the sealing lip of the variable section seal ring should be:

$$\tau_x\Big|z=h=\frac{h\partial p}{2\partial x}+\eta\frac{U}{h} \tag{6}$$

Integrating over the sealing region in the above equation, the circumferential frictional force obtained is:

$$f=\iint\left(\frac{h\partial p}{2\partial x}+\eta\frac{U}{h}\right)dxdy \tag{7}$$

3.2. Numerical Solution Process

The computational process of elastohydrodynamic lubrication for variable section seal rings is as follows:

(1) Input the basic parameters of the variable section seal ring, and assign initial non-dimensional values for the initial oil film pressure, elastic deformation, and initial pressure inside the sealing cavity:

$$p_0=0, D_e=0, p_{seal}=p_a \tag{8}$$

(2) Input the initial distribution of rough peaks for the variable section seal ring.

(3) Assume a non-dimensional initial oil film thickness and solve for the oil film pressure values at each point of the contact area by simultaneously solving the Reynolds equation based on the boundary conditions.

(4) Calculate the elastic deformation and oil film thickness of the seal ring under the pressure results obtained in step (3), and determine whether the oil film thickness converges. The convergence criterion [8] is: $\frac{\sum_{j=2}^{n}\sum_{i=2}^{m}\left|p_{i,j}^{(k)}-p_{i,j}^{(k-1)}\right|}{\sum_{j=2}^{n}\sum_{i=2}^{m}\left|p_{i,j}^{(k)}\right|} \leq \varepsilon$, where ε is 10^{-4} . If it does not converge, correct the oil film thickness and repeat step (3).

(5) If convergence is achieved, continue to calculate the tangential deformation. If the deformation converges, output parameters such as friction force and oil film thickness. If it does not converge, remodel the rough peaks on the sealing surface of the variable section seal ring [9] and repeat steps (3) to (5) until convergence is achieved.

(6) Reduce the estimated maximum oil film thickness and repeat steps (3) to (6), simulating the distribution of the oil film under different initial compression rates until H_{min}. End the program.

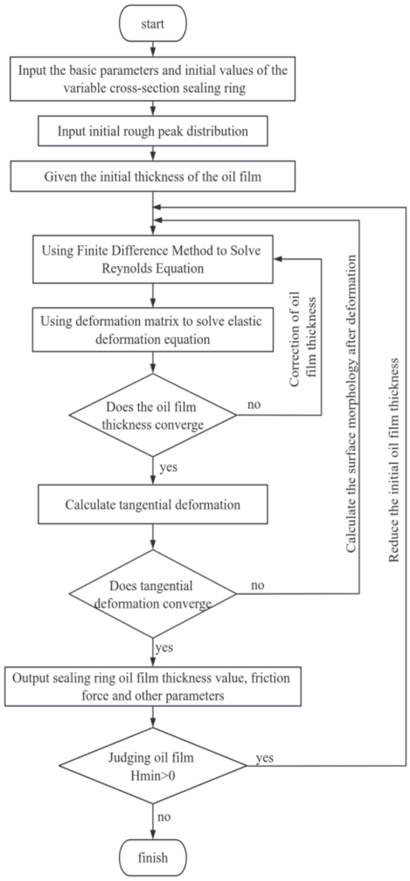

Figure 3. The Computational Process

4. Case Study

Taking a variable section seal ring made of Dingqing rubber material as an example, the above numerical calculation method is applied to lubrication performance analysis.

The selected parameters for the seal ring are as follows: the inner diameter of the seal ring is $55mm$, the number of waveforms on the lubrication boundary of the seal ring is $n = 6$, the amplitude of the seal ring is $e = 0.8mm$, the average contact width is $2.6mm$, the elastic modulus of the seal ring is $E = 9.8MP_a$, Poisson's ratio is $\nu = 0.49$, the viscosity of the lubricating oil is $\eta = 0.02Pa \cdot s$. The atmospheric pressure is $p_a = 0.1MP_a$, the speed is $100r/min$, the initial compression ratio of the seal ring is 8%, and the surface roughness of the seal lip is $1.8\mu m$.

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The Reynolds equation is discretized using the finite difference method, and the solution domain is divided into 80×200 grids. For convenience of study, we unfold the sealing surface into two dimensions and analyze two wave peaks and one wave trough. The numerical analysis results are shown in Figure 4:

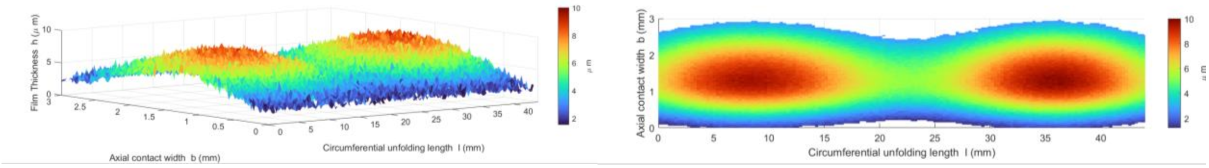
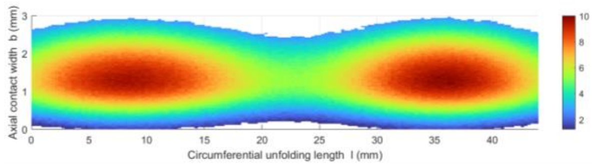


Figure 4. Distribution of Oil Film Thickness on the Seal Ring

According to the calculation results, the oil film thickness on the entire sealing surface exhibits an uneven distribution in the axial direction. The axial oil film thickness of the sealing surface is thick in the middle and thin on both sides, with a maximum thickness of $9.86\mu m$, The distribution pattern of the oil film thickness on the sealing surface in the circumferential direction shows that the oil film thickness at the wave peaks is significantly greater than that at the wave troughs. The average oil film thickness on the entire sealing surface is $5.42\mu m$. It can be seen that the sealing surface between the variable section seal ring and the shaft is not fully lubricated. There are ruptured parts in the oil film, and since the oil film thickness is thicker in the middle and thinner on both sides from the axial perspective, the oil film is most likely to rupture on both sides of the sealing surface. In other words, the lubrication effect on both sides of the seal ring is the poorest. The lubrication effect on the side closer to the outside world is worse than that on the inside, and the rupture of the oil film is more severe. Although the lubrication effect on the outer side is the worst, it effectively ensures that external abrasives are less likely to enter the interior of the seal ring, thereby affecting the service life of the seal ring.



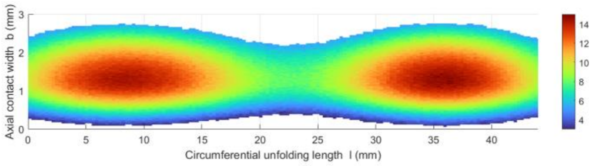


Figure 5. The Distribution of the Oil Film when the Compression Ratio is 8%   Figure 6 The Distribution of the Oil Film when the Compression Ratio is 10%

From Figures 5 and 6, it can be seen that under the same parameters of the seal ring, the distribution of the oil film on the sealing surface remains basically unchanged under different compression ratios, but there are slight differences in the distribution of the ruptured parts of the oil film. The larger the compression ratio, the larger the ruptured part of the oil film. From Table 1, it can be observed that the larger the compression ratio of the seal ring, the thicker the oil film, the higher the sealing pressure, and the better the sealing effect.

Table 1. Relationship between Compression Ratio and Oil Film Thickness and Pressure

Compression Ratio $\phi/\%$	8	10	12	14	16
Average Oil Film Pressure p/MPa	1.53	2.15	2.33	2.54	2.72
Average Oil Film Thickness $h/\mu m$	5.42	8.77	11.26	14.31	17.85

Figure 7 shows the relationship curve between the frictional force of the variable section seal ring and the compression ratio. It can be seen from the figure that the frictional force generated when the variable section seal ring operates increases with the increase in compression ratio.

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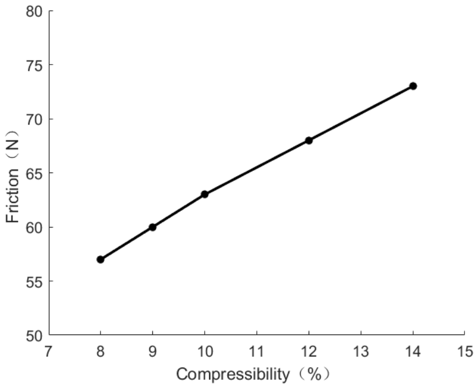

Figure 7. Relationship Curve between Compression Ratio and Frictional Force

5. Conclusion

(1) This paper utilized numerical simulation to study the lubrication performance of variable section seal rings, comprehensively considering the roughness of the seal lip, the compression ratio of the seal ring, and the elastic deformation of the seal lip. The method successfully calculated the lubrication condition of variable section seal rings under different working conditions.

(2) The oil film thickness on the sealing surface of the variable section seal ring is not uniformly distributed. In terms of the circumferential direction, the oil film thickness at the wave peaks of the sealing surface is greater than that at the wave troughs of the seal ring; in terms of the axial direction, the oil film thickness distribution of the variable section seal ring is thicker in the middle and thinner on both sides. The frictional force of the variable section seal ring increases with the increase in the compression ratio of the seal ring.

(3) Different compression ratios of the variable section seal ring have a significant impact on the location of oil film rupture. Excessive compression ratio may cause the oil film at the edges to be easily squeezed out, affecting the lubrication performance and service life of the variable section seal ring. However, with a larger compression ratio, the oil film pressure increases, leading to better sealing effects. Therefore, it is essential to select an appropriate compression ratio when using variable section seal rings.

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