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Numerical Study on Vane Seal Lubrication Model Considering Surface Morphology

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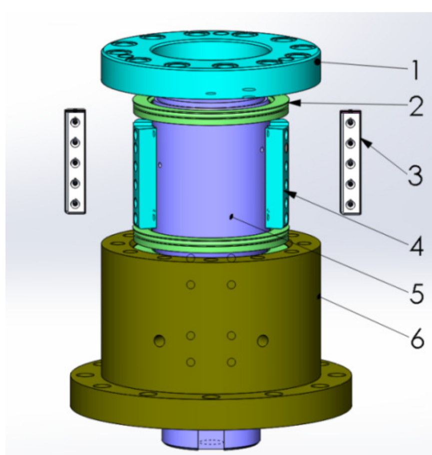
Abstract. Vane seal - hydraulic cylinder inner wall (rotor outer circle) friction pair is studied as object of this paper. The contact pressure mathematical model of vane seal was established by using the generalized Hooke's law, and the distribution of pressure on the contact surface was obtained. Based on the transient average Reynolds equation and the G-W micro-bulge contact model, the hydrodynamic lubrication model of the surface of the hydraulic rotary vane actuators with the surface topography was established. Under the condition of the contact pressure of the vane, the lubrication model was carried out by inverse solution method and the distribution of oil film thickness were obtained. The influence of the surface morphology on the lubrication seal of the vane was analyzed by the known curve of the surface performance parameters of the vane seal surface, which provides a theoretical basis for the sealing of the hydraulic rotary vane actuators.

Introduction

The hydraulic rotary vane actuators is an efficient hydraulic actuator. During the working process, the vane seal surface is in a mixed lubrication state. In the mixed lubrication state, the surface roughness have important influence on the oil film pressure and the lubrication oil film distribution. In 1966, Greenwood and Williamson [1] proposed a robust contact model (G-W model) between rough surfaces and smooth surfaces based on statistical analysis, and obtained a theory of elastic contact near the actual conditions of engineering, pointing out that contact deformation and surface shape related. In 1969, Christensen [2] established a stochastic model that takes into account the effects of pure transverse and longitudinal textures on surface morphology, and corrected the Reynolds equation to allow it to consider the effects of surface roughness. The lubrication problem of the vane sealing surface is an important technical difficulty in the research process of the hydraulic rotary vane actuators sealing system.



1-Cover plate,2-Rotating the ring,3-Vane seal,4-Vane baffle,5-Rotor,6-Stator

Figure 1. Schematic diagram of the overall structure of the hydraulic rotary vane actuators.

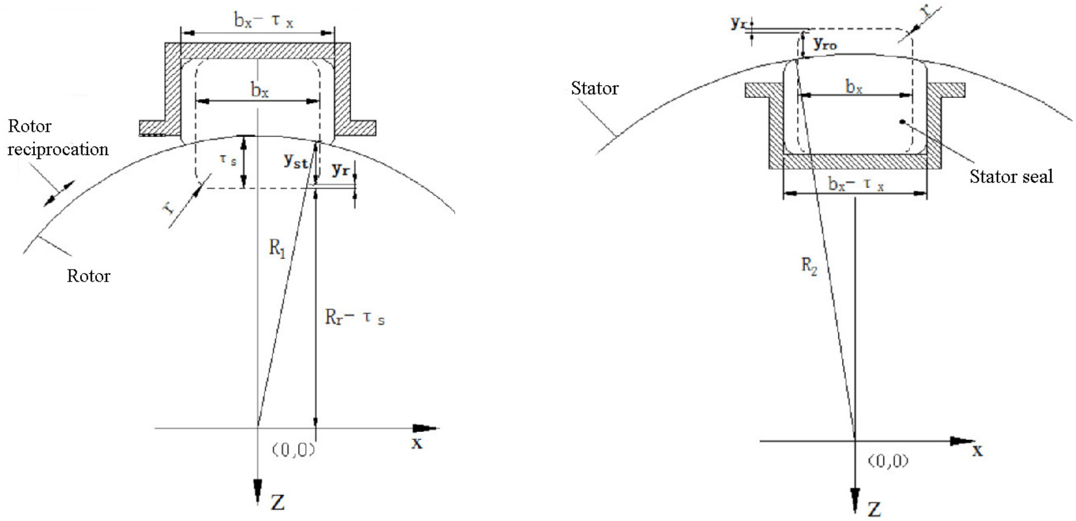
Vane Contact Pressure Mathematical Model

Ignoring environmental factors, the distribution of the contact pressure of the vane is

$$p = \zeta_y = \frac{1}{E}\left[E(\varepsilon_y + \nu\varepsilon_z) + \nu(1+\nu)\zeta_x\right] \tag{1}$$

where ζ_x is equal to oil pressure, and ε_y and ε_z are y and z -direction of the rubber seal normal strain. when considering the thickness of the oil film and the deformation of the vane in the y-direction, the distribution of the contact pressure of the vane is

$$p_J = \zeta_J = \frac{1}{E}\left[E(\varepsilon_y + h + \nu\varepsilon_z) + \nu(1+\nu)\zeta_x\right] \tag{2}$$



a) Rotor seal contact pressure calculation model

b) Stator contact pressure calculation model

Figure 2. Stator and rotor vane seals.

Vane Seal Lubrication Model

Calculating the one-dimensional average Reynolds equation based on the assumption that the temperature and density of the lubrication oil are constant, ie

$$\frac{\partial}{\partial x}\left(\phi_x \frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) = 6U\frac{\partial \bar{h}_T}{\partial x} + 6U\sigma\frac{\partial(\phi_s)}{\partial x} \tag{3}$$

where u , p , U and σ are lubricant viscosity,average oil film pressure, vane running speed and integrated surface roughness, respectively.

The actual film thickness is the sum of the nominal film thickness and the roughness height.

$$h_T = \begin{cases} h + \delta_T & \textit{when not in contact} \\ 0 & \textit{when touched} \end{cases} \tag{4}$$

The average film thickness is expressed as

$$\bar{h}_T = \int_{-h}^{+\infty} (h + \delta_T) f(\delta_T) d(\delta_T) \tag{5}$$

$$f_{(x)} = \frac{1}{\sqrt{2\pi\delta_T}} e^{-\frac{x}{2\delta_T^{2}}} \tag{6}$$

$H = h/\sigma$, where H is film thickness ratio, then equation (3) can be reduced to

$$\frac{\partial}{\partial x}\left(\phi_x \frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) = 6U\phi_c \frac{\partial h}{\partial x} + 6U\sigma \frac{\partial(\phi_s)}{\partial x} \tag{7}$$

Numerical Solution of Lubrication Model

Substituting $H = h/\sigma$ into equation (3) and simultaneously achieving differentialism on both sides

$$\frac{\sigma^2}{\eta}\frac{\partial^2 p}{\partial x^2} H^3\phi_x + \frac{\sigma^2}{\eta}(3H^2\cdot\phi_x + H^3\cdot\phi_x^{\ T})\frac{\partial p}{\partial x} = 6U\phi_c \frac{\partial H}{\partial x} + 6U\sigma \frac{\partial(\phi_s)}{\partial x} \tag{8}$$

where $\phi_x^{\ T} = \partial\phi_x / \partial H$ and $\phi_s^{\ T} = \partial\phi_s / \partial H$. We can simplify equation (13) as

$$\frac{\partial H}{\partial x} = \frac{\dfrac{\sigma^2}{\eta}\dfrac{\partial^2 p}{\partial x^2} H^3\phi_x}{6U(\phi_c + \phi_s^{\ T}) - \dfrac{\sigma^2}{\eta}(3H^2\cdot\phi_x + H^3\cdot\phi_x^{\ T})\dfrac{\partial p}{\partial x}} \tag{9}$$

In the case of known distributions, the oil film distribution is solved using the inverse solution method. The differential equation is used to find the partial derivative in the solution domain

$$\left.\begin{aligned} \left(\frac{\partial p}{\partial x}\right)_i &= \frac{p_{i+1} - p_{i-1}}{2\Delta x} \\ \left(\frac{\partial^2 p}{\partial x^2}\right)_i &= \frac{p_{i+1} + p_{i-1} - p_i}{(\Delta x)^2} \end{aligned}\right\} \tag{10}$$

the forward difference method and the backward difference method are used on the boundary of the solution domain

$$\left.\begin{aligned} \left(\frac{\partial p}{\partial x}\right)_1 &= \frac{p_2 - p_1}{\Delta x} \\ \left(\frac{\partial p}{\partial x}\right)_n &= \frac{p_i - p_{i-1}}{\Delta x} \end{aligned}\right\} \tag{11}$$

according to the hypothetical boundary conditions

$$\left.\begin{aligned} \frac{\partial^2 p_1}{\partial x_1^{\ 2}} &= 0 \\ \frac{\partial^2 p_n}{\partial x_n^{\ 2}} &= 0 \end{aligned}\right\} \tag{12}$$

Results Analysis

The contact pressure distribution of the vane seal is shown in Figures 3 and 4. It can be seen that the dry contact pressure of the stator seal and the rotor seal is gentle in the middle area, and the two ends are larger and the minimum pressure is greater than the cylinder oil pressure.

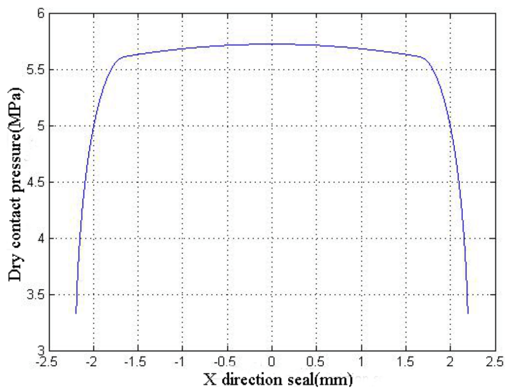


Figure 3. Rotor vane seal dry contact pressure.

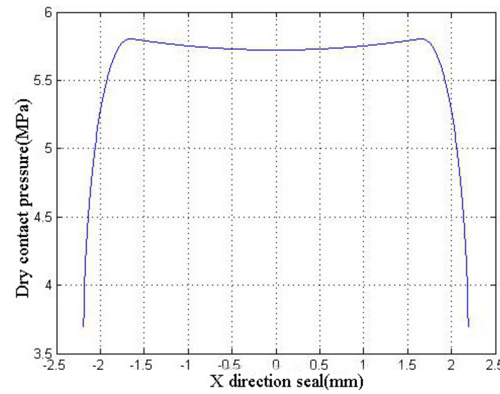


Figure 4. Stator vane seal dry contact pressure.

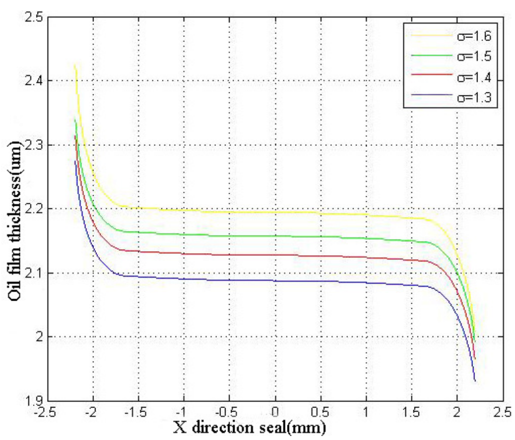


Figure 5. Rotor seal oil film thickness.

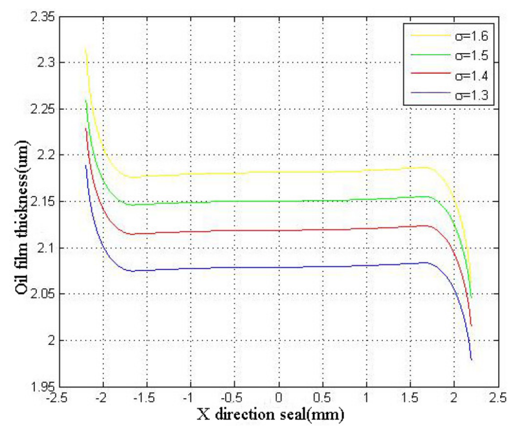


Figure 6. Stator seal oil film thickness.

In the same surface direction parameters ($\gamma = 1$), different surface roughness ($\sigma = 1.3, 1.4, 1.5, 1.6$), the rotor and stator vane seal surface oil film thickness distribution is shown in Figure 6 and 7. It can be seen that the oil film thickness curve in the middle part of the approximate line, at both ends of the larger changes.

Summary

Based on the influence of the surface morphology of the vane, considering the elastic deformation of the vane seal surface in the initial pre-compression and the fluid pressure of the lubrication oil film, a method for calculating the hydrodynamic lubrication of the vane seal surface of the hydraulic rotary vane actuators is constructed more accurately the influence of surface morphology on the lubrication was studied. The results show that the surface morphology has a great influence on the lubrication performance.

1. Based on the generalized Hooke 's law, the mathematic model of vane contact pressure was established. The formula of dry contact pressure of vane seal and the contact pressure calculation formula were given, considering the effect of oil film thickness.

2. Based on the average Reynolds equation and the G-W micro-bulge contact model, the hydrodynamic lubrication model of the surface of the hydraulic rotary vane actuators with the surface topography was established.

Acknowledgement

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References

[1] Greenwood J A, Tripp J H. The contact of two nominally flat rough surfaces[J]. Proceedings of the Institution of Mechanical Engineers, 1970, 185:625-633.

[2] Christensen H. A theory of mixed lubrication[J]. Proc. Inst. Mech. Engrs, 1969~1 970, 184(55):1013-1022.

[3] Patir N, Cheng H S. Application of average flow model to lubrication between rough sliding surfaces[J].Journal of Lubrication Technology,1979:220-229.

[4] Nikas G K, Sayles R S. Modeling and optimization of composite rectangular reciprocating seals[J].Proceedings of the Institution of Mechanical Engineers, Part J:Journal of Engineering Tribology, 2006, 220(J4):395-412.

[5] Nikas G K. Theoretical study of solid back-up rings for elastomeric seals in hydraulic actuators[J].Tribology International, 2004, 37(9), 689–699.

[6] C J Hooke. Elastohydrodynamic Lubrication of Highly Deformed Contacts. ARCHIVE Journal of Mechanical Engineering Science, 1972, 14(14):34-48.

[7] Nikas G K. Transient elastohydrodynamic lubrication of rectangular elastomeric seals for linear hydraulic actuators[J]. Proc. Instn Mech. Engrs, Part J: J. Engineering Tribology, 2003, 217(J6):461-47.