Effect of Roughness on Lubrication Performance of Bionic Micro-Textured Surface for Hydraulic Cylinder

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Abstract. For the gap seal hydraulic cylinder, the bionic rhombic micro-texture is constructed on the inner surface of the cylinder. The coupling effect of the surface roughness and micro-texture is studied. The equivalent flow method is used to solve the coupling effect between micro-texture and surface roughness. The results show that there is obvious coupling effect between surface roughness and micro-texture. There is the optimal combination of the rhombic micro-texture morphology and roughness on the inner surface of the cylinder, which makes the inner surface of the cylinder have the lowest friction coefficient and the best lubrication performance. This has a positive effect on the realization of energy-saving and green technology of hydraulic cylinders.

1. Introduction

Micro-texture plays a significant role in improving interfacial performance of mechanical components, such as piston rings [1], bearings [2], mechanical seals [3], and cutting tools [4]. However, research in the field of hydraulics has just begun. It is the key to realize energy-saving and greening of hydraulic system to reduce the friction and wear and improve the working efficiency of hydraulic system by applying bionic micro-texture technology to hydraulic system components. Mao and her co-workers [5] established an analytic mathematic model to describe a uniformly distributed micro-texture cavitation and the simulation results suggested that the proposed analytic model was valid and feasible for use in predicting the cavitation performance of various surface textures. Zhang Ying and her group [6] constructed a variety of bionic micro-textures on the surface of the piston, such as circular, cyndrical, elliptical, and square, etc., and analysed the effects of different micro-textures on the dynamic lubrication of hydraulic cylinder friction pair. Chen and her co-workers [7] developed a diamond-like micro-texture on the surface of hydraulic cylinder and optimised to determine the optimal texture about 9 times improvement for the bearing capacity and a quarter reductions for the friction coefficient.

The above researches have established the bionic micro-texture on the smooth inner surface of the cylinder or the outer surface of the piston, without considering the roughness on the parts surface. Moreover, when the average gap between the friction pairs of the hydraulic cylinder is on the same order of magnitude as its own surface roughness, the coupling effect of roughness and bionic micro-texture cannot be ignored [8].
Here, for the gap-sealed hydraulic cylinder, the bionic micro-texture is constructed on the inner surface of the cylinder, and the roughness on the friction pairs (on the inner surface of the cylinder and the outer surface of the piston) is taken into account. The equivalent flow method is used to establish the average Reynolds equation which is solved by using the five-point differential and super-relaxation iterative method to calculate the coupling effect of different rough surface and bionic micro-texture on the lubrication performance of the cylinder surface.

2. **Mathematical model**

2.1. **Texturing cylinder**

According to the working condition of the hydraulic cylinder, the gap between the cylinder and the piston is much smaller than the radius of the cylinder, just a few micrometers. So the cylinder can be unrolled into a plane for analysis and the diamond-like micro-texture is applied on the inner surface of the cylinder (Fig.1). To simplify calculations, a single control unit is analyzed and their structure parameters as follow in Table 1.

![Figure 1. Texture configuration (a) the friction pair; (b) the shape and distribution and (c) the structure parameters of micro-texture](image)

**Table 1.** The structure parameters of micro-texture

| Structure parameters                       | Symbol | Values(mm) |
|-------------------------------------------|--------|------------|
| Length of single control unit             | \(L_x \times L_y\) | 1.0 \times 1.0 |
| The diagonal length of the diamond texture| \(2a \times 2b\)  | 1.0 \times 0.8 |
| Depth of the diamond texture              | \(h_p\)     | 0.006      |
| Texture control lines                     | \(L_1,L_2,L_3,L_4\) | None  |
| Textured area                             | \(\Omega_0\) | None        |
| Non-textured area                         | \(\Omega_1\) | None        |

2.2. **Roughness contact model**

When considering the random roughness on the surface of the friction pairs, the roughness on the inner surface of the cylinder and the outer surface of the piston is set to be \(\sigma_1\) and \(\sigma_2\) respectively. The gap between friction pairs is filled with the Newtonian oil. If the moving speed of the piston is \(U\) and the minimum gap thickness of the gap is \(h_0\), considering their direct contact, the roughness contact model is established as shown in Fig.2.
Thus, the gap between the friction pairs is the thickness of the oil film in the flow field. When the roughness is not considered, it can be seen from Fig. 1 that the thickness of the oil film can be expressed as

\[ h = \begin{cases} h_0 + h_p, & x, y \in \Omega_0 \\ h_0, & x, y \in \Omega_1 \end{cases} \]  

(1)

Where: \( \Omega_1 \) represents a non-diamond micro-textured region, and \( \Omega_0 \) represents a rhombic micro-textured region, consisting of 4 straight lines, namely:

\[
\begin{align*}
L_1 : & -b(x - \frac{L_x}{2}) + a(y - \frac{L_y}{2}) = ab \\
L_2 : & b(x - \frac{L_x}{2}) + a(y - \frac{L_y}{2}) = ab \\
L_3 : & b(x - \frac{L_x}{2}) - a(y - \frac{L_y}{2}) = ab \\
L_4 : & -b(x - \frac{L_x}{2}) - a(y - \frac{L_y}{2}) = ab 
\end{align*}
\]  

(2)

When considering the roughness, it can be seen from Fig. 2 that the actual oil film thickness becomes:

\[ h_f = h + \sigma_1 + \sigma_2 \]  

(3)

2.3. Governing equation

Assuming that the fluid can be characterized as Newtonian and slip phenomenon is not considered, the modified equation is given as follows by introducing the above parameters [9]:

\[
\frac{\partial}{\partial x}(\phi, \frac{h^2}{\eta} \frac{\partial \phi}{\partial x}) + \frac{\partial}{\partial y}(\phi, \frac{h^2}{\eta} \frac{\partial \phi}{\partial y}) = 6U(\phi, \frac{\partial h}{\partial x} + \sigma \frac{\partial \phi_s}{\partial x})
\]  

(4)

Where \( \phi_x \) and \( \phi_y \) are the pressure flow factors along the \( x \) and \( y \) directions, \( \phi_s \) is the surface contact factor and \( \phi_r \) is the shear flow factor. Their values can be found in reference [10]. The term \( \sigma \) is the comprehensive roughness and can be expressed by

\[ \sigma = \sqrt{\sigma_1^2 + \sigma_2^2} \]  

(5)
Meanwhile, the film thickness ratio $H$ and surface direction parameters $\gamma$ are introduced to describe the effect of roughness. They can be given by

$$H = \frac{h}{\sigma}, \quad \gamma = \frac{H_{0.5x}}{H_{0.5y}}$$  \hspace{1cm} (6)

When the film thickness ratio is greater than or equal to 3, it means that the thickness of the lubricating oil film is much larger than the surface roughness value, the two surfaces of the friction pair cannot be contacted, and the flow field area is in a completely fluid lubrication state. In the contrast, when it is less than 3, it means the flow field area is in a mixed lubrication state and there may be contact on the surface, and the coupling effect cannot be ignored at this time.

In the other hand, when the surface direction parameter is greater than 1, it appears as a horizontal stripe. At this time, the resistance of the rough peak to the pressure flow is small, and only a small lateral flow is allowed. This is because the surface roughness peak is the same as the flow direction of the fluid; when the surface direction parameter is equal to 1 and there is no directionality, the rough peak has neither hindered nor promoted the flow of the oil; when the surface direction parameter is less than 1, it appears as vertical stripes, and the flow of the rough peak at this time acts as a hindrance because the lateral flow is increased as the surface direction parameter is reduced, so that the main flow rate is reduced.

Finally, according to the oil supply condition of the friction pair and the convergence-diverging lubrication property of the diamond micro-texture, the inlet and outlet boundaries of the gap oil film are determined, and the Reynolds cavitation boundary conditions are selected:

$$\left\{ \begin{array}{l}
p = (x = 0, y) = (x = L_x, y) = p_0 \\
p = (x, y = 0) = (x, y = L_y) = p_0 
\end{array} \right.$$ 

3. Simulation and discussions

3.1. Solution of Lubrication Performance Parameters on the Cylinder Surface

When the gap flow field is in the mixed lubrication state, the oil film bearing capacity $W$ of the friction pair surface is composed of two parts: the micro-texture surface oil film bearing capacity $W_1$ and the surface roughness peak oil film bearing capacity $W_2$. The micro-textured oil film bearing capacity $W_1$ can be obtained by integrating the average oil film pressure in the gap flow field region and the surface roughness peak oil film bearing capacity $W_2$ can be solved by Greenwood model [11].

When the roughness is included, the frictional force is also composed of two parts: the viscous sheer force of the lubricating oil film and the frictional force when the surface rough peak is in contact. They can be obtained by

$$F_f = -\int_{0}^{L_x} \int_{0}^{L_y} \left[ \frac{\eta U}{h} (\varphi_f + \varphi_s) + \mu_f p_s \right] dx dy$$ 

Where: $\eta$ is the dynamic viscosity of the hydraulic oil; $\mu_f$ is the friction coefficient between the rough peaks of the surface, which is assumed to be 0.1; $\varphi_f$ and $\varphi_s$ are the shear force factors, their values can also be referred to [10].

The coefficient of friction $\mu$ is then the ratio of friction to oil film bearing capacity. These lubrication performance parameters are solved by Matlab and the working parameters are shown in Table 2.
Table 2. Simulation parameters

| Simulation Parameters | Values |
|-----------------------|--------|
| Hydraulic oil dynamic viscosity $\eta$ (pa s) | 0.04686 |
| Hydraulic oil density (g/cm3) | 0.897 |
| Velocity of the moving piston (m/s) | 0.5 |
| Atmospheric pressure $p_0$ (pa) | 101325 |
| Minimum oil film thickness $h_0$ (μm) | 10-50 |

3.2. Simulation and analysis

When the roughness of the cylinder surface changes, the contour distribution of the rough surface with different roughness values is also follow changes, as shown in Fig.3.

![Figure 3. Contour map of oil film pressure on the surface of friction pair under different roughness](image)

(a) $\sigma=0.1$  
(b) $\sigma=0.2$  
(c) $\sigma=0.4$  
(d) $\sigma=0.8$  
(e) $\sigma=1.6$  
(f) $\sigma=2.4$

It can be concluded that the oil film pressure contour distribution on the cylinder surface with different roughness is basically the same, which is because the oil film pressure distribution area and distribution law are basically the same due to the same rhombic micro-texture geometry parameters. Their maximum pressures increases with the roughness.

Under the combined action of surface roughness and surface orientation parameters, the friction properties of the cylinder surface changed significantly. When the surface roughness peaks appear as horizontal stripes (surface parameters greater than 1), the oil film bearing capacity is significantly larger than that of the rough peaks (longitudinal parameters less than 1), as shown in Fig.4a. At the same time, it can be found that with the increase of surface roughness, the friction coefficient of the friction pair surface firstly decreases and then increases when the rough peaks are different, that is, there is an optimal roughness value, which minimizes the coefficient of friction of the textured surface. This indicates that the coupling effect of the diamond micro-texture and the surface roughness is the strongest when the surface roughness value is about 0.8 μm, as shown in Fig.4b.
Figure 4. Effect of different directional parameters on friction coefficient: (a) effect of different roughness on the bearing capacity and (b) effect of different roughness on the friction coefficient

4. Conclusion

It can be summarized through simulation analysis:

i) The method of equivalent flow model shows that the surface roughness and the surface texture have a very obvious coupling effect on the friction pair surface in the mixed lubrication area. Even if the surface roughness of the parts is different, the optimal roughness value on the frictional surface can be obtained.

ii) The effect of hydrodynamic lubrication is not only related to the texture parameters, but also related to the roughness of the textured surface. In order to enhance the wedge effect and avoid the countercurrent phenomenon, the roughness value will be about 0.8, and then the film bearing capacity and friction coefficient are both in the best value.

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