Roughness optimisation of textured surface for the gap seal hydraulic cylinder

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Abstract: The circular micro-texture is constructed on the inner surface of the gap seal hydraulic cylinders. The coupling relationship of the surface roughness and micro-texture is studied and solved by the equivalent flow method. At the same time, the tribological experiment is carried out with 45 steel materials commonly used as hydraulic cylinders, and the testing results are compared. The results show that the roughness has a very significant effect on the frictional properties of the annular textured surface. The coupling effect is generated and makes the friction coefficient first decrease and then increase. There are optimal roughness and optimal gap which make the further improvement of the frictional properties on the cylinder surface. This helps to improve the response frequency of the hydraulic cylinder, thereby improving the hydraulic system efficiency. Also, it has positive effects on the improvement of the working efficiency in the hydraulic system.

1 Introduction

In the servo-hydraulic system, the low-frequency response of the hydraulic cylinder severely limits the hydraulic system, and the key factor is the surface friction between hydraulic elements, that is the cylinder and the piston. In recent years, the development of micro-texture lubrication technology provides a new way of thinking to solve the problem. Mao et al. [1] established an analytic mathematical model to study the modelling and optimisation of cavitation on a textured cylinder surface coupled with the wedge effect and obtained the results that the friction and the support force of the friction pair were improved. Ying et al. [2] compared the effects of four micro-textures (circles, ellipses, diamonds and hexagons) on the friction and concluded that the shapes of the best lubrication performance were the circles. Chen et al. [3] developed a diamond-like micro-texture on the inner surface of the hydraulic cylinder and optimised to determine the optimal texture about 9 times improvement for the bearing capacity and quarter reduction for the friction coefficient. It is possible to effectively reduce the friction between the piston and the inner surface of the cylinder by processing the geometrical shape of the micro-structure, so as to enhance the dynamic pressure effect, improve the bearing capacity of the oil film and reduce furrow friction, thereby enhancing the piston surface lubrication performance. These references mentioned-above have demonstrated the improvement but they do not consider the influence of roughness of hydraulic cylinder components.

In order to further understand the behaviour of roughness in hydrodynamic lubrication for hydraulic cylinders, it is necessary to study the lubrication and anti-friction properties under the conditions of full fluid in the presence of roughness by establishing models and optimising the roughness on the textured surface, and then analyse the effect mechanism with roughness coupling with the texture technology, so as to achieve the active control of roughness for the gap seal hydraulic cylinder, thereby improving the lubrication performance and enhancing the frequency response ability.

2 Background

2.1 Geometric texture

The hydraulic cylinder and the piston are a pair of moving friction whose relative movement speed is \( U \) (Fig. 1a). The surface of the cylinder is textured with dimples (Fig. 1b). The parameters of a single dimple are shown (Fig. 1c). The length and width of the single texture unit are denoted as \( L_x \times L_y \), and the dimple radius and depth are expressed as \( r_p \) and \( h_p \), respectively. Considering the ease of texture processing and the best performance, the cylindrical texture will be analysed in this work. Since the film thickness and the radial size of the micro-structure relative to the piston size are small, the oil film radius of curvature between the two can be ignored and the cylinder surface can be developed into a plane [4, 5].

2.2 Roughness coupling model

The friction pair is filled with lubricant and remains relatively moving at a speed of \( U \) (Fig. 2). Assume that the friction pair ideal gap is \( h_0 \) and the depth of a single dimple is still \( h_p \). The roughness of the inner surface of the hydraulic cylinder is set to \( \sigma_1 \) and the...
roughness of the outer surface of the piston is \( \sigma_2 \) when considering the random distribution of the friction pair. Meanwhile, due to the non-uniformity of the surface roughness, a solid surface contact area where the cylinder and the piston surface are in direct contact may occur. Therefore, the roughness coupling model can be established and the coupling effect of dimples and surface roughness will be studied in this work.

The combined effect of the friction pair roughness is expressed as a comprehensive roughness defined as \( \sigma (\sigma' = \sigma_1' + \sigma_2') \). In theoretical calculations, it is usually assumed that one is a smooth surface and the other is a synthetic rough surface. In addition, the coefficient of friction primarily depends on the surface texture of the harder material surface according to Menezes and Kaial [6]. The piston is a brass and the cylinder tube is a carbon steel in this work. Then it can be learned that \( \sigma = \sigma_1 + \sigma_2 = 0 \).

3 Mathematical models

In order to analyse the coupling effect of pit textures and the surface roughness, the average flow model will be introduced which was proposed by Patir and Cheng [7]. They studied the effect of roughness on lubrication by introducing some optional statistical parameters.

3.1 Governing equations

The lubrication between the friction pairs can be characterised as the Newtonian without considering the slip phenomenon, and the governing equation can be modified as follows according to [8, 9]

\[
\frac{\partial}{\partial x} \left( \phi_s \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_s \frac{h^3}{12\eta} \frac{\partial p}{\partial y} \right) = \phi_s \left( U \frac{\partial h}{\partial x} \frac{\partial h}{\partial x} + \frac{\partial h}{\partial y} \frac{\partial h}{\partial y} \right)
\]

(1)

where the terms \( \phi_s \) and \( \phi_t \) are the factors of the pressure-flow along the \( x \)- and \( y \)-directions, \( \phi_s \) is the surface contact factor and \( \phi_t \) is the shear flow factor. These parameters mentioned above are the optional statistical parameters and their values can be found in [8]. Meanwhile, the symbol \( \bar{p} \) is the average pressure of lubricant and \( U \) is the piston velocity in (1). The following introduction of parameters and derivation of mathematical models are based on this model.

3.1.1 Introduction of parameters: Two condition parameters are introduced to describe the actual state of the friction surface. One condition parameter is the film thickness ratio (defined as \( \lambda \)).

Its value is equal to the ratio of nominal oil film thickness (\( h \)) and comprehensive roughness (\( \sigma \)). Taking a cylindrical micro-texture unit as the research object, the nominal film thickness can be expressed as

\[
h = \begin{cases} h_0 + h_p + \sigma \quad x^2 + y^2 \leq r_p^2, \\ h_0 + \sigma \quad x^2 + y^2 > r_p^2. \end{cases}
\]

(2)

There are two cases of oil film thickness of the friction pair. One is the oil film thickness in the pit texture region and can be expressed by \( x^2 + y^2 \leq r_p^2 \). The other is the oil film thickness outside the pit texture region and can be expressed by \( x^2 + y^2 > r_p^2 \). Finally, film thickness ratio (\( \lambda \)) can be given as

\[
\lambda = \frac{h}{\sigma}
\]

(3)

There are two cases here. One is that the flow area is in a completely fluid lubrication status. At this time, the thickness of the lubrication oil film is much larger than the surface roughness value and \( \lambda \geq 3 \). The other case is \( \lambda < 3 \), which means the flow area is in a mixed lubrication state and the coupling effect should be considered.

(ii) Surface direction parameter (defined as \( \gamma \))

Mechanical processing will form a different rough surface resulting in a certain directional roughness, which can be described by the surface direction parameter. Its value is related to the film thickness ratio, and it can be take seven values, which is \( 1/9, 1/6, 1/3, 1, 3, 6 \) and \( 9 \) according to Patir and Cheng [7] and Patir [10].

When \( \gamma = 1 \), the roughness topography shows horizontal stripes. When \( \gamma > 1 \), the roughness topography shows longitudinal stripes.

3.1.2 Dimensionless equation: The parameters in (1) can be dimensionless as follows:

\[
\begin{aligned}
X &= \frac{x}{\lambda}, & Y &= \frac{y}{\lambda}, \\
H &= \frac{h}{h_0}, & P &= \frac{\bar{p}}{h_0\eta},
\end{aligned}
\]

(4)

Substituting these parameters in (4) into (1), then it can be obtained as

\[
\frac{\partial}{\partial X} \left( \phi_s H^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Y} \left( \phi_s H^3 \frac{\partial P}{\partial Y} \right) = \Lambda_1 \frac{\partial H}{\partial X} + \Lambda_2 \frac{\partial \phi_t}{\partial X}.
\]

(5)

The terms \( \Lambda_1 \) and \( \Lambda_2 \) are the coefficients after simplification, which can be derived as

\[
\Lambda_1 = \frac{6U\eta L_a}{h_0 h_p}, \quad \Lambda_2 = \frac{6U\eta L_a}{h_0 h_p}.
\]

(6)

3.1.3 Boundary conditions: The hydrodynamic pressure of oil film is predicted on the basis of the Reynolds cavitation boundary conditions. The starting point of the ruptured oil film meets the low pressure condition, so the oil film pressure and the normal pressure gradient are also equal to zero in the cavitation area. Then it can be described as

\[
\begin{aligned}
P(X = 0, Y) &= P(X = 1, Y) = 1, \\
P(X = 0, Y) &= P(X, Y = 0) = P(X, Y) = 1, \\
\frac{\partial P}{\partial Y}(X, Y = 0) &= \frac{\partial P}{\partial X}(X, Y = 1).
\end{aligned}
\]

(7)

3.2 Calculation of friction coefficient

The friction performance of the textured surface coupling roughness studied in this paper is described by the friction coefficient (defined as \( \mu \)), which is equal to the ratio of friction

\[
\mu = \frac{\text{Force of friction}}{\text{Normal force}}.
\]

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the roughness peak bearing capacity calculated by force will be equal to the sum of the texture bearing capacity and similar to normal force. It is also divided into two parts: viscous parts: the bearing capacity of the textured surface \((F_{Nt})\) and the bearing capacity of the rough peak \((F_{N2})\). They can be calculated by (8) and (9)

\[
F_{Nt} = \int_0^{L_x} \int_0^{L_y} \bar{p} \, dx \, dy \quad \text{(8)}
\]

\[
F_{N2} = \int_0^{L_x} \int_0^{L_y} \bar{p}_d \, dx \, dy \quad \text{(9)}
\]

where \(E^*\) is the equivalent elastic modulus of cylinder and piston, which is equal to 120 G Pa. \(F_{2.5}(\lambda)\) is the statistical function of the oil film ratio. If \(\lambda > 4\), \(F_{2.5}(\lambda)\) will equal to 0, or it will be calculated by \(F_{2.5}(\lambda) = 4.3058 \times 10^{-5}(4-\lambda)^6\). Finally, the normal force will be equal to the sum of the texture bearing capacity and the roughness peak bearing capacity

\[
F_N = F_{Nt} + F_{N2} \quad \text{(10)}
\]

3.2.2 Friction force \((F_f)\): The calculation of friction force is similar to normal force. It is also divided into two parts: viscous shear and asperity friction

\[
F_f = - \int_0^{L_x} \int_0^{L_y} \left[ \frac{U}{R} (\phi_1 + \phi_0) + \mu_f \bar{p}_d \right] \, dx \, dy \quad \text{(11)}
\]

where the term \(\mu_f\) is the coefficient of friction between rough peaks and \(\mu_f = 0.1\). The terms \(\phi_1\) and \(\phi_0\) are the friction-induced flow factors, which can be solved by Greenwood and Tripp [11].

3.3 Simulation

The coefficient between two friction surfaces can be solved by (10) and (11) in Fig. 3, which is given as \(\mu = F_f/F_N\). In the actual working, the clearance will lead to increased leakage if too large, in the contrast the deformation lip may contact with the cylinder surface causing an increase in friction if the gap is too small. So the initial gap between them is from 10 to 50 μm. Meanwhile, due to the best depth ratio of 0.5–0.75 and the best area occupancy ratio of 0.6–0.8 [12, 13], the pit radius and the depth are set to 250 and 40 μm, respectively.

The friction coefficient of the textured surface will be calculated and the initial simulation parameters are shown in Table 1. The solution and simulation of the oil film pressure are implemented using a MATLAB.

### 4 Results and discussion

According to the design parameters by Bo et al. [14], when the hydraulic cylinder is subjected to the load of 0–20 Mpa, the maximum deformation of the lip is 7 μm. On account of installation errors, the ideal gap is 25 μm and the hydraulic cylinder leakage is the least at this time. Meanwhile, the film thickness ratio \((\lambda)\) will be greater than 3. Hence, the cylinder and piston are in fully fluid lubricated. The simulation results are shown in Fig. 4.

#### 4.1 Effects of the surface direction parameter

The surface direction parameter is used to describe the machined surface roughness direction. When the formation direction of the roughness peak is in the same direction as the movement of the fluid, the fluid resistance is the smallest, and the oil film support and friction coefficient are the largest (Fig. 5a).
case, when it is $\geq 1$, the friction coefficient of the textured surface first decreases and then increases with the increase of roughness. There is an optimum roughness value in this process that minimizes the coefficient between the two friction surfaces. From the simulation results, we can find out that the optimal roughness value is about 2.5. These can be discussed in two cases:

(i) **When the comprehensive roughness is $<2.5$ ($\sigma < 2.5$):** For non-textured surfaces, the comprehensive roughness can be regarded as micro converging wedges so that a plurality of roughness could act as a set of micro bearings [15]. If the comprehensive roughness is smaller, the formation of the converging area will be smaller, and which will result in the reduced lifting force and increased friction coefficient.

(ii) **When the comprehensive roughness is $>2.5$ ($\sigma > 2.5$):** The friction performance is reduced because of the reflow phenomenon occurred at deeper troughs [16]. The presence of counter-current weakens the wedge effect in the convergent region (Fig. 6). Therefore, the counter current area and the friction coefficient gradually increase with the trough depth.

5 Verification and optimisation

5.1 Analysis of the gap between cylinder and piston

According to the standard in China (GB/T 1880.1-2009) [17, 18], the fit tolerance of the cylinder and piston is $H_6/f_5$ for the gap seal hydraulic cylinder. When the diameters of the cylinder and piston are $40 \text{ mm} (\phi 40H_6/50f_5)$, the maximum gap is $52 \mu m$ and the minimum is $25 \mu m$ [19, 20]. The actual assembly error and piston eccentricity will make the gap with the change, so the initial thickness of the simulation parameters for the value of $10-50 \mu m$. Meanwhile, the roughness of the piston is easy to control because of its outer cylindrical surface. Therefore, the accuracy of the outer surface of the piston can be taken higher than the inner surface of the cylinder.

5.2 Preparation of samples

Medium carbon steel is commonly used for hydraulic cylinder material and so No. 45 steel is made as the test material. The preparation of samples is divided into two processes. One is the CNC machining of the samples and the other is laser processing of sample surfaces. A round sample with a diameter of $70 \text{ mm} (\varphi 70)$ was obtained by CNC machining. Polishing is also required for highly accurate samples. During the preparation of the substrate, it is necessary to continuously measure the roughness of the samples. The subsequent laser processing process is shown in Fig. 7.

The circular dimple texture is processed on the surface of the sample by laser, and the texture morphology parameters are the same as the simulation parameters. Finally, four samples with different roughness are prepared to prove the best friction performance by laser processing (Fig. 8).

5.3 Measurement and testing

The roughness of the four samples is measured by 3D profilometer to obtain their texture parameters. The No. 2 sample is shown in Fig. 9.

The friction performance parameters of the test pieces are tested on a rheometer with the same parameters, which can be clearly seen from Fig. 10. The clamping container is placed on the rheometer base and the sample is fixed in the clamping container by a locating pin. The surface of the sample is filled with hydraulic oil. The rheometer's rotating shaft simulates the movement of the piston, and the sample simulates a stationary hydraulic cylinder. The gap between the upper plate and the sample simulates the gap between the piston and the cylinder.

The test results are shown in Fig. 11. It can be found that the bearing capacity and friction coefficient of the third sample are the best. This is basically consistent with the simulation results analysed above. It is further explained that when the comprehensive roughness of the surface of the friction pair is 2.5
μm, the friction resistance of the friction pair is optimal. That is because the counter-current phenomenon is effectively avoided. Thus, the roughness values of the inner surface of the hydraulic cylinder and the outer surface of the piston can be taken to be 2.0 and 1.6 μm, respectively.

Meanwhile, the friction performance can be obtained that is little affected by the gap between the cylinder and piston. Too much clearance will increase the internal leakage, otherwise will lead to increased friction. In addition, the size of the gap affects the thickness of the oil film, which affects the wedge effect. The proper clearance is critical to the anti-friction properties of the friction pair. Therefore we can still find the optimal gap of 20 μm, which is very advantageous to suppress leakage for gap seal hydraulic cylinders.

6 Conclusions

The results in simulation and experiment suggest that the coupling effect of surface roughness and textures on the hydrodynamic lubrication can not be neglected and the following conclusions can be drawn from the study:

(i) The hydrodynamic lubrication effect is not only determined on the texture parameters but also related to the textured surface roughness. In order to enhance the hydrodynamic lubrication effect and avoid the counter-current phenomenon, the comprehension roughness value will be about 2.5 μm. The oil film support and friction coefficient are both in the best value at this time. Moreover, the roughness values of the inner surface of the hydraulic cylinder and the outer surface of the piston can be taken as 2.0 and 1.6 μm, respectively. Compared with the existing hydraulic cylinder inner surface of 0.8 μm and the piston outer surface of 0.4 μm, this will reduce many manufacturing costs.

(ii) Clearance between the friction pair not only has a great influence on the frictional performance of the rough surface but also determines the internal leakage of the gap seal cylinder. We
can conclude that friction performance is the best when the gap is 20 μm. This has an important guiding role in the manufacture and assembly process of hydraulic cylinders and pistons.

(iii) The equivalent flow method is used to study the coupling effect of roughness, which is equivalent to the influence of the flow factor, so there is no need to consider the random form of roughness.

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8 References

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