Eccentric correction of piston based on bionic micro-texture technology for the gap seal hydraulic cylinder

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The eccentricity of the piston in the continuous casting machine cannot be ignored, which not only increases the internal leakage of the hydraulic cylinder, but also increases the friction on the surface of the friction pair, and even has serious consequences such as deformation failure. In this work, the biomimetic micro-texture technology is used to construct the microtextures on the surface of the cylinder so that it produces a continuous centring force during operation, and the problem of piston eccentricity is corrected. Moreover, the influence of hydraulic cylinder operating parameters and biomimetic microtexture parameters on the dynamic pressure support force is also analysed. This has a very positive effect on improving the response frequency of the hydraulic cylinder and the working efficiency of the hydraulic system.

1. Introduction: The gap sealing principle of the hydraulic cylinder is to use a small gap to form the liquid flow resistance, the tension between the oil and the mating surface and the molecular force to achieve the seal, which greatly reduces the frictional resistance of the friction pair [1]. When there is a certain gap between the moving pairs and there is a pressure difference between the two ends of the slit, a gap flow, which is leakage and inevitably formed. However, due to the presence of gaps, the gap sealing effect is poor when the working pressure of the hydraulic cylinder increases. Moreover, because of machining and installation errors, the piston will be eccentric, which will not only increase leakage but also result in the direct contact between piston and cylinder and increase friction [2, 3]. Finally, these will cause the deformation and failure of the piston (Fig. 1). Therefore, the excessive eccentricity of the piston will reduce the response speed of the hydraulic cylinder, thereby reducing the operating efficiency of the hydraulic system.

The bionic texture technology that has emerged in recent years provides new ideas for solving piston eccentricity and reducing friction problems [4]. That is, a series of regular grooves or bulges are machined on the surface of the friction pair to store oil to form a dynamic pressure lubricant film, which has been applied in mechanical seals, bearings, piston rings and cutting tools and the like.

It can build the oil film dynamic pressure lubrication, reduce the friction between the friction pair, and provide effective centring force by improving the structure on the inner surface of the cylinder. This method of using dynamic pressure to provide centring force can well solve the problem of piston eccentricity.

2. Influence of eccentricity on leakage in the hydraulic cylinder: Assume that the gap between the cylinder and the piston is h_0, the length of the piston is L, and the radius of the piston and cylinder are, respectively, r_1 and r_2 when the piston is not eccentric. Under eccentric conditions, the eccentricity is e and the gap is h at any angle θ (Fig. 2).

When the fluid in the seal gap makes a constant laminar flow in the x-axial direction and because the gap is small (i.e. $r_1 \simeq r_2 = r$), the flow between the annular gaps corresponding to the small arc $r\mathrm{d}\theta$ can be approximated as a parallel plate gap flow [5]. At this time, the annular gap h can be approximated as

$$h \simeq h_0 - e\cos\theta = h_0(1 - \varepsilon\cos\theta) \tag{1}$$

where ε is the relative eccentricity and equal to e/h_0. When the velocity of the piston is U, the density of oil is ρ, the kinematic viscosity of oil is η and the pressure difference across the piston is Δp, the leakage of the annular gap can be expressed as

$$\mathrm{d}q = \frac{rh^3\,\mathrm{d}\theta}{12\rho\eta L}\Delta p \tag{2}$$

Substituting (1) into (2) and integrating, (3) can be obtained

$$q = \frac{\pi r h_0^3 \Delta p}{6\rho\eta L}(1 + 1.5\varepsilon^2) \tag{3}$$

From (3), it can be seen that the leakage of the annular gap is proportional to the relative eccentricity when the other parameters are not changed. There are two extreme cases: (i) $\varepsilon = 0$, the piston and cylinder are concentric (its leakage is $(\pi r h_0^3 \Delta p)/6\rho\eta L$) and (ii) $\varepsilon = 1$ ($e = h$), at this point, the eccentricity reaches its maximum (its leakage is $2.5(\pi r h_0^3 \Delta p)/6\rho\eta L$). That is in the case of maximum eccentricity, the leakage is 2.5 times greater than in the concentric state. Moreover, in this case, direct contact occurs between the cylinder and the piston surface. This will bring serious consequences such as wear and deformation.

So the key to the problem is to control the eccentricity of the piston and can be automatically centred when the piston is eccentric. These will be discussed in the next section.

3. Influence of eccentricity on friction in the hydraulic cylinder: In actual production, the cooperation between the piston and the cylinder will inevitably produce eccentricity due to the processing and installation errors. The piston is subject to a certain lifting force when the piston is biased, which will solve this problem well.

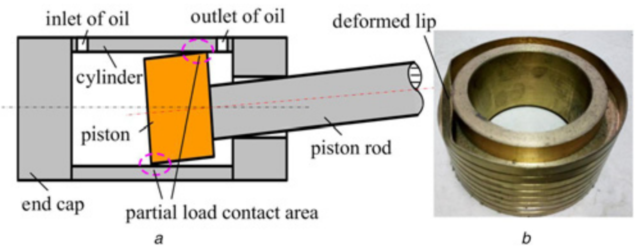

Fig. 1 *Deformation and failure of the piston*
a Eccentricity of the piston
b Deformation

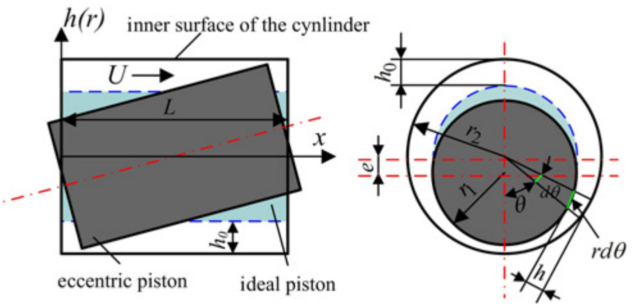

Fig. 2 *Clearances between ideal position and eccentric position*

In 1966, Hamilton *et al.* [6] found that microirregularities on the surfaces of rotary-shaft seals were capable of producing hydrodynamic pressure and created a load carrying capacity. After 30 years, Etsion and Burstein [7] investigated that an increase in load carrying capacity was obtained with hemispherical dimples on the mating surfaces in 1996. Since then, hundreds of theoretical and experimental studies have been conducted worldwide in order to improve the performance of tribological contacts.

3.1. Principle of generating hydrodynamic lift: Microsurface textures may act as lubricant reservoirs, providing lubricant to the contact in case of starved lubrication. And they lead to the additional hydrodynamic lift in cases of mixed and hydrodynamic lubrication [4].

The hydrodynamic lift will generate when there is relative motion and a viscous liquid filled between the two surfaces. The reason for support load is the occurrence of local cavitation, which can lead to an asymmetric pressure distribution over a single microtexture, as negative pressures caused the diverging region of a texture are limited by the lubricant's cavitation pressure (Fig. 3).

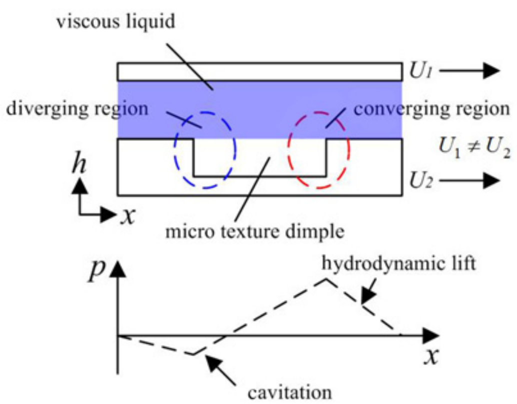

Fig. 3 *Principle of microtexture dynamic pressure generation*

Using the mechanism of hydrodynamic pressure to build a microtexture on the friction pair of the hydraulic cylinder, a certain lifting force can be generated on the surface of the cylinder barrel.

3.2. Analysis of microstructure hydrodynamic pressure on the cylinder surface: Dung beetles move freely in damp soil without being adhered and are not worn due to their special surface structure [8]. Imitating the dung beetles' surface microstructure with high wear-resistant ability, an ellipse-shaped microstructure is built on the inner surface of the cylinder barrel because reasonable elliptical texture can improve the lubrication performance of the cylinder surface according to the literature [9]. The value of the gap between the cylinder and the piston is typically a few micrometres and much smaller than the curvature radius of the cylinder, so the cylinder can be unrolled into a plane for analysis. The direction of axial movement of the piston is the x-axis and the radial direction is the y-axis. The elliptical diameters along the x-axis and y-axis directions are $2a$ and $2b$, respectively. Meanwhile, the texture depth is h_p (Fig. 4).

3.2.1. Calculation of lifting force: The hydrodynamic pressures are calculated by solving the Reynolds equation derived from the Navier–Stokes equation by assuming the classical hypotheses of lubrication theory in [10–12]; i.e. (i) the surfaces are smooth; (ii) the film thickness is small compared with the size of the other features at the contact interface; and (iii) the hydraulic components are rigid bodies. For an isothermal and incompressible lubricant, the Reynolds equation can be written as

$$\frac{\partial}{\partial x}\left(\rho h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\rho h^3 \frac{\partial p}{\partial y}\right) = 6\eta \frac{\partial}{\partial x}(U\rho h) \tag{4}$$

where p is the hydrodynamic pressure, U is the sliding velocity of the piston. The film thickness and boundary conditions can be given by

$$\begin{cases} h(x, y) = \begin{cases} h_0 & (x, y) \notin \Omega \\ h_0 + h_p & (x, y) \in \Omega \end{cases} \\ p(0, y) = p(L_x, y) = p(x, 0) = p(x, L_y) = p_0 \\ \left(\dfrac{\partial p}{\partial h}\right)_{p=p_{\text{cav}}} = 0 \end{cases} \tag{5}$$

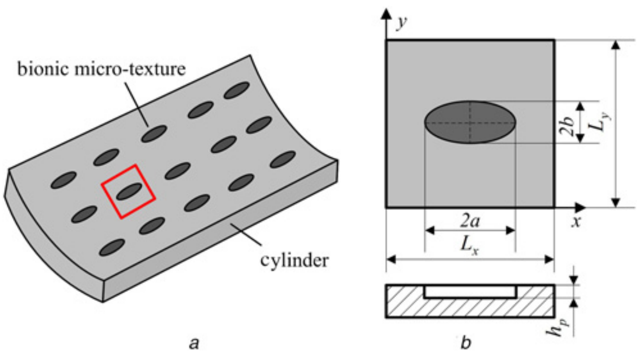

Fig. 4 *Bionic microtexture morphology and its texture parameters*
a Texture distribution on cylinder surface
b Microtexture unit parameters

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where Ω is the elliptical region and can be expressed by

$$\frac{(x-(L_x/2))^2}{a^2}+\frac{(y-(L_y/2))^2}{b^2}\le 1 \quad a\in\left[0,\ \frac{L_x}{2}\right];\ b\in\left[0,\ \frac{L_y}{2}\right] \tag{6}$$

The Reynolds cavitation boundary is described as the starting points of the oil film rupture, which must be equal to zero, and the oil film pressure and gradient of the normal pressure in the cavitation area must be simultaneously zero.

According to Wen and Huang [12], the hydrodynamic lift can be obtained by integrating the pressure (4) along the lubrication area since the forces acting on the microelements are equal in all directions. Therefore, after calculating the pressure distribution of the oil film, the friction parameters (including the bearing capacity of the lubricating film, the friction force exerted by the lubricating film and the friction coefficient) can be solved.

Over the entire range of the lubricating film, the pressure can be integrated to obtain the bearing capacity of the lubricating film (defined as F) as

$$F=\int_0^{L_y}\int_0^{L_x} p\,\mathrm{d}x\,\mathrm{d}y \tag{7}$$

Meanwhile, the friction force exerted by the lubricating film (defined as F_f) on the solid surface can be obtained by integrating the shear stress in the fluid layer, which is in contact with the surface over the entire lubricating film

$$F_f=\int_0^{L_y}\int_0^{L_x}\left(\frac{h}{2}\frac{\partial p}{\partial x}+U\frac{\eta}{h}\right)\mathrm{d}x\,\mathrm{d}y \tag{8}$$

Finally, the friction coefficient (defined as μ) on the hydraulic cylinder surface can be obtained

$$\mu=\frac{F_f}{F} \tag{9}$$

3.2.2. Calculation method and simulation parameters: The Reynolds equation is an elliptic second order PDE, which is difficult to solve analytical solution with the existing mathematical theory. The Finite Difference Method (FDM) is the most popular discretisation method used in the study of textured surfaces [13]. Therefore, the FDM is used to solve (4) in this Letter.

In the gap seal hydraulic system, the piston is running at high speed (1 m/s) under working conditions and other operating parameters are shown in Table 1.

The solution and simulation of the oil film pressure are implemented using a MATLAB based on a single microtexture.

4. Simulation and discussions: In this section, different numerical results obtained for the different gap between cylinder and piston considering the eccentricity.

Table 1 Initial values of working parameters

Working parameters	Values
oil dynamic viscosity η, Pa s	0.04678
density of oil ρ, g/cm^3	0.897
gap between cylinder and piston h_0, μm	10
gap between cylinder and piston considering eccentricity h, μm	1–10
depth of texture h_p, μm	10
standard atmospheric pressure p_0, Pa	101,325
the major and minor axis radius of the ellipse $a\times b$, mm	0.45 × 0.35
length of square control unit $L_x \times L_y$, mm	1.0 × 1.0

4.1. Effect of an annular eccentric gap on hydrodynamic pressure: When the hydraulic cylinder is operated under the conditions of the parameters shown in Table 1, the thickness of the oil film on the surface of the cylinder with the elliptical texture is shown in Fig. 5. Ellipse microtexture can produce hydrodynamic pressure effects according to the principle of generating hydrodynamic lift in Section 3.1. At this time, cavitation effects occur in the microtextured divergence region and hydrodynamic pressure effects in the convergence region (Fig. 6).

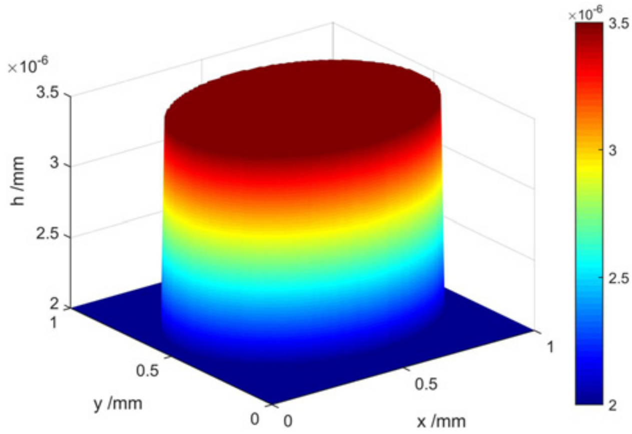

Fig. 5 *Film thickness between cylinder and piston*

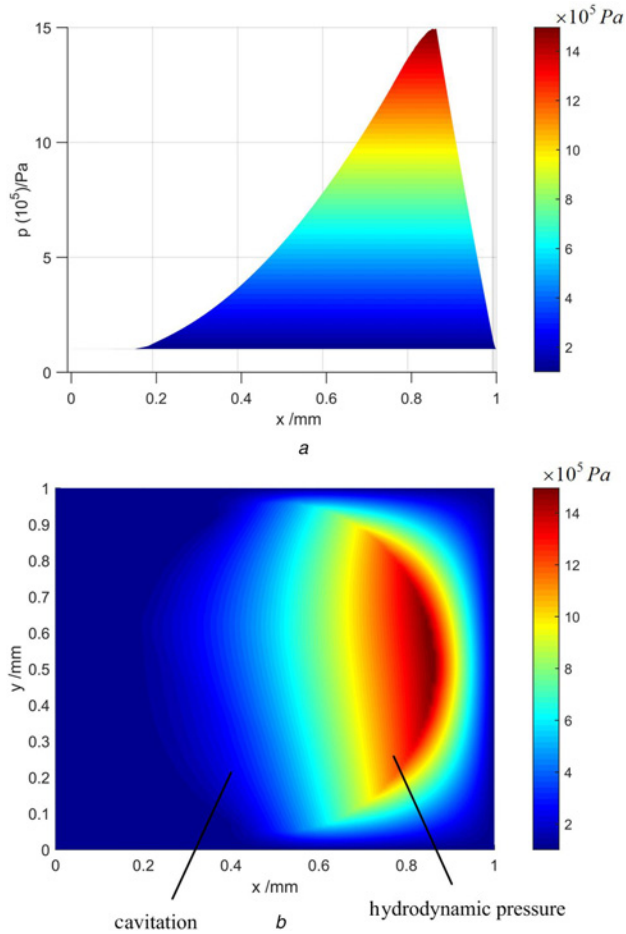

Fig. 6 *Hydrodynamic pressure distribution of a single texture*
a Maximum dynamic pressure distribution of ellipse microtexture
b Cavitation and dynamic pressure distribution

When the parameters such as the depth of texture and the speed of the piston are constant, the supporting force on the surface of the cylinder decreases as the clearance increases firstly, and then becomes stabile (Fig. 7).

As the gap between the cylinder and the piston increases, the thickness of the oil film also increases, the cavitation area in the pocket area gradually weakens, and the pressure accumulation in the downstream of the pocket area gradually declines, so the hydrodynamic pressure effect weakens, and the bearing capacity of the oil film also decreases, according to (4). When the gap is further increased, there is basically no cavitation effect in the pocket, and there is no dynamic pressure effect.

4.2. Impact of dynamic pressure on piston eccentricity: When the piston is eccentric due to objective reasons, unequal annular gaps are formed on both sides of the annular gap of the eccentric piston. The unequal gaps are defined as h' and h'', respectively (Fig. 8*a*) and it can be seen clearly that h' is larger than h''. At this time, we can assume that their corresponding oil film bearing capacities are F' and F''.

From the analysis of the previous section, we can know that F'' is larger than F', which will cause the piston to move toward the centre gradually and eliminate eccentricity eventually (Fig. 8*b*).

Then the neutral force (The force generated by the oil film to move the piston to the centre of the heart when the piston is eccentric) is defined as F_N *and* can be expressed by

$$F_N = F'' - F' \tag{10}$$

Therefore, the effects of different eccentricities on the friction coefficient on the surface of the cylinder and the neutral force of the piston can be, respectively, obtained from (9) and (10) (Fig. 9).

It can be found that as the eccentricity increases, the centring force of the piston increases significantly, which causes the piston

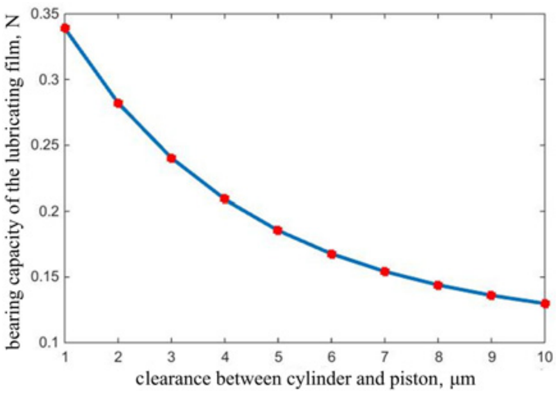

Fig. 7 *Effect of annular gap on hydrodynamic pressure*

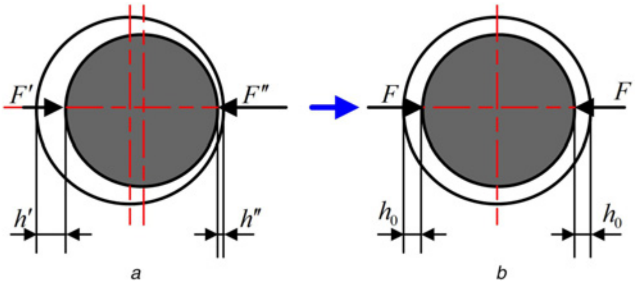

Fig. 8 *Impact of dynamic pressure on piston eccentricity*
a Analysis of bearing capacity when eccentric piston
b Analysis of bearing capacity when the piston is concentric

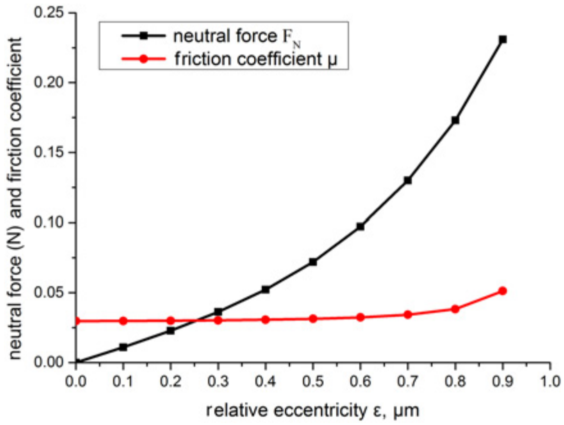

Fig. 9 *Effects of different eccentricities on the friction coefficient on the surface of the cylinder and the neutral force of the piston*

to gradually move toward the centre of the heart from Fig. 9. When the piston becomes eccentric due to its own weight again, it will be returned to the situation shown in Fig. 8*a* and then the piston will move toward the centre gradually again. So this situation will be repeated under working. Meanwhile, the friction coefficient of the cylinder surface also increases slowly as the eccentricity increases. When the eccentricity is close to 1μm, the coefficient of friction of the cylinder surface is nearly doubled. But when the eccentric state is improved due to dynamic pressure, the friction coefficient is also reduced.

4.3. Influence of piston speed on hydrodynamic pressure: For a gap-sealed hydraulic cylinder of a caster, the speed of the piston is varied throughout the operation, which has an effect on the supporting force of the cylinder surface (Fig. 10). It can be clearly obtained that as the speed of the piston increases, the bearing capacity of the cylinder surface also increases. This is because the increase in the piston speed enhances the effect of the cavitation area in the pit region and thus increases the hydrodynamic pressure.

4.4. Influence of microtexture depth on hydrodynamic pressure: Tens of thousands of living body surfaces in nature have different drag-reducing and wear-resistant properties because of their different surface microstructures. For the microtexture of the cylinder surface, it must also be affected by its special structural parameters. From Fig. 11, it can be found that the greater the depth of the texture, the lower the hydrodynamic pressure on the cylinder surface. This is similar to the effect of Section 4.2

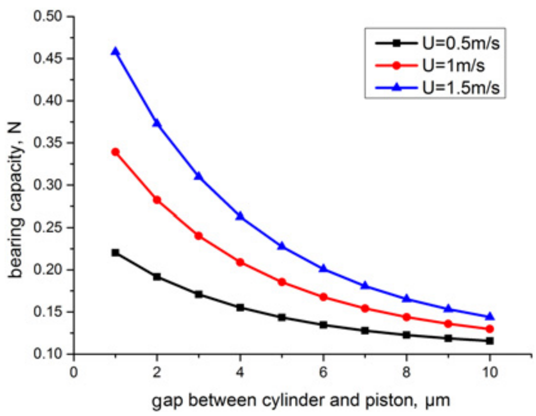

Fig. 10 *Influence of piston speed on hydrodynamic pressure*

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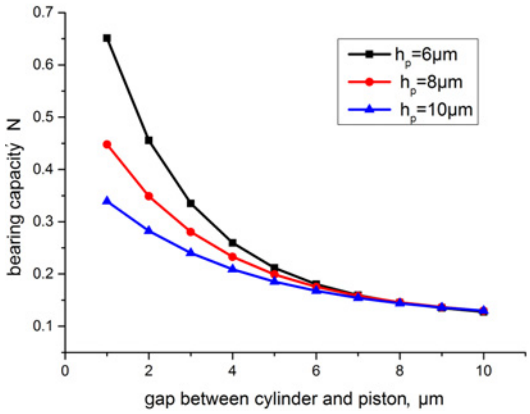

Fig. 11 *Influence of microtexture depth on hydrodynamic pressure*

above, which also is due to the fact that the increase in the thickness of the oil film leads to a reduction in the accumulation at the pit. Moreover, when the depth increases to a certain value, the lubricating oil begins to flow backward. With the increasing depth, the countercurrent area increases and weakens the bearing capacity of the texture.

5. Conclusion: The eccentricity of the piston in the continuous casting machine cannot be ignored. This not only increases the internal leakage of the hydraulic cylinder, but also increases the friction on the surface of the friction pair, and even has serious consequences such as deformation failure.

In this Letter, the biomimetic micro-texture technology is used to construct the microtextures on the surface of the cylinder so that it produces a continuous centring force during operation, and the problem of piston eccentricity is corrected. When the piston is eccentric, not only the amount of leakage increases but also the frictional resistance between the friction pairs increases. Therefore, this has a very positive effect on improving the response frequency of the hydraulic cylinder and the working efficiency of the hydraulic system.

However, on the one hand, the working parameters of the hydraulic cylinder in actual operation are changed, which will inevitably affect the dynamic pressure of the textured surface. For example, the greater the piston movement speed, the stronger the dynamic pressure on the cylinder surface. On the other hand, the bionic texture parameters of the cylinder surface, such as the shape of the texture, the depth of the texture, the arrangement and so on, also have an effect on the hydrodynamic pressure on the surface. For example, as the depth of texture increases, the dynamic pressure effect decreases. These are also important questions worthy of further study in the future.

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