WEAR PREDICTION OF PISTON/CYLINDER PAIR IN AXIAL PISTON PUMP

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ABSTRACT

The piston/cylinder pair is the key lubricating interface of axial piston pumps. It suffers from excessive wear due to the huge lateral force, especially under high output pressure. In order to achieve predictive maintenance, it is significant to detect the performance degradation of the piston/cylinder pair.

In this paper, a method to predict the wear of the piston/cylinder pair is proposed. The wear regions and corresponding wear depths under different conditions are investigated. The distributive characteristic parameters of the oil film are obtained, which can reflect the load-bearing and lubrication conditions at each region of the friction pair. Based on the oil film characteristic parameters, the most suitable wear model is chosen to calculate the wear depth, and then the entire wear profile of the piston/cylinder pair is obtained. The experimental investigation is carried out, and the results show that the accuracy of the wear regions and corresponding wear depth prediction is high. This method can be used to pump healthy management and choose the suitable working conditions of the axial piston pump.

Keywords: wear prediction, axial piston pump, validation

1. INTRODUCTION

The axial piston pump is the core component of a hydraulic system which provides the hydraulic power [1]. One of the most effective ways to enhance its power/weight ratio is to increase the pressure. The high working pressure leads to severer working conditions and a higher possibility of failure. In order to avoid the damage caused by sudden failure, performance degradation process and failure prevention have drawn the attention of more and more researchers. Statistical results show that the main failure in the axial piston pump is excessive wear of the friction pairs (piston/cylinder pair, cylinder/valve-plate pair, slipper/swash-plate pair). There are many methods for obtaining the wear condition [2]. It can be seen that most of the methods are related to the load-bearing and lubrication condition of the sliding surface. In the piston pump friction pair, the load-bearing and lubrication condition is directly reflected by oil film characteristic parameters. The oil film characteristic parameters will accordingly change after the surface wear occurs. Therefore, there is a strong interaction between the oil film

characteristics and wear of the piston/cylinder pair.

Many researchers have contributed to the calculation of oil film characteristic parameters of the piston/cylinder pair. The pressure distribution of the oil film between friction pair was numerically solved by Kolk [3] firstly, but it was simplified to a sloping plain bearing with increased pressure on the laterally loaded side edge. The influence factors of the oil film thickness changing and the oil film extrusion effect were taken into account by Fang and Shirakashi [4] to calculate the thickness field of the oil film between the piston/cylinder pair. Ivantysynova [5] established the non-isothermal flow calculation model of the oil film in the friction pair clearance based on the Reynolds Equation and the energy equation. Based on these researches, there are several friction pair oil film calculation programs, like CASPAR [6], PUMA [7] and ViSPA [8]. The oil film parameters can be calculated by the above mentioned software.

As for wear mechanism of friction pair, Archard [9] put forward adhesive wear theory in 1953. It has been improved in the next few decades, and the Archard wear model was formed. Huang Ping,

et al [10] established an abrasive wear model for lubricated sliding contacts, which is suitable to analyze friction pairs in an axial piston pump. Hsu [11] summarized 32 parameters that influence the wear of the sliding pair, including fluid temperature, thermal capacity, velocity, contaminants, materials of contact surfaces, surface processing level, lubrication condition, load and so on. Meng [1] systematically elaborated 28 types of wear models, involving over 100 relevant parameters reflected in the wear process. Ma Jiming, et al [12] presented a method to analyze the wear behavior of slipper pair based on the EHL wear model, but the effects of surface wear on the oil film characteristics were not taken into account.

In this paper, a wear prediction method of piston/cylinder friction pair is put forward. The interaction between the distributed oil film characteristics and the surface wear is taken into account. In this way, the change of piston/cylinder pair load-bearing and lubrication condition and the wear profile with operating time can be analyzed. The accelerated lifetime test of an axial piston pump was carried out and the wear profile of piston/cylinder pair was measured for the purpose of validating the simulated results.

2. WEAR PREDICTION METHOD

2.1. Overview

Distributivity

There exists a micro-motion of the piston because of the clearance between the piston and the cylinder. Therefore, the wear conditions of the cylinder bore are variable in different regions. So it is necessary to analyze the different parts of the cylinder bore surface using a distributed approach. The most convenient and effective method is to divide the surface of the cylinder bore into several nodes and analyze the load-bearing and lubrication conditions of each node. The cylindrical surface of the cylinder bore can be spread out into a plane as shown in **Figure 1**.

Figure 1: Divided cylinder bore surface

In this figure, D_C is the diameter of the cylinder bore, L_C is the effective length of the cylinder bore, δ_a and δ_c are the axial and circumferential lengths of the node, respectively.

For each node, the load-bearing and lubrication condition is described by the distributive oil film characteristics, and its wear volume is calculated separately. Finally, all the nodes are combined to fit the wear profile of the entire piston/cylinder pair.

Multiple sub-model

It is important to choose the most suitable wear calculation model under the corresponding load-bearing and lubrication condition. On the one hand, the adhesive node is considered to be formed when the hydrodynamic pressure or solid contact stress of this node is greater than a threshold value. The wear depth is suitable to be calculated by the adhesive wear model. On the other hand, the lubrication wear of cylinder bore occurred if the oil film thickness of this node is lower than the height of the rough peak. The Partial-EHL contacts sliding wear model is suitable to utilized to calculate the wear depth. The switch method of the wear calculation models is expressed as

$$\begin{cases} p_{\text{node}} \geq p_t & \text{adhesive wear model,} \\ h_{\text{node}} < h_t & \text{sliding wear model,} \end{cases} \quad (1)$$

where p_{node} is the hydrodynamic pressure of each node, h_{node} is the oil film thickness of each node, p_t is the pressure thresholds for using adhesive wear model and sliding wear model, h_t is the oil film thickness threshold to determine whether sliding wear occurs or not.

Update with operating time

With the pump operating, the profile of the cylinder bore surface will change when the wear occurs. The oil film thickness field will be affected by the wear as shown in

$$h_{\text{node}}^T = c_{\text{node}} + w_{\text{node}}^T, \quad (2)$$

where w_{node} is the wear depth of each node, c_{node} is the clearance between the piston and cylinder bore of each node without wear, T is the operating time of the pump.

The oil film thickness field is a key component for calculating the oil film characteristics. Therefore, the oil film characteristics should be updated after the wear occurs. The semi-steady solution is used to obtain the cumulative wear. The wear depth of each node w_{node} is calculated based on the oil film characteristic parameters, and then, the wear depth is utilized to recalculate the new oil film characteristic parameters.

This is a process that takes into account the interaction between wear and oil film.

2.2. Wear depth calculation

Adhesive wear

When the piston is working in the high-pressure area, the main external forces applied to the piston is shown in **Figure 2**, and the large lateral force F_L is bold. It should be noted that only main forces related to the piston lateral force including the pressure force F_p , the slipper friction force F_{SF} , the slipper supporting force F_{SS} , the axial component F_{SSz} and the lateral component F_{SSy} are presented. Other forces such as gravity and centrifugal force are relatively small, and they are not displayed for brevity. Due to the tiny clearance between the piston/cylinder pair and the large lateral force F_L which is bold in **Figure 2**, partial solid squeezing may appear between the piston and the cylinder bore as indicated in the red circle in **Figure 2**. The wear condition of the squeezed areas is suitable to be analyzed by the adhesive wear theory.

Figure 2: Main external forces applied to the piston

The adhesive nodes will be formed due to high pressure and contact stress of the contact areas when the friction pair surface slides relative to each other. It may lead to shearing fracture and surface abrasion. The wear volume V is related to

the load on the sliding surface, and it can be calculated in

$$\frac{dV}{ds} = k_s \frac{p_{\text{node}} \delta_a \delta_c}{3\sigma_s}, \quad (3)$$

where s is the sliding distance, σ_s is the compressive yield limit which is related to the rigidity of the worn material, and k_s is the wear constant which represents the probability of shearing fracture in all adhesive nodes. It can be obtained from the existing literature.

Partial-EHL contacts sliding wear

As mentioned above, the adhesive wear model is suitable to calculate the wear when the pressure on the squeezed nodes is relatively high. However, the adhesive nodes will not form when the lateral force is relatively small. The external force still lets the tilt of the piston which will cause the contact of roughness peak. The sliding wear will occur when the height of the roughness peak after viscoelastic deformation is still higher than the oil film thickness. Therefore, the expression of h_t is shown in Equation 4:

$$h_t = \delta_0 / e^{\frac{\Delta}{2u\tau}}. \quad (4)$$

The wear depth of the cylinder bore can be calculated by the partial-EHL contacts sliding wear model which is shown in Equation 5:

$$\frac{dV}{dt} = u \left(\delta_0 e^{-\Delta/2u\tau} - h_{\text{node}} \right)^2 \Delta / \delta_0, \quad (5)$$

where t is the sliding time, u is the sliding velocity, δ_0 is the height of roughness peak, Δ is the wavelength of roughness peak, τ is the delay time of cylinder bore material.

The wear depth w can be obtained by

$$w = V / \delta_a \delta_c. \quad (6)$$

2.3. Oil film characteristics calculation

It can be seen from the wear models above that the oil film characteristics are the significant parameters of wear depth calculation. As for adhesive wear, the load on the worn surface, which is closely related to the hydrodynamic pressure or the solid contact stress, is one of the key parameters for calculating the wear depth. For partial-EHL contacts sliding wear, the comparison of oil film thickness and the roughness peak height is very important. For the reasons above, the requirement for solving the oil film characteristics is put forward. A numerical model is established to calculate the distributive oil film characteristics. The oil film

characteristics can be described by Reynold Equation as shown in Equation 7:

$$\frac{\partial}{\partial x} \left(\frac{\partial p}{\partial x} \frac{h^3}{\mu} \right) + \frac{\partial}{\partial z} \left(\frac{\partial p}{\partial z} \frac{h^3}{\mu} \right) = \mathfrak{A} \omega_p \frac{D_c}{2} \frac{\partial h}{\partial x} + u \frac{\partial h}{\partial z} + 2 \frac{\partial h}{\partial t} \quad (7)$$

where p is the hydrodynamic pressure, h is the oil film thickness, μ is the viscosity of the oil, and ω_p is the speed of piston spin.

The first two items on the right side of Equation 7 represent the dynamic pressure effects of the wedge-shaped oil film pair. The third item represents the squeezing effect of oil film caused by the micro-radial motion of the piston. As a partial differential equation with diffusion term and source term, it cannot be solved directly. Therefore, the finite volume method is used to discretize Equation 7.

The oil film of the entire piston/cylinder pair can be discretized into several nodes. At the beginning of the calculation, an initial oil film thickness field and corresponding thickness changing rates are assigned. In order to determine the action force of oil film on the piston at each node, the pressure distribution of piston/cylinder pair oil film is obtained by solving the discretized Reynolds Equation. After that, the dynamics of the piston and the hydrodynamic pressure distribution can be substituted into the force and moment balance equation. Then the Newton-Raphson method is used for the iterative solution. The oil film thickness and thickness changing rate will adjust automatically if the force and moment balance equations are not satisfied. The process above will repeat until the forces and moments are balanced. The calculation process of the numerical model is shown in **Figure 3**.

Figure 3: Calculation process of the numerical model

In **Figure 3**, k is the rotating angle of the cylinder with the resolution of 1 degree, p is the discretized hydrodynamic pressure, h is the discretized oil film thickness, $\dot{h}$ is the discretized

oil film thickness changing rates, $\Delta \dot{h}$ is the variation of $\dot{h}$ in each iteration.

2.4. Process of the wear prediction method

Here is the process of the wear prediction model.

1. Firstly, a set of initial oil film characteristic parameters is calculated by the numerical simulation model.
2. Then, the oil film characteristic parameters are fed to the wear model as key boundary conditions, and the wear depth at each node per cycle can be gained and superimposed to the previous cycle. Adhesive wear model and partial-EHL contacts sliding wear model are used to calculate the wear depth. They are switched according to the load-lubrication condition.
3. Since the wear influences the oil film thickness, the wear depth will be input to the numerical simulation model to recalculate the oil film characteristic parameters when the maximum wear depth exceeds a certain threshold.
4. The above steps repeat until the specified number of revolutions is reached.

The Schematic diagram of the model structure is shown in **Figure 4**.

Figure 4: Schematic diagram of the model structure

In **Figure 4**, m is the number of oil film calculation, r is the revolution of the cylinder, Δw is the wear distribution during one degree, w_r is the wear distribution during one revolution, w is the wear distribution before the next oil film calculation, and w^0 equals to zero, w_t is the wear threshold for calculating oil film.

The simulation was carried out in accordance with this method. The simulation condition is shown in **Table 1**.

Table 1: Simulation condition

Condition	Value
Running time	240 [h]
Loading pressure	42 [MPa]
Rotating speed	1450 [rpm]

2.5. Simulated results

The simulated results can be displayed in the same way as the oil film parameters be displayed as shown in **Figure 5**. It can be seen that the wear of two ends of the cylinder bore is relatively serious in the specific ranges of circumferential angle.

Figure 5: Simulated result of wear distribution

3. EXPERIMENTAL VALIDATION

3.1. Test of pump

In order to verify the wear regions and corresponding simulated wear depth, a real-machine experiment for an axial piston pump was carried out. A new KD-H180 axial piston pump was tested in an accelerated lifetime test bench, which lasted 240 hours. The test condition is consistent with the simulation condition as shown in **Table 1**.

The test bench is shown in **Figure 6**.

Figure 6: Test bench and test pump

3.2. Measurement of wear

The tested pump was disassembled and the cylinder was taken out to be measured before and after the durability test. In order to obtain the complete wear distribution of the cylinder bore, the cylinder bore was divided into several sections along the axis for measurement as shown in **Figure 7**. It can be seen from **Figure 5** that the axial wear gradient near the ends of the cylinder bore is much larger than that elsewhere, so the measurement section near the ends of the cylinder bore was more intensive than that elsewhere.

Figure 7: Site of the measured cross-section

The circumferential profile of each section was measured by a coordinate measuring machine (CMM), and these circumferential profiles can be fitted into a three-dimensional profile as shown in **Figure 8**.

Figure 8: Three-dimensional profile of the cylinder bore

3.3. Support of measured results

The three-dimensional profile of the cylinder bore can be spread out into a plane for intuitively comparing the measured result with the simulated result more. The correspondence between the simulated result and the measured result can be shown in **Figure 9**. As shown in **Figure 9(a)**, both the circumferential angle coordinate and the axial length coordinate are the same as those of the simulated results. The wear regions and

corresponding wear depth in the simulated results are reflected in **Figure 9(b)**.

Figure 9: Entire profile of the cylinder bore

It can be seen from **Figure 9** that in the axial direction, the most serious wear regions are near the ends of the cylinder bore. The wear profile near the ends can be displayed in the circumferential direction for intuitive analysis. **Figure 10** shows the comparison of the simulated and measured profile of the cylinder bore.

Figure 10: Circumferential profile of the cylinder bore

It can be seen that the measured profiles have a good correspondence with the simulated results. For the side of valve plate, the most severe wear regions are at the angle of 210 °. For the side of swash plate, the most severe wear regions are at the angle of 50 °. The cause of the above phenomena needs to be explored. When the piston is working in the high-pressure area, the piston inclines in the cylinder bore under the huge lateral force, and the oil film at the ends of the cylinder bore is thinner. The squeezing effect of oil film is strengthened in these regions, in other word, the load on these regions is enhanced. Solid contact is easy to occur at the ends of the cylinder bore because of the thinner oil film.

4. CONCLUSION AND OUTLOOK

In this paper, a wear prediction method of piston/cylinder friction pair is put forward. The

wear regions and corresponding wear depth of piston/cylinder pair can be obtained by this method. The validity of the method is proved by the experiment, and the measured results show that the accuracy of the wear regions and corresponding wear depth prediction is high.

In the future, the effect of medium viscosity, swashplate angle, and other related structure parameters of the pump can be studied. The wear prediction method can be used to pump healthy management, and can help to choose the suitable working conditions and structural parameters of the axial piston pump.

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NOMENCLATURE

D_C	Diameter of cylinder bore	[mm]
L_C	Effective length of cylinder bore	[mm]
δ_a	Axial lengths of node	[mm]
δ_c	Circumferential lengths of node	[mm]
p_{node}	Pressure of each node	[bar]
p_{ta}	Pressure thresholds for adhesive wear model	[bar]
p_{ts}	Pressure thresholds for sliding wear model	
h_{node}	Oil film thickness of each node	[μ m]
h_t	Oil film thickness threshold for sliding wear model	[μ m]
c_{node}	Clearance without wear of each node	[μ m]
T	operating time of pump	[s]
F_p	Pressure force	[N]
F_{ss}	Slipper supporting force	[N]
F_{SSz}	Axial component of slipper supporting force	[N]
F_{SSy}	Lateral component of slipper supporting force	[N]
F_{sf}	Slipper friction force	[N]
F_L	Lateral force	[N]
V	Wear volume	[mm ³]
s	Sliding distance	[mm]
σ_s	Compressive yield limit	[MPa]
k_s	Wear constant	[-]

t	Sliding time	[s]
u	Sliding velocity	[mm/s]
δ_0	Height of roughness peak	[μm]
λ	Wave length of roughness peak	[μm]
τ	Delay time of cylinder bore material	[s]
p	Hydrodynamic pressure	[bar]
h	Oil film thickness	[μm]
μ	Dynamic viscosity of oil	[Pa·s]
ω_p	Speed of piston spin	[rad/s]
k	Rotating angle of cylinder	[°]
p	Discretized hydrodynamic pressure	[bar]
h	Discretized oil film thickness	[μm]
$\dot{h}$	Discretized oil film thickness changing rates	[$\mu\text{m/s}$]
$\Delta\dot{h}$	Variation of $\dot{h}$ in each iteration	[$\mu\text{m/s}$]
m	Number of oil film calculation	[-]
r	Revolution of cylinder	[-]
Δw	Wear distribution during one degree	[μm]
w_r	Wear distribution during one revolution	[μm]
w	Wear distribution before next oil film calculation	[μm]
w_t	Wear threshold for calculating oil film	[μm]

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