

Power loss of slipper/swashplate based on elastohydrodynamic Lubrication model in axial piston pump

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Abstract. Energy saving of the hydraulic system is significantly realized by improving the efficiency of the pump. The slipper/swashplate lubricating surface is one of the main energy consumption parts in axial piston pump which is always used in the hydraulic system. Being a hydraulic bearing in axial piston pump, the slipper/swashplate pair exerts both the bearing function and the sealing function under the oscillating working condition. Because of viscous friction and leakage flow, power loss of slipper/swashplate is also one of the main power loss sources in axial piston pump. A precise model of lubricating surfaces, which includes oil film thickness, dynamic pressure field, load carrying ability and energy dissipation, is indispensable to develop more effective lubricating surface morphology. An elastohydrodynamic lubricating oil film model has been established for the sliding surfaces between slipper/swashplate interfaces. The model includes an isothermal fluid model, micro-motion of slipper and pressure deformation of the bounding solid bodies using a partitioned solution scheme. The power loss model between the slipper and swashplate has been developed and solved by numerical analysis method. Numerical results demonstrate that the diameter of the fixed damping orifice in the slipper has an important role in power loss of slipper/swashplate except the axial rotational velocity and operating pressure. Finally, the accuracy of numerical results of power loss is verified by experiments.

Nomenclature

γ	=	swashplate tilt angle ($^{\circ}$)
φ	=	cylinder instantaneous rotational angle ($^{\circ}$)
ω	=	rotational angular velocity (r/min)
R_f	=	radius of piston distribution circle on cylinder (m)
d_z	=	piston diameter (m)
p_b	=	displacement chamber pressure (Pa)
p_s	=	pressure on central cavity of slipper (Pa)
p_d	=	pressure on slipper bearing surface (Pa)
f_{gz}	=	piston-bore fictional force (N)
m_h	=	mass of slipper (kg)
R_{yH}	=	distance between center of slipper and center of swashplate while φ
M_{LH}	=	slipper centrifugal moment (Nm)
l_{LH}	=	distance between center of mass of slipper and ball (m)
τ	=	oil film shearing stress of slipper/swashplate (Pa)



μ	= hydraulic dynamic viscosity (Pa.s)
h	= oil film thickness (m)
R_1	= outer radius of slipper (m)
R_2	= radius of the central cavity of slipper (m)
f_{SH}	= viscous frictional force of oil film of slipper/swashplate (Pa)
M_{SH}	= viscous frictional moment of oil film of slipper/swashplate (Nm)
l_{SH}	= distance between bottom of slipper and ball socket (m)
A	= area of sealing surface on slipper (m ²)
F_{HH}	= retaining force (N)
n	= number of pistons
k	= spring stiffness (N/m)
Δx	= previous compressed length of spring (m)
M_{DXH}	= dynamic pressure moment of slipper/swashplate along x_H axis (Nm)
M_{DYH}	= dynamic pressure moment of slipper/swashplate along y_H axis (Nm)
ρ	= oil density (kg/m ³)
v	= velocity vector of fluid flow (m/s)
p	= fluid pressure (Pa)
f	= fluid mass force (N)
W_1, W_2	= shearing and frictional power loss of slipper pair, respectively (W)
W	= total power loss of slipper pair (W)

Introduction

Axial piston pumps have the compact structure and the high performance, which could be suitable for any hydraulic applications and are widely used in industry field. The properties of the three crucial frictional pairs determine the performance of axial piston pumps. Professor Yang and Zhou who are from Zhejiang University researched the critical technology of frictional interfaces in the seawater pump and the oil pump [1-3]. Professor Yuan who is from Beijing Institute of Technology mainly studied the pressure distribution and the oil film leakage of frictional pairs in axial piston pump [4-6]. The frictional pair between slipper and swashplate is one of three key frictional interfaces of axial piston pumps. Previous researchers paid much attention to lubrication mathematical models of the slipper/swashplate interfaces. Koc and Hooke established analytical numerical models and conducted experiments to analyze slipper movements under different working conditions [7,8]. Kazama and Yamaguich [9] established a mixed frictional model in the water pump, and Bergada [10,11] established the computational fluid dynamics (CFD) numerical models to get the slipper leakage flow. Many mathematical models have been established for the slipper/swashplate pair of axial piston pumps [12-14], however, these models have not considered the elastohydrodynamic lubrication characteristic of the slipper/swashplate pair.

Modeling approach

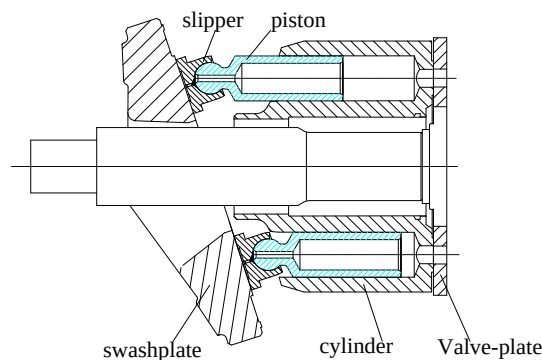


Figure 1. Structure of axial piston pump.

1.1. Kinematic model

Figure 1 shows the structure of axial piston pump. Slippers are connected with pistons by ball joints. Forces acted on slippers depend on forces acted on pistons, which are relative to the piston displacement. The operating trail of piston is seen in figure 2.

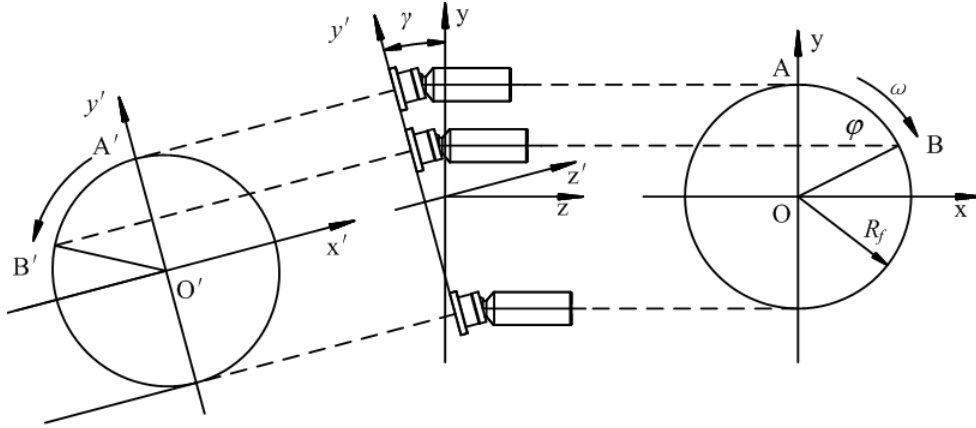


Figure 2. The operating trail of piston.

The coordinate system (oxyz) is located on cylinder. Point A which is the center of the ball joint is defined as the starting position of the operating trail of piston. The displacement of piston from A to B along the z axis is described as,

$$s = R_f \tan \gamma (1 - \cos \varphi) \quad (1)$$

In the coordinate system (ox'y'z') being located on swashplate surface, the operating trail of slipper is elliptical path, as follows:

$$\begin{cases} y' = R_f \cos \varphi / \cos \gamma \\ z' = R_f \sin \varphi \end{cases} \quad (2)$$

Rotational angular velocity of slipper along its operating trail on the swashplate can be described as,

$$\omega_h = \frac{\cos \gamma}{\cos^2 \varphi + \sin^2 \varphi \cos^2 \gamma} \omega \quad (3)$$

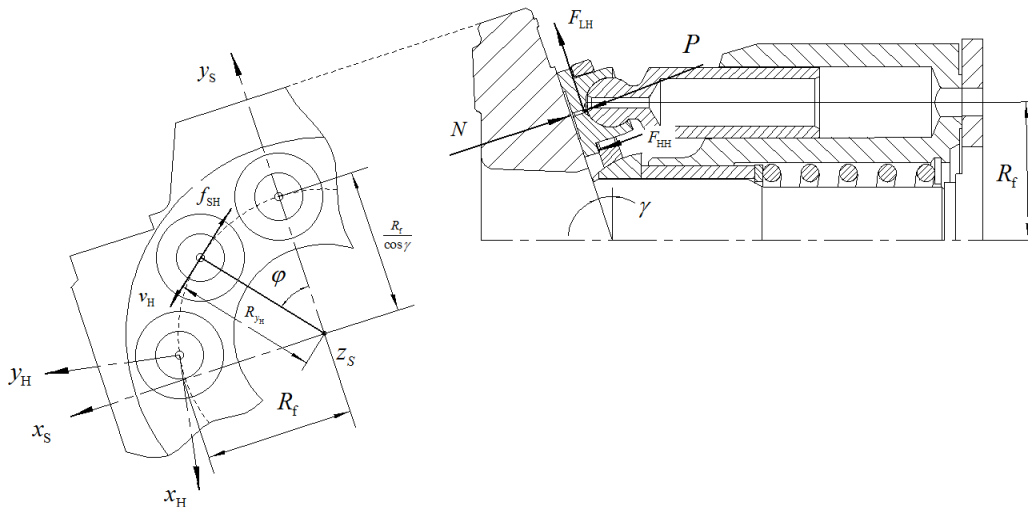


Figure 3. Dynamic analysis diagram of slipper.

1.2. Dynamic model

As shown in figure 3, there are five main forces acted on slipper, such as an external load force, a centrifugal force, a viscous frictional force, a bearing force and a retaining force. The external load force acted on the slipper coming from the piston through the ball center is a net force of the displacement chamber pressure, the inertia force of the piston/slipper and the frictional force between piston/cylinder interfaces, which could not generate a ball joint moment.

The rest forces acted on the slipper will bring simultaneously an axial force and a moment.

$$P = \frac{\frac{\pi}{4} d_z^2 \cdot p_b + m_{zh} \cdot \omega^2 \cdot R_f \tan \gamma \cos \varphi + f_{gz}}{\cos \gamma} \quad (4)$$

The centrifugal force and moment are described as,

$$F_{LH} = m_h R_{yH} \omega^2 \quad (5)$$

$$M_{LH} = F_{LH} \cdot l_{LH} \quad (6)$$

And the viscous frictional force and moment are written as,

$$f_{SH} = \int_A \tau dA \quad (7)$$

$$M_{SH} = f_{SH} \cdot l_{SH} \quad (8)$$

Where $\tau = \mu \frac{\partial v}{\partial h}$

The bearing forces of slipper, including the hydrostatic bearing force from the central cavity and the hydrodynamic bearing force from the sealing surface, are equal to the integral of pressure acted on the slipper plane with respect to the area of sealing surface. The bearing force through the ball center, which could not generate the moment, can be analytically expressed as,

$$N = \int_A (p_s + p_d) dA \quad (9)$$

By a spring type retainer in the center of cylinder block, the spring force is considered as an invariable force. The spring compress would change slightly because of the slipper micro-motion, but the spring stiffness is always less than 1 N/μm. The retaining force on the single piston is expressed as,

$$F_{HH} = \frac{1}{n} k \cdot \Delta x \quad (10)$$

When the pump operates steadily, the net forces and moments should take a dynamic balance, as follows,

$$\begin{cases} N - P - F_{HH} = 0 \\ M_{DxH} - M_{LH} = 0 \\ M_{DyH} - M_{SH} = 0 \end{cases} \quad (11)$$

1.3. The oil film pressure model and the central cavity pressure model

As seen in figure 4, provided that the pressure in pump house is atmosphere, the hydraulic oil of the

displacement chamber flows into the pump house through the orifices of piston and slipper and the gap between slipper and swashplate.

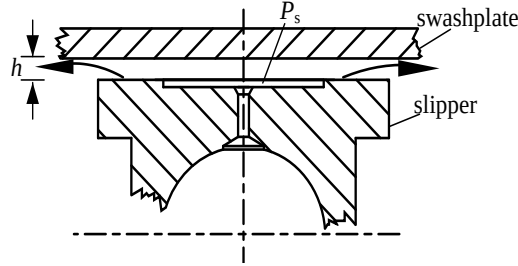


Figure 4. Structure of the slipper/swashplate pair.

The Navier-Stokes equation describing the viscous fluid flow can be expressed as,

$$\frac{\partial \mathbf{v}}{\partial t} + (\mathbf{v} \cdot \nabla) \mathbf{v} = \mathbf{f} - \frac{1}{\rho} \nabla p + \frac{1}{\rho} \nabla \cdot (\mu \cdot \nabla \mathbf{v}) \quad (12)$$

According to the actual working conditions, making the common assumptions, such as the incompressible Newton fluid, the constant fluid flow, no axial separated velocity, neglecting mass force, layer fluid flow only along radial direction and neglecting thermal, equation (12) can be simplified as,

$$v_r \frac{\partial v_r}{\partial r} = -\frac{1}{\rho} \frac{\partial p}{\partial r} + \frac{\mu}{\rho} \left(-\frac{V_r}{r^2} + \frac{\partial^2 v_r}{\partial r^2} + \frac{\partial^2 v_r}{\partial z^2} + \frac{1}{r} \frac{\partial v_r}{\partial r} \right) \quad (13)$$

Equation (13) is analytically computed to get the pressure distribution on the bearing surface, as follows,

$$p_d = \frac{6\mu q \ln \frac{R_1}{r}}{\pi h^3} + \frac{3\rho \omega^2 (r^2 - R_1^2)}{20} + \frac{27\rho \cdot q^2}{140\pi^2 h^2} \left(\frac{1}{R_1^2} - \frac{1}{r^2} \right) \quad (14)$$

Provided that $r=R_2$, the pressure on central cavity of slipper can be expressed by,

$$p_s = \frac{6\mu q}{\pi h^3} \ln \frac{R_1}{R_2} + \frac{3\rho \omega^2}{20} (R_2^2 - R_1^2) + \frac{27\rho q^2}{140\pi^2 h^2} \left(\frac{1}{R_1^2} - \frac{1}{R_2^2} \right) \quad (15)$$

1.4. Slipper deformation model

The deformation of slipper affects the oil film thickness on the slipper/swashplate interfaces, and then changes the fluid pressure distribution. Influence method which is very useful for elastic deformation, is utilized to calculate the deformation of slipper on the rigid swashplate. Assuming that the deformation of slipper is linear elasticity, after getting the pressure distribution of oil film, the deformation of slipper is written as,

$$\delta h_i = \sum_{j=1}^{N_s} E_{ij} \frac{p_i}{p_r} \quad (16)$$

Where p_r is reference pressure. E_{ij} is the deformation of cell face j in solid mesh with the reference pressure acted on cell face i . And p_j is the real pressure acted on cell face j . The process of generating influence matrix is completed with C++ and OpenFOAM library. The solid meshes of slipper is imported from ANSYS, which is made of unconstructed tetrahedral elements.

1.5. Power loss model

Working at high shaft rational velocity, there are the oil film lubrication on slipper/swashplate interfaces and the oil flow through the gap between slipper and swashplate, due to the shearing movement and the differential pressure, as seen in figure 5. Based on the previous documents [15] of Prof. Xu from Zhejiang University, the comprehensive mathematic model of the power loss of the slipper bearing is developed and solved by numerical methodology.

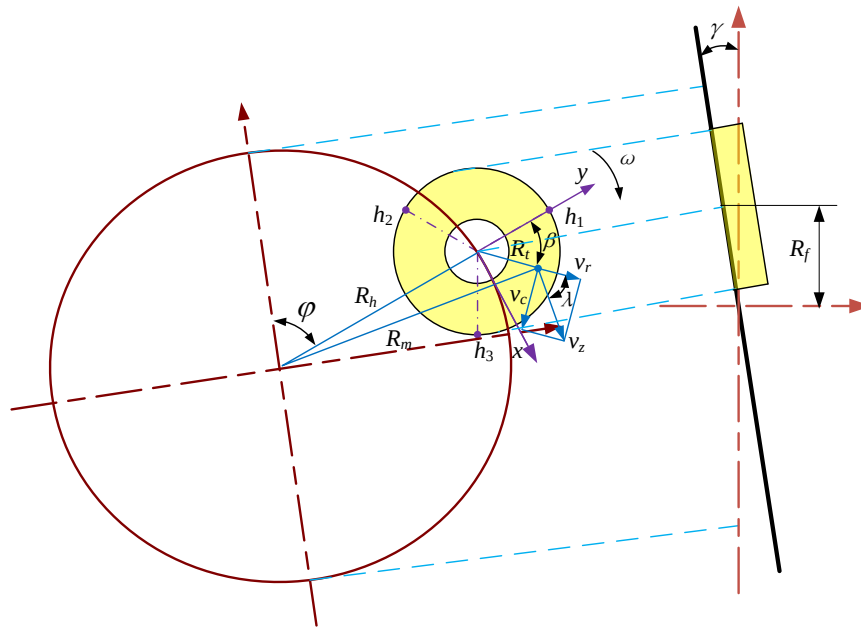


Figure 5. Oil film characteristics of slipper/swashplate interfaces.

Considering the parallel oil film on the slipper bearing surface, the radial and circumferential shearing stress is expressed as, respectively,

$$\tau_r = \frac{\partial p}{\partial r} \frac{h}{2} + \eta \frac{v_r}{h} = \frac{\partial p}{\partial r} \frac{h}{2} + \eta \frac{v_z \cos \lambda}{h} \quad (17)$$

$$\tau_c = \frac{\partial p}{r \partial \beta} \frac{h}{2} + \eta \frac{v_r}{h} = \frac{\partial p}{r \partial \beta} \frac{h}{2} + \eta \frac{v_z \sin \lambda}{h} \quad (18)$$

Therefore, the shearing frictional moment of oil film is expressed as,

$$M = \int_0^{2\pi} \int_{R_i}^{R_l} R_m (\tau_r \cos \lambda + \tau_c \sin \lambda) R_l dR_l d\beta \quad (19)$$

Where

$$R_m = R_f \sqrt{1 + \tan^2 \gamma \cos \varphi} + R_t \cos \beta \quad (20)$$

$$\lambda = \arcsin\left(\frac{R_h \sin(\phi)}{R_s}\right) \quad (21)$$

The power loss resulted from the shearing flow of the slipper bearing surface is written as,

$$W_1 = \frac{\omega}{\sqrt{\cos^2 \varphi + \sin^2 \varphi \cos^2 \gamma}} M \quad (22)$$

Considering the hydrodynamic shearing effect and the differential pressure effect, the radial velocity can be decomposed into two items, as follows:

$$v_a = \frac{1}{2\eta} \frac{\partial p}{\partial r} (z^2 - zh) + v_r \frac{z}{h} \quad (23)$$

The power loss resulted from the differential flow of the slipper bearing surface is written as,

$$W_2 = \Delta p Q = \Delta p \int_0^{2\pi} \int_0^h R_t v_a dz d\beta \quad (24)$$

Therefore the total power loss between slipper/swashplate interfaces is written as

$$W = W_1 + W_2 \quad (25)$$

The power loss of the slipper bearing surface is determined by rotational velocity and operating pressure. Though the effect of the rotational velocity and the operating pressure on the oil film is complicated, it is investigated by numerical simulation and experiment in this paper.

Numerical simulation results

1.6. Impact of damping orifice on oil film

The micro-motion of slipper depends on the damping orifice and the external load. The cross sectional area of the damping orifice determines the magnitude of the pressure decrease through the damping orifice and the delay time of pressure development in the central cavity of slipper. Therefore, at the high rotational velocity, the response frequency of pressure in the central cavity of slipper is required to be suitable for the oscillating condition. For a specific slipper, the function of the oil film thickness h , fluid pressure p and the pressure p_s on the bearing surface of slipper is expressed as,

$$\frac{p_s}{p} = \frac{1}{1 + \frac{64}{3 \ln R_1 / R_2} \frac{l}{d_1^4} h^3} \quad (26)$$

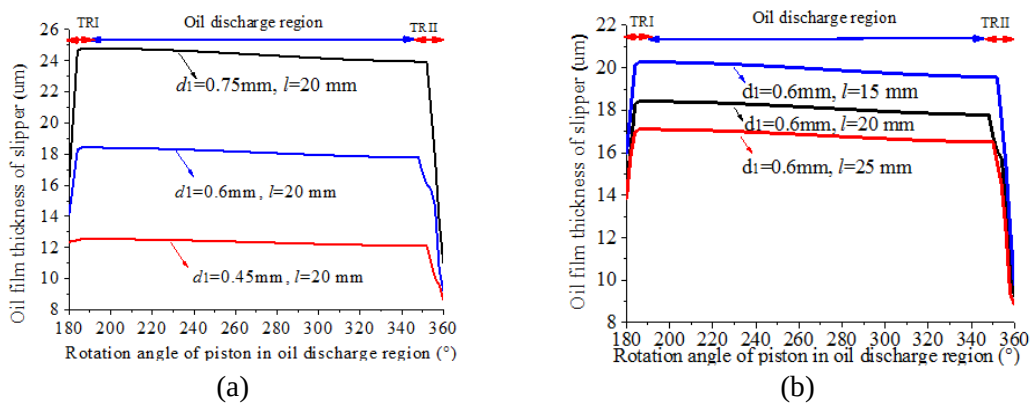


Figure 6. Effect of sizes of the damping orifice on the oil film thickness. (a) Effect of diameter on the oil film thickness and (b) Effect of length on the oil film thickness.

The wear of slippers always occurs in the oil discharge region. The influence of diameter and length on the oil film thickness is shown in figure 6, where d_1 and l are the diameter and length of the damping orifice, respectively. The effect of diameter on the oil film thickness is more than the effect of length on.

1.7. Power loss simulation

As shown in figure 7, under the same working condition, the influence of the working pressure on the power loss is smaller than the effect of the rotational velocity on. Under the working conditions of the constant rotational velocity and the different operating pressure, the power loss in suction oil region is nearly unchangeable, while the power loss in discharge oil region has already significantly changed. The pressure in the suction oil region is almost similar to the pressure in the pump house, while the pressure in the oil discharge region is changed with the external operating pressure.

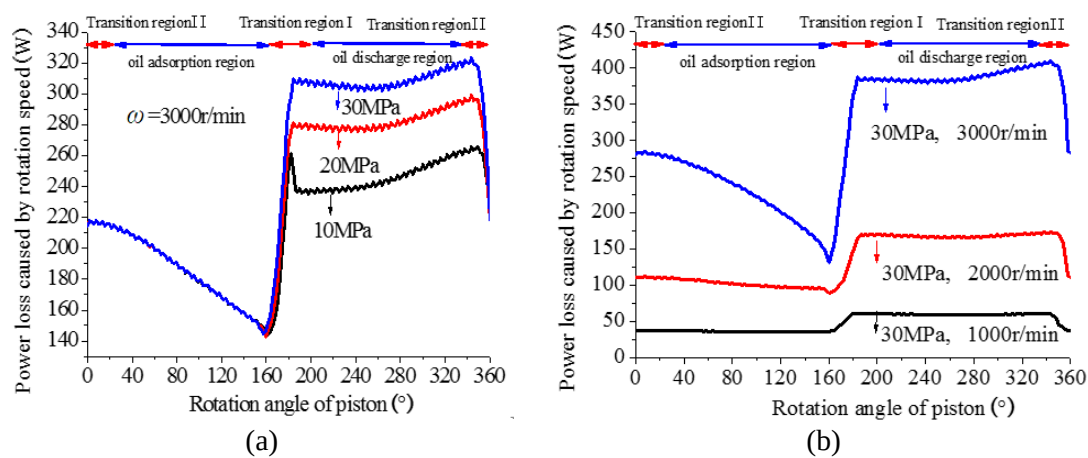


Figure 7. Power loss resulted from shearing flow of slipper under different operating conditions. (a) Effect of the operating pressure on the power loss resulted from shearing flow and (b) Effect of the rotational velocity on the power loss resulted from shearing flow.

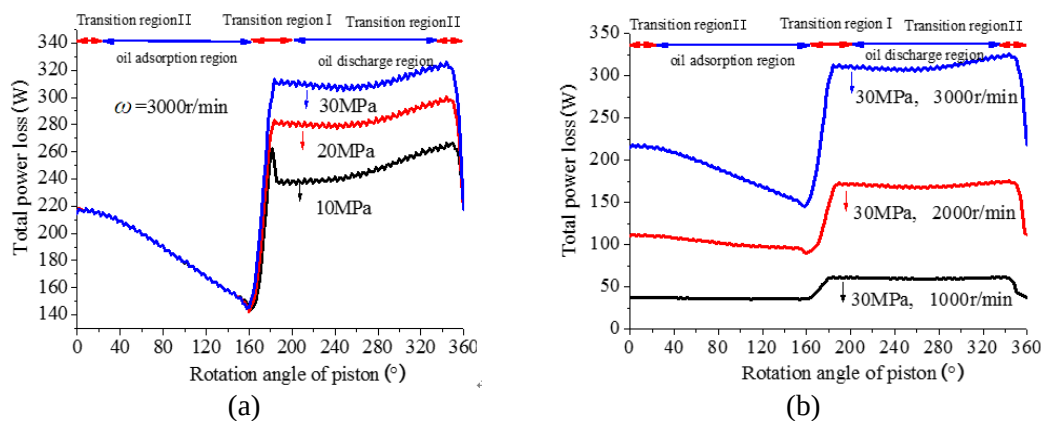


Figure 8. Total power loss of slipper/swashplate interfaces under different operating conditions. (a) Effect of the operating pressure on the total power loss and (b) Effect of the rotational speed on the total power loss.

With the operating pressure rising up, the oil film thickness decreases, resulting in decreasing the leakage and the power loss resulted from the differential pressure flow, while with the rotational velocity increasing, the oil film thickness increases, promoting the fluid leakage flow and shearing stress due to the dynamic and thermal effects. In other words, the rotational velocity plays a positive

role in the power loss. It is also shown in figure 8 that the influence of the total power loss on the rotational velocity is much greater than the effect of the operating pressure on.

Experiment

1.8. Experiment rig

It is very difficult in the real pump to install sensors to measure the power loss of slipper/swashplate and the power loss of cylinder/valve-plate, respectively. Therefore this rig is manufactured by modifying the structure of the type A4VSO45 pump and the function of pump is realized by a couple of check valves and the manual operational handle of the swashplate angle, as shown in figure 9. Because of the limitation of the dynamic response time of the check valves, the rotational velocity is slightly confined; it has no effect on the external operating pressure increasing.

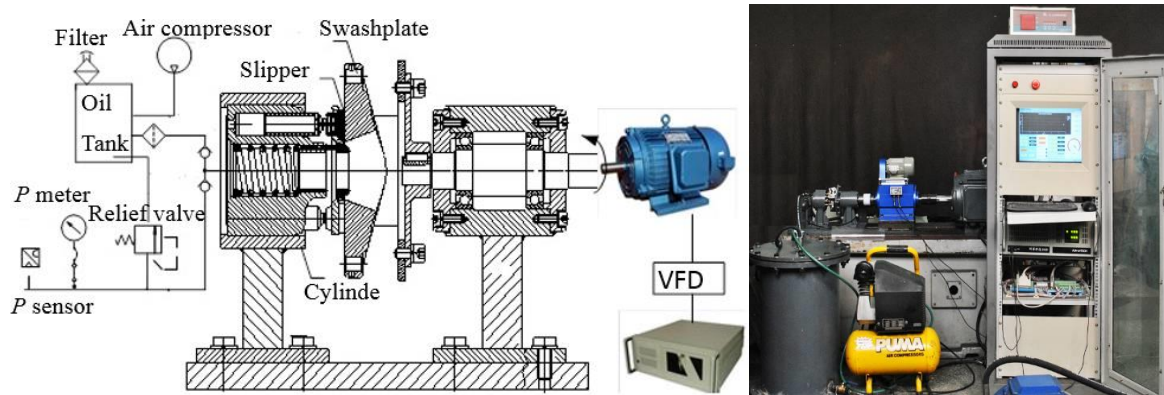


Figure 9. Schematic diagram and picture of the experimental rig.

1.9. Experimental results

Because of the sampling frequency limitation of sensors, curves of experimental results exist fluctuant, as seen in figure 10. At the constant operating pressure, the power loss changes significantly with the rotational velocity from 1000 r/min to 1500 r/min to 2000 r/min, while at the constant rotational velocity, it changes slightly with the operating pressure from 5 MPa to 7.5 MPa to 10 MPa, demonstrating that the influence of the shaft velocity on the power loss is greater than that of the operating pressure on it between slipper/swashplate interfaces in axial piston pump, which is consistent with the numerical simulation results of the power loss.

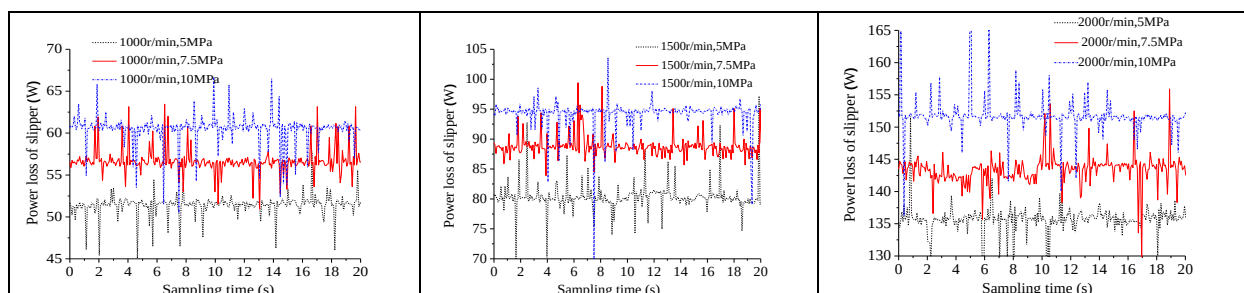


Figure 10. Experimental results of power loss of slipper/swashplate interfaces under different operating conditions.

Conclusion

The oil film lubrication model between slipper/swashplate interfaces in axial piston pump based on the Navier-Stokes equation has been established and the effect of the diameter and length of the damping

orifice on the oil film thickness has been researched. The simulation results show that the effect of the diameter of the damping orifice on the oil film thickness is greater than that of the length, which provides the feasible method to optimize slipper structure for improving the lubricating performance to save energy of hydraulic systems.

The power loss model of slipper/swashplate considering the slipper deformation has been developed. The influence of the external operating pressure and the rotational velocity on the power loss has been significantly investigated. The numerical and experimental results have demonstrated that the influence of the rotational velocity on the power loss is more obvious than the influence of the external operating pressure on. In addition, a measure method of power loss of slipper/swashplate has been proposed in axial piston pump.

Acknowledgments

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