A study on the behavior of the film thickness between piston shoe and swash plate using the squeeze film effect and the drag flow effect

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ABSTRACT

The purpose of this study is to explain the loss mechanism between piston shoe and swash plate of axial piston units and to derive new presuppositions for developing a new unit or improving an existing one. The film thickness was measured at constant pressure to find out the correlation between speed and the drag flow effect. The behavior of the film thickness between piston shoe and swash plate in the real unit was described through combining a squeeze-film-effect with a drag flow effect.

KEY WORDS

Piston-shoe, Squeeze-film-effect, drag-flow-effect, loss-mechanism

NOMENCLATURE

p : Pressure r : Radius
d : diameter A : Area
F : Force q_s : Bearing ratio
h : Film thickness

INTRODUCTION

The loss mechanism between piston shoe and swash plate is one of 3 main loss points of the axial piston pump and motor. In order to explain the exact mechanism, Böinghoff[B1] and

Rinck[R1] applied the squeeze film effect which explains dynamic pressure development under the falling shoe pad on the swash plate. But the behavior of the piston shoe is influenced by the drag flow effect that depends on the film thickness and the travelling speed. In this study the real characteristics of the piston shoe of the actual unit is described by the combination of the squeeze film effect and the drag flow effect using the measured film thickness.

1. The static and dynamic pressure profiles between piston shoe and swash plate

The piston shoe roles as the bearing to support the rotary unit on the swash plate. If the system is overloaded, then it causes to make a metal contact and grow into fatal failure. In the reverse case, it produces a big volumetric loss and makes the pad unstable. The main parameters, determining the dynamic pressure profile, are drag flow, thermal flow and squeeze film effect. The static pressure profile and the load capacity are obtained using Reynolds equation [R2].

$$\frac{1}{r} \frac{\partial}{\partial \theta} \left(\frac{\rho h^3}{\eta} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial r} \left(\frac{r \rho h^3}{\eta} \frac{\partial p}{\partial r} \right) = -6u \frac{\partial h}{\partial \theta} \sin \theta + 6ur \frac{\partial h}{\partial r} \cos \theta \quad (1)$$

where h is film thickness, ρ density, η dynamic viscosity, p pressure, θ

rotational angle and $\theta_{\max}$ the angle of the maximum film thickness position from the reference.

To derive the static pressure profile, we put the speed $u=0$.

$$p = p_s \cdot \left(1 + \frac{\ln(r/r_i)}{\ln(r_a/r_a)} \right) \quad (2)$$

Through the integral of the pressure profile, we can get the load capacity:

$$F_{st} = \frac{p_s \cdot \pi \cdot (r_a^2 - r_i^2)}{2 \ln(r_a/r_i)} \quad (3)$$

Figure 1 The piston shoe (a) actual model, (b) calculation model

The force acting on the piston by the high pressure is as follows:

$$F_k = \frac{p_k \cdot \pi \cdot d_k^2}{4} \quad (4)$$

We define the bearing ratio $q_s = F_{st} / F_k$ under the condition that p_k and p_s are equal.

$$q_s = \frac{2 \cdot (r_a^2 - r_i^2)}{d_k^2 \cdot \ln(r_a / r_i)} \quad (5)$$

Böinghoff derived the governing equations of the squeeze film effect from the Navier-Stokes equation and the continuous equation [B1].

The pressure profile is as follows:

$$p(r) = \frac{3\eta}{h^3} \frac{\delta h}{\delta t} \left\{ r^2 - r_i^2 - (r_a^2 - r_i^2) \frac{\ln(r / r_i)}{\ln(r_a / r_i)} \right\} \quad (6)$$

By the integral of the above equation, we can have the load capacity:

$$F_H = \frac{3\pi\eta}{2h^3} \frac{\delta h}{\delta t} \left\{ r_i^4 - r_a^4 + \frac{(r_a^2 - r_i^2)^2}{\ln(r_a / r_i)} \right\} \quad (7)$$

The film thickness h varies also as time changes.

$$h(t) = \frac{h_0}{\sqrt{1 - \frac{4h_0^2 \cdot F_H \cdot t}{3\pi\eta U}}} \quad (8)$$

where

$$U = \sum_{i=1}^n \left[r_{ii}^4 - r_{ai}^4 + \frac{(r_{ai}^2 - r_{ii}^2)^2}{\ln(r_{ai} / r_{ii})} \right]$$

and n is the number of lands.

2. The film thickness by the squeeze film effect

Böinghoff proposed the initial value of the film thickness as follows:

$$h_0 = d_k \cdot 10^{-3} \quad (9)$$

Figure 2 A typical pressure behavior at the piston chamber during one rotation

As we see in the Figure 2, the pressure at the piston chamber changes between low pressure and high pressure basically. While the piston shoe moves under the low pressure, the film thickness goes high because of the drag flow effect. As the chamber is connected to the high pressure line by the valve plate, the pressure at the chamber rises suddenly. Owing to this sudden change of the pressure, the shoe pad falls down to meet the force equilibrium through the squeeze film effect.

As we see in the Figure 3, the initial value has less meaning because of its steep falling gain. Therefore we can accept the Böinghoff's proposal on the initial value. Figure 4 shows the calculated behavior of the film thickness at 100bar and 1500rpm. Here

the hydraulic oil is HLP46 and the temperature is 40 °C.

Figure 3 The behavior of the film thickness at 100 bar

Figure 4 The calculated behavior of the film thickness at 100bar, 1500rpm and $h_0=17\mu\text{m}$.

3. The film thickness between piston shoe and swash plate through the drag flow effect and the squeeze film effect

In order to measure the film thickness between piston shoe and swash plate, the model test bench was developed.

Figure 5 The schematic diagram of the model test bench

As we find in the Figure 5, pump, accumulator and relief valve are applied to build up the power supply. A hydraulic servo motor is used as a driving unit under the table. At the center of the system the model for a piston shoe is placed with the oil tank which keeps the oil temperature constant. The press cylinder and sensors are located at the top of the bench.

Figure 6 The model for piston shoe

In the model for piston shoe, only three pistons are applied to give a room to install sensors and jigs.

Figure 7 Layout of the sensing system for the film thickness

Three gap sensors are used to measure the film thickness between the piston shoe and swash plate, as it is shown in the Figure 7.

Figure 8 Film thickness between piston shoe and swash plate at 100 bar

The mean film thickness can be formulated as follows:

$$h_m = \sqrt[3]{(155 + 0.6n)} - 5.85 \quad (10)$$

Here n is the rotational speed in rpm and effective between 50 rpm and 2,000 rpm, h_m is the mean film thickness in micro meter. The required dynamic load capacity (F_{sd}) is constant over all the rotational speed range in the model test.

$$F_{sd} = \frac{0.6 \cdot F_{sd0} \cdot n}{(h + 5.85)^3 - 155} \quad (11)$$

Here $F_{sd0} = 100 \text{ bar} \cdot A_k \cdot (1 - q_s)$ and q_s is bearing ratio, A_k the cross section area of the piston. This form is similar to the load capacity of the hydrodynamic bearing.[R2] This dynamic load capacity is produced by the drag flow effect. In the actual unit the summation of this dynamic load capacity and that of the squeeze film effect meets the required force to the force balance.

$$F_{sd}|_{Sum} = F_{sd}|_{Squeeze-Film-E} + F_{sd}|_{Drag-Flow-E} \quad (12)$$

By the putting Eq.(7) and (11) into the Eq.(12),

$$F_{sd0} = \frac{3\pi\eta U}{2h^3} \frac{dh}{dt} + \frac{0.6 \cdot n \cdot F_{sd0}}{(h(t) + 5.85)^3 - 155} \quad (13)$$

can be obtained.

Figure 9 A simulated behavior of the hydrodynamic load capacities

On coming into the high pressure zone, the piston shoe starts to fall on the swash plate, namely the film thickness decrease steeply. Therefore the load capacity by the squeeze film thickness is dominant at the beginning and that by the drag flow effect increases as the thickness decreases, as we can see in the Figure 9.

Figure 10 A comparison of the film thickness behaviors at 1,500 rpm, 100 bar and $h_0=17\mu\text{m}$

In order to describe the film thickness in the real unit, three kinds of

behaviors are displayed in the Figure 10. The film thickness considering both effects is larger than that considering only the squeeze film effect.

CONCLUSION

- 1) The squeeze film effect and the drag flow effect are introduced to describe the behavior of the film thickness in the real unit.
- 2) The characteristic curve of the film thickness for the drag flow effect is formulated from the measured data and the governing equation for the dynamic load capacity is proposed.

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