Relation between Sliding-Part Temperature and Clearance Shape of A Slipper in Swashplate Axial Piston Motors*

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Temperature and behavior of a piston motor slipper were examined using a test rig. The experimental setup comprised a tester, a hydraulic circuit, control units, and measuring instruments. The tester included a servomotor and a chamber enclosing two piston assemblies with slippers, a rotating disk, displacement sensors, and thermocouples. The sliding surface diameter of the test slipper was 32 mm; the test oil was hydraulic oil with VG46. Tests were conducted at supply pressures up to 35 MPa, rotational speeds up to 26.7 s^{-1} , and oil temperature of $30\text{--}50^\circ\text{C}$. The pad temperature increased with the supply pressure and rotational speed. The pad established a temperature gradient, being maximized near the slightly-outer trailing edge and minimized near the leading edge. The temperature differences were influenced by the rotational speed, but were almost independent of the supply pressure.

Keywords: Tribology, Hydraulics, Bearing, Motor, Pump, Friction, Sensor, Experimental Analysis

1. Introduction

Hydraulic motors and pumps must transform energy with high power density and efficiency¹⁾⁻²⁾. As the discharge pressure and shaft speed increase, the bearing and sealing parts generate more heat. Moreover, as the component size becomes smaller, less heat is transferred to the outside and the cooling effect reduces. These phenomena cause performance degradation, wear, and seizure of the equipment, as well as property changes and reduce the life of the hydraulic oils.

Hydraulic motors and pumps are categorized into four types: swashplate axial piston, bent-axis piston, vane, and gear. The construction of swashplate axial piston motors and pumps is based on hydrostatic balance and a variable displacement mechanism. This category is consequently characterized by several advantageous properties, including high efficiency, high power density, and high response.

Swashplate axial piston motors and pumps are thus widely utilized in equipment, such as construction machinery, aircrafts, and machine tools. The prime bearing and sealing parts of the motors and pumps are the parts between the

slipper and the swashplate, between the piston and the cylinder bore, and between the cylinder-block and the valve-plate.

The slipper is one of the main elements because it supports the ends of the pistons under highly fluctuating load and sliding speed. The behaviors and motions of slippers have been established in several pioneering studies. In the 1960s and 1970s, Shute and Turnbull³⁾ and Böinghoff⁴⁾ investigated slippers under conditions of fluid film lubrication and parallel film thickness. In the 1980s Hook et al. discussed the effects of concave and convex shape of slipper lands⁵⁾ and the size of restrictors⁶⁾. Iboshi and Yamaguchi considered the effects of eccentric load and pad inclination⁷⁾, conducted numerical calculations under mixed lubrication condition⁸⁾, performed test pump experiments⁹⁾, and proposed an optimum design based on minimum power loss¹⁰⁾. Other important studies have been conducted by Matsumoto and Ikeya¹¹⁾, Yabe and Kubo¹²⁾, Harris et al.¹³⁾, Manring et al.¹⁴⁾, Kumar et al.¹⁵⁾, Bergada et al.¹⁶⁾, and Shenk and Ivantysynova¹⁷⁾.

Kazama and Yamaguchi investigated circular hydrostatic thrust pad bearings theoretically¹⁸⁾ and experimentally¹⁹⁾. Kazama et al. studied the effects of elastic deformation²⁰⁾ on the hydrostatic bearings. They further extended the bearing model to the slipper model to examine the motion and the tribological characteristics²¹⁾⁻²²⁾, but have yet to completely clarify how the slipper motion²³⁾⁻²⁴⁾ is related to thermal behavior²⁵⁾. The temperature variations are connected to the slipper behavior as well as to motor and pump performance.

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In this study, the temperature of the slipper land and the position of the slipper pad are simultaneously examined. The effects of operating conditions on the temperature rise and the relation between the temperature distributions and the clearance shape are discussed.

2. Nomenclature

h	:	Clearance
h_c	:	Center clearance
N	:	Disk rotational speed
p_s	:	Supply pressure
R_2	:	Outer radius of slipper land
t	:	Temperature
t_c	:	Oil temperature in chamber
α	:	Slipper inclination angle
Δt	:	Pad temperature rise
Δt_m	:	Pad mean temperature rise
δt	:	Difference in temperature
θ	:	Circumferential coordinate
ϕ	:	Slipper azimuth angle
Subscript		
max	:	Maximum
min	:	Minimum

3. Experiment

The experimental setup consisted of a tester, measuring instruments, control units, and a hydraulic circuit²⁶⁾. The tester (see Fig. 1) primarily included an oil-bath type chamber and an electric servomotor (rated output = 4.4 kW, rated torque = 28.4 N·m, maximum rotational speed = 50 s⁻¹).

The chamber enclosed two piston assemblies with slippers, displacement sensors (range 0–0.5 mm, resolution = 0.1 μ m), thermocouples (type K, tolerance $\pm 2.5^\circ\text{C}$ (class 2 in JIS, Japan Industrial Standards), $\phi 0.25$), heaters (2 kW $\times$ 2), and a rotating disk (diameter = 272 mm). One of the slippers was used for testing, the other for counter-balancing. The hydrostatic balance ratio, defined as the ratio of the load to the maximum hydrostatic load-carrying capacity¹⁸⁾ (equivalent to over-clamp ratio⁶⁾), was 1.02.

The main measuring instruments were as follows: 6 contacting type displacement sensors, 4 thermocouples, a pressure sensor (rating = 50 MPa, nonlinearity = 0.3% RO), and a data logger (internal effective resolution of 14 bits, sampling time 0.01–60 s) with an A/D converter (16 bit, sampling frequency = 250 kHz). These instrumental systems

allowed simultaneous measurement of the clearance shapes between the slipper and disk and the temperature distributions in the slipper pad²⁶⁾.

Figure 2a is the top-view schematic of the slippers and disk and Figure 2b is the photograph of the test slipper and the thermo-couples and the displacement sensors installed in the chamber. The effective diameter of the test slipper and the revolution diameter around the disk center were 32 mm and 110 mm, respectively. An annular ring was mounted at the slipper pad because the pad was too small to fix the displacement sensors.

Fig.1 Illustration of the tester.

Fig.2a Schematic of the slippers and disk (Top-view).

Fig.2b Photograph of the slipper and sensors (installed).

For measuring the disk fluctuation and distortion, the tips of three displacement sensors were contacted to the disk's surface. The absolute slipper displacement was measured by contacting the tips of the remaining three displacement sensors to the annular ring, which were placed behind the ring in Fig. 2b. Each pair of sensors was circumferentially positioned at 120° intervals. The pitch diameter of the sensors' location on the annular ring was 76 mm. The clearance between the slipper and disk was obtained from the differences in the outputs of the paired displacement sensors.

The four thermocouples were inserted at the side wall of the slipper pad. These were circumferentially positioned at 90° intervals for measuring the temperature distributions across the pad. All displacement sensors and thermocouples were placed in the chamber, where they were exposed to oil.

The pressure sensor was positioned in the conduit between the discharge port of the power unit and the inlet port of the test rig. The temperature of the oil in the chamber was measured by another thermocouple placed close to the rotating disk. The rotational speed was measured by a tachometer fitted between the electric servomotor and disk. The tachometer also maintained a constant speed through feedback systems.

The hydraulic circuit comprised a hydraulic pump, a filter, valves, and a reservoir. The pump had a maximum discharge pressure of 72 MPa and a discharge flow rate of $8.3 \times 10^{-6} \text{ m}^3/\text{s}$, while the filter had a maximum working pressure of 35 MPa and a mesh size of $8 \text{ }\mu\text{m}$.

The experiment was conducted as follows: the test chamber was filled with the test oil, and the remaining air was carefully removed. The oil temperature in the chamber

was heated to the test condition by the heaters. The slipper and disk were closely contacted by a preload (supply pressure = 1 MPa) applied to the test slipper, and adjusted to zero. The disk was then rotated under the test condition. Furthermore, the supply pressure was increased from 5 MPa to 35 MPa in 5 MPa increments and decreased from the maximum pressure to atmospheric pressure in 5 MPa decrements.

The rotational speed was retained at the set condition by feedback with the servomotor. The operation was performed twice, and the second round data were repeatable and stable; hence, they were used in the analysis. The period between the previously set condition and the present condition was 5 s (1 MPa/s). At each stage, the condition was held for 20 s; the displacement, temperature, pressure, flow-rate, and torque were recorded by the PC and the logger.

In the experiment, the supply pressure p_s was varied as 1, 5, 10, 15, 20, 25, 30, and 35 MPa (8 conditions); the rotational speed N was varied as 3.3, 6.7, 10.0, 13.3, 16.7, 20.0, 23.3, and 26.7 s^{-1} (8 conditions); and the oil temperature t_c was varied as 30, 40, and 50°C (3 conditions). The test oil was hydraulic oil with ISO VG (viscosity grade) 46 (density = 862 kg/m^3 at 15°C). The kinematic viscosity of the oil was 48.85 and $7.674 \text{ mm}^2/\text{s}$ at 40 and 100°C , respectively; thus, at 30 and 50°C , the kinematic viscosity was 78.4 and $32.5 \text{ mm}^2/\text{s}$, respectively.

4. Results and discussion

4.1 Clearance and temperature distributions

Figures 3 and 4 plot the circumferential distributions of the estimated clearance h and the temperature difference Δt , respectively. The experimental conditions were $p_s = 1, 5\text{--}35 \text{ MPa}$ in 5 MPa increments, the rotational speed of the disk $N = 26.7 \text{ s}^{-1}$ (sliding speed at the slipper center = 9.2 m/s), and the oil temperature in the chamber $t_c = 30^\circ\text{C}$ (corresponding to a kinematic viscosity of $66.7 \text{ mm}^2/\text{s}$).

The clearance h at the outer circumference of the rigid and flat slipper pad is given by

$$h = h_c + \alpha R_2 \cos(\theta - \phi) \quad (1)$$

where α and ϕ denote the inclination and azimuth angles of the slipper, respectively, calculated from the normal vector obtained by the measured data of the displacement sensors; h_c is the center clearance, given by the average of the three displacements; θ is the coordinate in the cylindrical system;

and R_2 is the outer radius of the sliding surface of the pad. The coordinates and definition of inclination angle and azimuth of pad are described in Ref. 26.

The temperature rise is evaluated by the difference in temperature Δt defined by

$$\Delta t = t - t_c \quad (2)$$

where t_c is the oil temperature in the chamber and t is the temperature of the slipper pad. In Fig. 4 the symbols are the measured data and the curves are fitted to a sinusoidal temperature distribution hypothesis given by

$$t = a_i + b_i \cos(\theta - c_i) \quad (3)$$

Fig.3 Clearance distributions h under different supply p_s pressures p_s ($N = 26.7 \text{ s}^{-1}$, $t_c = 30^\circ\text{C}$).

Fig.4 Distributions of temperature difference Δt at different supply pressures p_s ($N = 26.7 \text{ s}^{-1}$, $t_c = 30^\circ\text{C}$).

where a_i , b_i , and c_i denote the mean temperature, half the difference between the maximum and minimum temperature, and the circumferential angle at the maximum temperature, respectively.

The slipper was inclined, and the temperature of the pad was circumferentially distributed as shown in Fig. 4. As p_s increased, the clearance h decreased and the overall temperature increased. Under these conditions, the minimum clearance appeared at $\theta(h_{min}) = 90-105^\circ$ ($\phi = 270-285^\circ$), while the maximum temperature obtained from the fitted curves was approximately at $\theta(t_{max}) = 120^\circ$.

Figure 5 compares the circular positions at minimum clearance $\theta(h_{min})$ and maximum temperature $\theta(t_{max})$ for various rotational speeds ($N = 3.3, 13.4$, and 26.7 s^{-1} , with $t_c = 30^\circ\text{C}$). The angles $\theta(t_{max})$ and $\theta(h_{min})$ differ by around $30-50^\circ$, except at $p_s = 14-21 \text{ MPa}$ and below. It should be noted that the pad attitude ϕ was unstable at lower p_s because the load of pushing the pad onto the disk was small.

In other words, the minimum clearance appeared at the trailing edge and the temperature was also maximized near the trailing edge side. In contrast, the clearance was maximized and the temperature was minimized close to the leading edge side. The temperature may have been circumferentially maximized slightly near outer side of the disk beyond the trailing edge because the viscous friction generated by the rapid sliding speed might be significantly larger than the solid friction induced by partial contact. That is, the system operated in fluid film lubrication rather than mixed and boundary lubrication.

Fig.5 Comparison of the circular position at minimum clearance $\theta(h_{min})$ and maximum temperature $\theta(t_{max})$ ($t_c = 30^\circ\text{C}$).

4.2 Effects of operating conditions on clearance shape

Figures 6–8 show contour plots of h_c , α , and ϕ of the slipper pad, respectively. In all plots, the oil temperature t_c is 30°C.

The h_c was estimated by averaging the measured data at three positions. The angle α and azimuth ϕ were also calculated from the clearance sensors' data²⁶⁾. As the p_s increased and the rotating speed N decreased, the clearance h_c decreased. Similarly, the clearance of a sliding bearing was determined by the wedge effect of the hydrodynamic lubrication. In contrast, the angles α and ϕ showed little variation. At speeds N around 10–13.3 s⁻¹ and 20 s⁻¹, h_c , α , and ϕ were all rather singular, perhaps, because the rotational speed N almost coincided with the natural resonance frequency of the tester.

4.3 Effects of operating conditions on temperature rise

Figures 9–11 plot the maximum temperature differences Δt_{max} of the slipper land at oil temperature $t_c = 30, 40$, and 50°C, respectively. The temperature difference Δt_{max} at the top right corner in Fig. 9 exceeded 20°C, and Δt_{max} at the bottom left corner in Figs. 10 and 11 was under zero. The contours of the temperature Δt_{max} undulated around $N = 10\text{--}13.3$ s⁻¹ and $N = 20$ s⁻¹. As discussed for Figs. 6–8, this phenomenon might be attributable to resonance, which yielded instability of the pad attitude and variability of the minimum clearance location. With increasing p_s and rotational speed N , the temperature Δt_{max} increased at all oil temperatures t_c .

Fig.6 Contour plots of center clearance h_c , varying the supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

Fig.7 Contour plots of inclination angle α , varying supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

Fig.8 Contour plots of pad azimuth ϕ , varying the supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

Fig.9 Contour plots of maximum temperature Δt_{max} , varying the supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

A representative example of the minimum temperature Δt_{min} distribution at $t_c = 30^\circ\text{C}$ is given in Fig. 12. As a function of p_s and N , the minimum temperature Δt_{min} followed similar trends to the maximum temperature Δt_{max} .

The rise in temperature of the slipper causes thermal deformation and the difference in pad temperature may cause thermal distortion. Therefore, these features are worthy of further investigation. Figures 13–15 display the contours of the differences δt between the maximum and minimum temperatures of the slipper land, defined by $\delta t = \Delta t_{max} - \Delta t_{min}$ ($= t_{max} - t_{min}$).

The temperature differences δt depended on the rotational speed N , but were independent of p_s at all oil temperature t_c . Since the sliding parts were mainly operated in fluid film lubrication, it is supposed that heat generation was almost

Fig.12 Contour plots of minimum temperature Δt_{min} , varying the supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

Fig.10 Contour plots of maximum temperature Δt_{max} , varying the supply pressure p_s and rotational speed N ($t_c = 40^\circ\text{C}$).

Fig.13 Contour plots of differences in temperature δt , varying the supply pressure p_s and rotational speed N ($t_c = 30^\circ\text{C}$).

Fig.11 Contour plots of maximum temperature Δt_{max} , varying the supply pressure p_s and rotational speed N ($t_c = 50^\circ\text{C}$).

Fig.14 Contour plots of differences in temperature δt , varying the supply pressure p_s and rotational speed N ($t_c = 40^\circ\text{C}$).

Fig.15 Contour plots of differences in temperature δt , varying the supply pressure p_s and rotational speed N ($t_c = 50^\circ\text{C}$).

dominated by the dissipation term of the energy equation, and that solid friction generated by metallic contact at most exerted a negligible effect. Comparing Figs. 13–15, both δt and $\Delta t_{\max}$ were observed to be higher at lower oil temperature ($t_c = 30^\circ\text{C}$) than at higher oil temperature ($t_c = 50^\circ\text{C}$).

5. Conclusions

The temperature and behavior of the slipper pad of swashplate type axial piston motors were measured under real operating conditions up to 35 MPa and 26.7 s^{-1} . The difference in the temperature distributions of the pad was about 5°C and the rise in the maximum temperature was over 20°C under the high pressure and speed conditions. The salient conclusions are summarized below:

- (i) The temperature difference distributions of the slipper pad between the maximum and minimum depended on the rotational speed but were independent of supply pressure.
- (ii) The temperature of the pad was highest near the trailing edge of the outer-leaning disk rim and lowest near the leading edge.
- (iii) The pad temperature increased with increasing supply pressure and rotational speed.
- (iv) As the oil temperature was lowered, the increase in temperature became more pronounced.

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