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Simultaneous temperature measurements of bearing and seal parts of a swash plate type axial piston pump - effects of piston clearance and fluid property †

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Abstract

Temperatures of three bearing and seal parts of a swash plate type axial piston pump were measured simultaneously using a slip ring with thermocouples' amplifiers. The rotating cylinder block had five thermocouples, the swash-plate had four, and the valve-plate had four. Three sets of pistons with different diameters were prepared. Hydraulic mineral oil with VG22 and a water–glycol type hydraulic oil with VG32 were used as test fluids. The maximum discharge pressure was 20 MPa; the maximum rotational speed was 25 rps. The inlet oil temperature was 20-40°C. Temperatures and temperature differences between the bottom-dead-center and the top-dead-center increased concomitantly with the discharge pressure. For larger piston clearance ratios, the temperatures rose monotonically. The bearing and seal parts were markedly hotter than the discharge oil. Temperatures rose much less using the water–glycol type oil than when using the mineral oil.

Keywords: Cylinder block; Fluid power systems; Piston pump; Swash plate; Temperature; Tribology; Valve plate

1. Introduction

Hydraulic pumps are principal components of fluid power systems used for high-pressure sources. Particularly, swash plate type axial piston pumps are expected to be compact, to operate at high pressures and under widely various speed conditions, and to have a long useful life while maintaining high reliability and high efficiency. Higher power density necessitates severe operating conditions for tribological parts of the pumps, resulting in heat generation and seizure in addition to hydraulic oil degradation.

A tool of optimum design and precise estimation including the influence of heat generation and thermal lubrication is needed. The three principal piston pump bearing and seal parts are those between the pistons and the cylinder-block bores, those between the slippers and the swash plate, and those between the valve plate and the cylinder block.

The authors measured the swash plate temperature [1] and the valve plate temperature [2] experimentally using a rotating cylinder block type pump and the cylinder block [1] using a rotating swash plate type pump under actual operating conditions. Pumps of two different types were prepared and measured separately to overcome the difficulties of measuring the rotating parts' temperatures and to maintain data accuracy. The pumps yielded interesting results showing temperatures of respective parts, but the experiment was performed separately, so that conditions did not match completely.

Later, we produced a test pump using a piston pump of a rotating cylinder block type using a slip ring with installed amplifiers for thermocouples. We then measured the cylinder block, swash plate, and valve plate temperatures simultaneously [3]. In this report, using the test pump with the slip ring, we pursued the experiment to examine the effects of piston clearance and the fluid properties on the bearing and seal part temperatures.

2. Experimental apparatus and methods

The hydraulic circuit of the test rig [1] was comprised of test pumps (maximum discharge pressure, 21 MPa; theoretical displacement of 10 ml/rev) with thermocouple amplifiers and a slip ring, a three phase induction motor (7.5 kW), an electric inverter, a strain-gage type torque sensor (20 N•m), flow-rate meters (4 and 2 kl/h), a pressure gage, platinum thermoresistors, thermocouples, valves, an oil-cooler, and a reservoir. Locations of the thermocouples installed in the cylinder block, swash plate, and the valve plate are presented in Fig. 1.

The induction motor drove the test pump through the torque sensor. Two platinum thermoresistors were placed in the suc-

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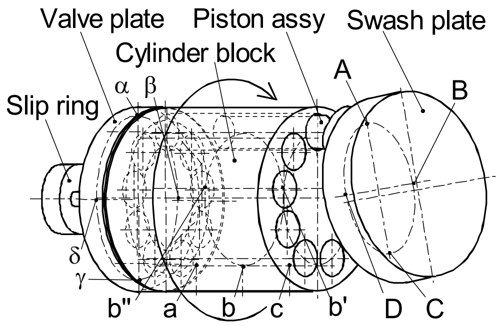

Fig. 1. Schematic of bearing and seal parts of the test pump and locations of the thermocouples.

tion and delivery conduits. The flow meters were installed in discharge and drain lines. The test oils were a mineral oil type hydraulic fluid with viscosity grade of ISO VG22 and a water–glycol type hydraulic fluid with VG32 (50% water content). The respective fluid density and kinematic viscosities of the mineral oil were 866 kg/m^3 and 23/4.4 mm^2/s at 40/100°C; those of the water–glycol oil were 1069 kg/m^3 and 33/7.4 mm^2/s.

The experiment was conducted as follows: the oil temperatures at the test pump inlet and the rotational speed of the pump were set; the discharge pressure was increased from atmospheric pressure to the maximum of 20 MPa by 1 MPa, then decreased to atmospheric pressure by 1 MPa steps. At each setting pressure, the discharge flow rate, drain flow rate, torque, and temperature were measured.

3. Results and discussion

3.1 Effect of piston clearance

Figs. 2 and 3 portray temperature rises of the cylinder block Δt_{CB}, the swash plate Δt_{SP}, the valve plate Δt_{VP}, and the discharge oil Δt_{out} for the mineral oil with VG22, designated as MO22, respectively, for piston clearance ratios ψ=0.004 and 0.002. The temperature rise Δt was defined by the difference in the average temperature of each bearing and seal part from the inlet oil temperature t_{in} of the pump. The temperature t_{in} was carefully maintained as 30±0.1°C. The rotational speed n was set at 25 s^{-1}.

As the discharge pressure p_d increased, all temperature rises Δt_{CB}, Δt_{SP}, Δt_{VP}, and Δt_{out} increased, irrespective of the piston clearance ratios ψ. The increase in Δt for ψ=0.002 increased markedly at high discharge pressure p_d, whereas Δt increased almost linearly for ψ=0.004. In this experiment, Δt_{SP} was largest and Δt_{CB} was smallest, but the influence of the discharge pressure p_d on Δt_{CB} was largest and Δt_{VP} was smallest. The temperature rise Δt_{out} of the discharge oil was markedly smaller than Δt_{CB}, Δt_{SP}, or Δt_{VP} of the bearing and seal parts.

3.2 Effects of fluid properties

Fig. 4 depicts temperature rises Δt_{CB}, Δt_{SP}, Δt_{VP}, and Δt_{out} in the case of the water–glycol type oil with VG32, designated as

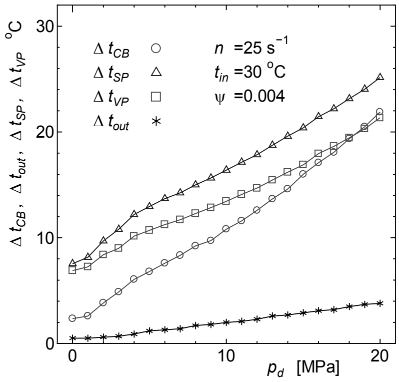

Fig. 2. Temperature rises of the bearing/seal parts Δt_{CB}, Δt_{SP}, and Δt_{VP} and of discharge oil Δt_{out} for the large piston clearance ratio ψ=0.004 (n=25 s^{-1}, t_{in}=30°C, MO22).

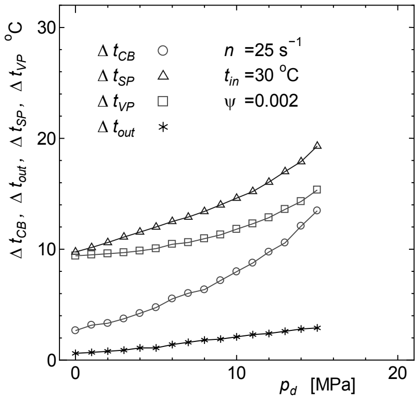

Fig. 3. Temperature rises of the bearing/seal parts Δt_{CB}, Δt_{SP}, and Δt_{VP} and of discharge oil Δt_{out} for small piston clearance ratio ψ=0.002 (n=25 s^{-1}, t_{in}=30°C, MO22).

WG32. In comparison with the mineral oil, although the viscosity grade was higher than that of the mineral oil of VG22, the increase Δt in temperature were markedly smaller. The effect of the discharge pressure p_d on the temperature rise Δt and the differences among the rises Δt_{CB}, Δt_{SP}, and Δt_{VP} were slight.

Fig. 5 shows a comparison of the temperature rises Δt_a and Δt_c at the points 'a' and 'c' of the cylinder block for the mineral oil, MO22 and for the water–glycol oil, WG32. The rise Δt_c at point 'c' corresponding to the bottom-dead-center of the pistons was markedly higher than Δt_a at point 'a' to the top-dead-center for MO22, especially under the condition of high discharge pressure p_d because, at point 'c', the reciprocating action of the piston engendered higher solid friction and larger heat generation; that at point 'a' was cooled by suction of the low-temperature fluid and by delivery of the heated fluid. In contrast to the case of the mineral oil, Δt_a and Δt_c were low and the difference in temperatures for WG32 was less.

Fig. 6 depicts effects of the inlet oil temperature t_{in} on temperature rise Δt for WG32. The temperature rises for the lower inlet temperature (t_{in} =20°C) were higher than those of the higher temperature (t_{in} =40°C).

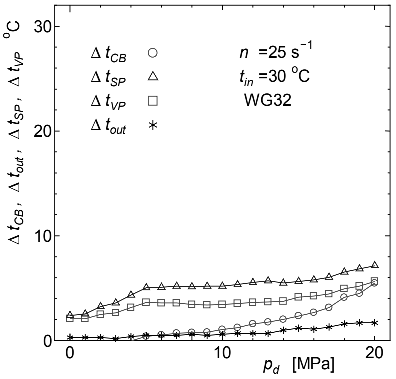

Fig. 4. Temperature rises Δt_{CB}, Δt_{SP}, Δt_{VP}, and Δt_{out} for water–glycol type oil WG32 (n=25 s^{-1}, t_{in}=30°C, ψ=0.003).

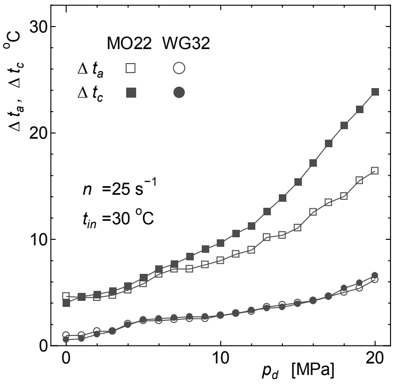

Fig. 5. Comparison of cylinder block temperature rises Δt_a and Δt_c between for MO22 and WG32 (n=25 s^{-1}, t_{in} =30°C, ψ=0.003).

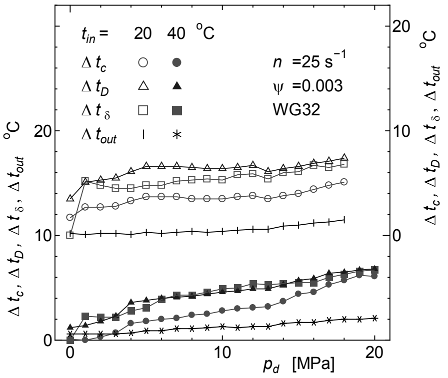

Fig. 6. Temperature rises Δt_c, Δt_D, Δt_δ, and Δt_{out} in terms of inlet temperature t_{in} (n=25 s^{-1}, ψ=0.003, WG32).

4. Conclusions

Using the swash plate type axial piston pumps with the slip ring for thermocouples, temperatures of all three main sliding parts between the swash plate and the slipper, the rotating cylinder block and the pistons, and the valve plate and the cylinder block were measured simultaneously, particularly addressing effects of the piston clearance and the fluid properties. Experimental data supported the following results.

As the discharge pressure increased, the swash plate, cylinder block, and valve plate temperatures increased independently of the piston clearance and fluid properties. Temperatures of all bearing and seal parts were markedly higher than temperatures of discharge oils, irrespective of test conditions.

For larger piston clearance ratios, the temperature rose monotonically; for a smaller ratio, the temperature was higher at high discharge pressure. The effect of the discharge pressure on the cylinder-block temperature rise was large and the effect on the valve-plate temperature rise was small, independent of the piston clearance ratios.

The temperature rises of all bearing and seal parts and the discharge oil using the water–glycol type fluid were noticeably smaller than the rises when using the mineral oil. At lower inlet temperatures, the temperature rise was greater even if the discharge pressure was low.

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Nomenclature

n	:	Rotational speed
p_d	:	Discharge pressure
Q_d	:	Discharge flow rate
t	:	Temperature
t_{in}	:	Inlet oil temperature
t_{out}	:	Inlet oil temperature
Δt	:	Temperature rise = $t - t_{in}$
η	:	Total efficiency
η_v	:	Volumetric efficiency
ψ	:	Piston clearance ratio

subscripts

A, B, C, D	:	Measuring points on the swash plate
a, b, c, d	:	Measuring points in the cylinder block
CB	:	Cylinder block
SP	:	Swash plate
VP	:	Valve plate
α, β, γ, δ	:	Measuring points on the valve plate
0	:	Standard

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