Temperature Measurement of Tribological Parts in Swash-Plate Type Axial Piston Pumps

Toshiharu KAZAMA, Tadamasa TSURUNO, Hayato SASAKI

Abstract: Temperatures of a swash plate, cylinder block, and a valve plate of swash-plate type axial piston pumps with a rotating cylinder block and a rotating swash plate were measured. Thermocouples were embedded underneath these parts. Hydraulic mineral oils with ISO VG22, 32, 46, and 68 and a water–glycol type hydraulic fluid with VG32 were used as test fluids. The maximum discharge pressure was 20 MPa and the maximum rotational speed was 28.3 rps. The inlet oil temperatures were specified as 293–313 K. At the atmospheric pressure to the maximum discharge pressure, the temperatures, flow rates, and the torque were measured. Results support the following conclusions: i) as the discharge pressure increased, the temperatures of the swash plate, cylinder block, and the valve plate increased in almost direct relation; ii) the cylinder block temperature at the bottom dead center of the pistons increased markedly; iii) the temperature increases using the water–glycol fluid were noticeably smaller than the rises using the mineral oils; and iv) the temperature rises became large for higher fluid viscosity and lower inlet oil temperature.

***Keywords**:* Fluid power, Tribology, Axial piston pump, Temperature, Experiment

1 Introduction

Hydraulic pumps and motors are expected to operate under high pressure and under a wide range of speed conditions to be compact, and to have a long useful life while maintaining high reliability and high efficiency. Higher power density forces severe operation at tribological parts of the pumps and motors, resulting in heat generation and seizure.

Dr. Toshiharu Kazama, Professor, Muroran Institute of Technology, Muroran, Hokkaido, Japan, Tadamasa Tsuruno, Graduates, Muroran Institute of Technology (Present address: Engineer, Fuji Techno-Service Co., Ltd., Japan), Hayato Sasaki, Graduates, Muroran Institute of Technology (Present address: Engineer, Suzuki Motor Corporation, Japan)

The need exists for a tool of optimum design and precise estimation including the influence of heat generation and thermal lubrication. For example, Wilson[1)] pointed out that the optimum clearance based on the isothermal theory is insufficient to design displacement pumps.

Swash-plate type axial piston pumps offer high efficiency and high power density. Yamaguchi et al.[2)-4)] experimentally investigated the effects of operation conditions and working fluids on the performance and temperature of an axial piston type test pump, where the thermocouples were installed in the cylinder block. Ivantysynova[5)] and Olems[6)] measured the temperature distributions of the cylinder block around the cylinder bores using the test pump with installed thermocouples in the cylinder block. They have been given the temperature distribution of and compared with the thermohydrodynamic lubrication (THL) analysis. Subsequently, Wieczorek and Ivantysynova [7)] developed simulation software for the swash-plate type axial piston pump. However, the specification of the test pumps and the condition of the experiment differed from those of actual hydraulic pumps.

On the other hand, for large-scale hydrodynamic bearings, many researchers have tackled the subject theoretically using THL theory[8)-9)]. Furthermore, experimental studies of journal bearings have been performed by Mitsui et al.[10)], Ferron et al.[11)], Gethin and Medwell[12)], and Wang et al.[13)]; experimental studies of thrust bearings have been performed by Horner et al.[14)] and Fillon et al.[15)].

Kazama et al. quantitatively examined the thermohydrodynamic performance of circular pad hydrostatic thrust bearings[16)] including the effect of the changes in physical properties

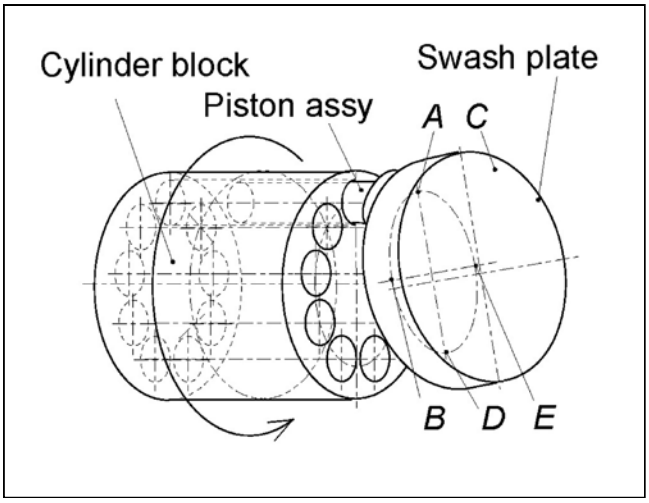

Figure1. *Location of thermo-couples installed in the swash plate (rotating cylinder-block type piston pump)*

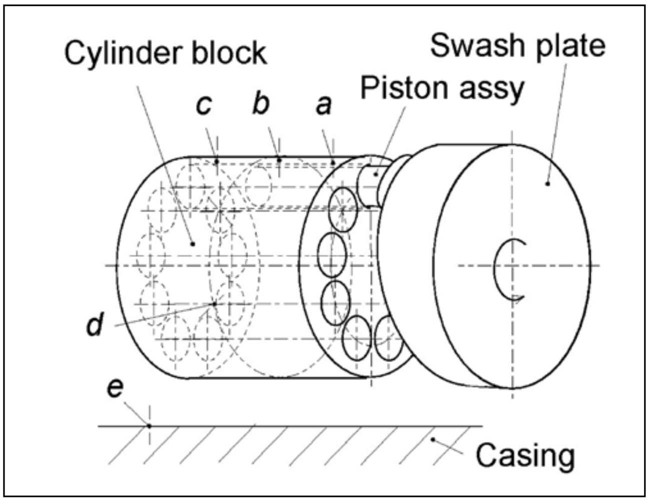

Figure 2. *Location of thermo-couples installed in the cylinder block and casing (rotating swash-plate type piston pump)*

of fluids as functions of temperature and pressure. Later, the authors experimentally measured the temperature of the swash plate and cylinder block of the piston pumps [17] under actual operating conditions. In this report, the temperatures of the valve plate as well as the swash plate and cylinder block of the piston pumps were measured. The results were compared and discussed in detail.

2 Experimental Apparatus and Methods

The hydraulic circuit of the test rig[17] consisted of test pumps (a rotating type cylinder block and swash-plate rotating type axial piston pumps, with maximum discharge pressure of 21 MPa, and theoretical displacement of 10 ml/rev), a three phase induction motor (7.5 kW), an electric inverter, a strain-gage type torque sensor (20 N•m), flow-rate meters (4000 and 2000 l/h), a pressure transducer, thermistors, thermocouples, valves, an oil-cooler, and a reservoir. The locations of the thermocouples installed in the swash plate, cylinder block and the valve plate are illustrated respectively in *Figures 1–3*. *Figures 1* and *3* depict the rotating cylinder block type pump; *Figure 2* is the rotating swash-plate type pump. Pumps of two types were prepared with thermocouples installed in the stationary parts of each pump.

The induction motor drove the test pump through the torque sensor. Thermistors were placed at the pump inlet and the flow meters were installed in the discharge line and the drain line. The test oils were mineral oil type hydraulic fluids with ISO VG22, 32, 46 and 68 (designated as MO22, MO32, MO46 and MO68 respectively) as well as a water–glycol type hydraulic fluid with ISO VG32 (50% water content, WG32). The fluid densities were 866, 869, 872, 875, and 1069 kg/m^3; the kinematic viscosities at 40/100°C were 23/4.4, 33/5.5, 46/6.9, 68/8.7, 33/7.4 mm^2/s, respectively.

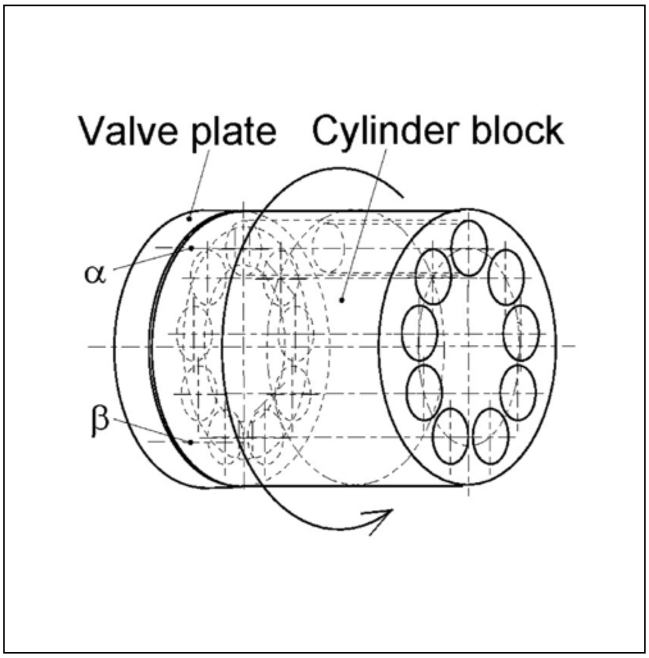

Figure 3. *Location of thermo-couples installed in the valve plate (rotating cylinder-block type piston pump)*

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

3. Results and Discussion

3.1 Swash plate temperature

Figures 4 and *5* respectively depict the pump performance curve and the swash plate temperature using the rotating cylinder-block type test pump. In the performance curve of Figure 4, it is readily apparent that the repeatability was good.

In Figure 5 the temperatures t_A

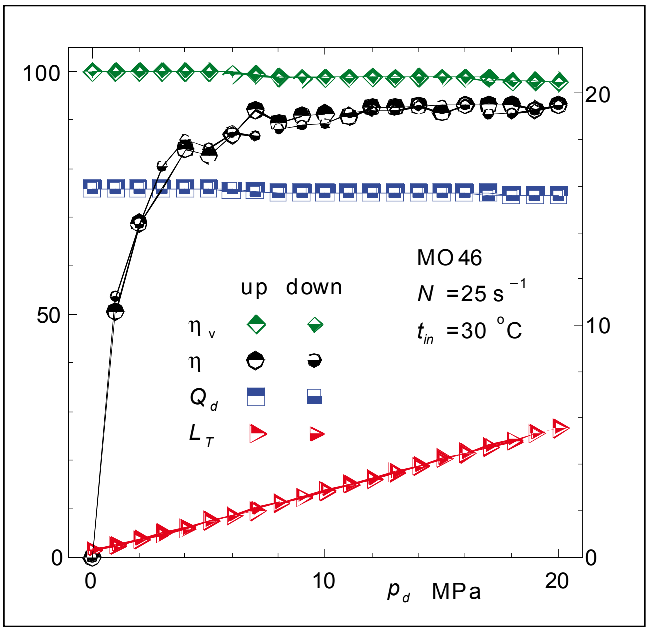

Figure 4. *Pump performance curve (rotating cylinder-block type, MO46, N=25 s^{-1}, t_{in}=30 °C)*

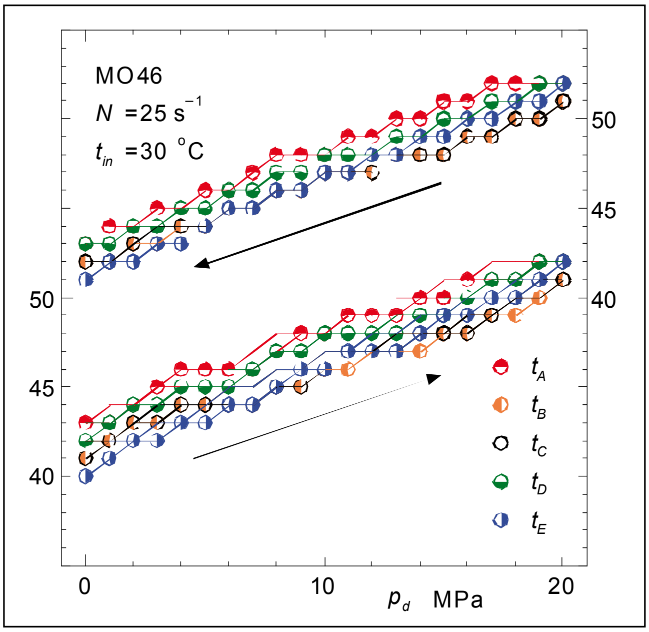

Figure 5. *Temperatures of the swash plate (rotating cylinder-block type, MO46, N=25 s^{-1}, t_{in}=30 °C)*

through t_E increased in almost direct relation to the discharge pressure p_d. The temperature t_A at the measuring point of the swash plate was highest; it corresponded to the trapping part between the crescent-shaped discharge and suction ports.

Figure 6 depicts effects of oil types on the temperature rise Δt_A of the swash plate at point 'A' [17], where the rise $\Delta t_A = t_A - t_{in}$ was defined. The tendency of the temperature rise Δt_A through Δt_E was similar. For all oils tested, the rise Δt_A increased as the discharge pressure p_d increased. The higher the oil viscosity grade, the higher the rise of Δt_A.

It is noteworthy that, from the atmospheric pressure to the maximum discharge pressure p_d=20 MPa, the rise Δt_A using the water–glycol hydraulic fluid (WG32) was only 13°C; it was lowest among the oils tested, but the rise Δt_A using MO68 was greater than 30°C.

Figure 7 portrays the effect of the rotational speed *N* on the swash plate temperature. In that figure, the lines are a guide to the reader's eye. As speed *N* increased, the temperature rise increased because of the viscous dissipation in the fluid film and frictional heating in metallic contact.

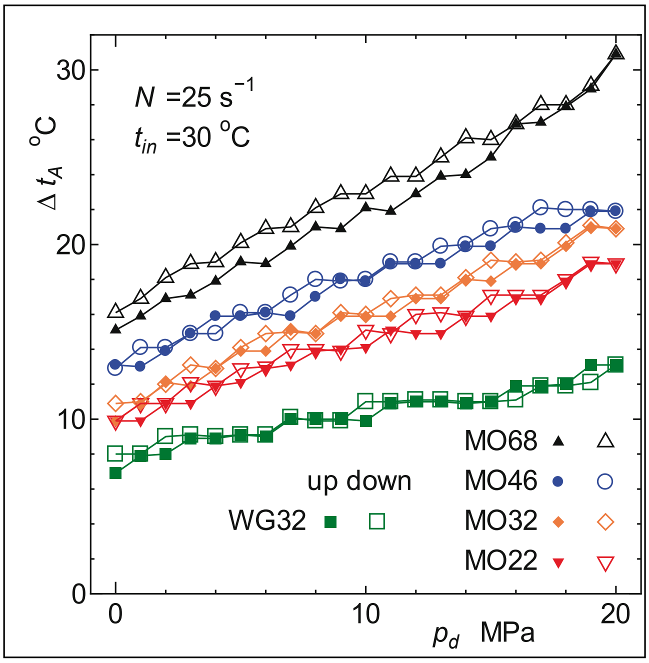

Figure 6. *Comparison of swash-plate temperature rise Δt_A for test oils (rotating cylinder-block type, N=25 s^{-1}, t_{in}=30 °C)*

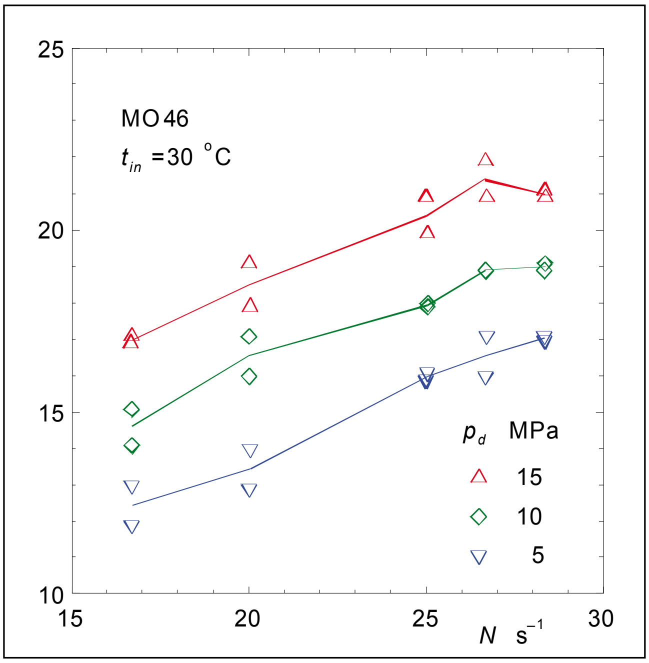

Figure 7. *Effect of rotational speed N on swash plate temperature rise Δt_A (MO46, t_{in}=30 °C)*

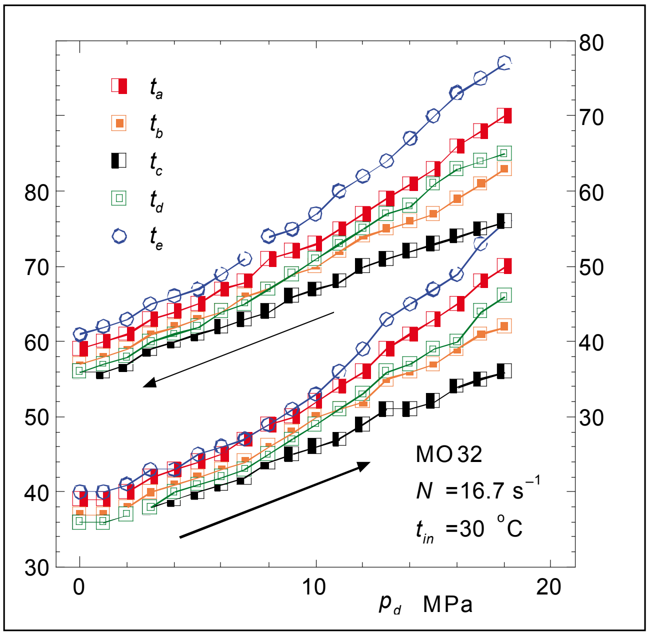

Figure 8. *Temperature t of the cylinder block (rotating swash-plate type, MO32, N=16.7 s^{-1}, t_{in}=30°C)*

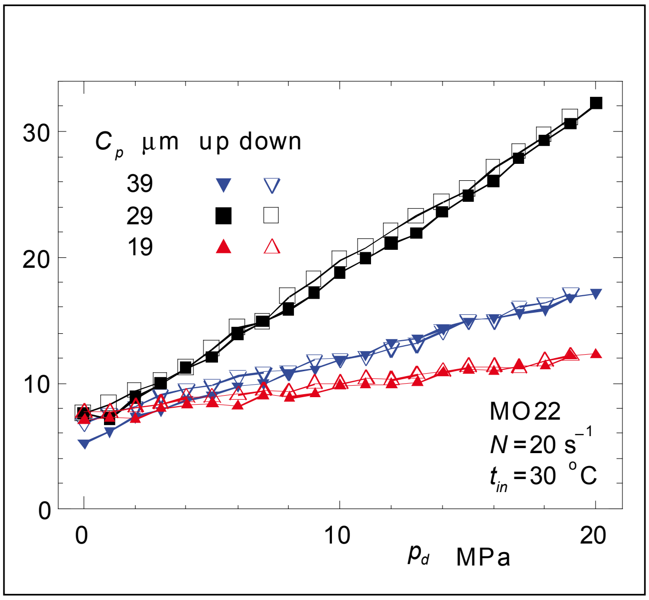

Figure 9. *Effect of piston clearance C_p on the mean cylinder block temperature rise $\Delta t_{m\,CB}$ (rotating swash-plate type, MO22, N=20 s^{-1}, t_{in}=30°C)*

3.2 Cylinder block temperature

The rotating swash-plate type axial piston pump was prepared to measure the temperature around the cylinder bores of the cylinder block.

Figure 8 depicts temperatures at points 'a' – 'e' presented in Fig. 2 [17]. Comparing the temperatures t_a, t_b, and t_c on the axial direction of the cylinder bore, t_a was highest and t_c was lowest. Measuring point 'a' was located near the edge of the cylinder bore: the bottom dead-center of the piston. The piston was acted on by the moment-load. Therefore, the piston inclined in the bore and locally contacted at the edge of the bore. The reciprocating action of the piston results in higher solid friction and larger heat generation.

On the other hand, point 'c' was corresponding to the top dead-center of the piston. The part around point 'c' was cooled by suction of the low-temperature fluid and by delivery of the heated fluid.

Figure 9 depicts the effect of the clearance between the piston and the cylinder bore on the mean temperature, which rises $\Delta t_{m\,CB}$ (= (Δt_a + Δt_b + Δt_c) / 3). When the clearance was small (C_p = 19 µm, average) the rise in $\Delta t_{m\,CB}$ was low, most probably because the inclination of the piston in the cylinder bore was suppressed and frictional heating caused by the metallic contact was low.

3.3 Valve plate temperature

Temperatures t_α and t_β of the valve plate between the delivery to suction ports and the suction to delivery ports were measured respectively using the rotating cylinder-block type axial piston pump. As presented in *Figure 10*, temperatures t_α and t_β increased larger than the discharge temperature t_d. Even if the temperature t_d elevated only 4°C from the inlet temperature t_{in}=30 °C, the temperatures t_α and t_β rose higher and became greater than 20 °C. The difference in temperatures t_α and t_β was not clearly shown.

Figure 11 shows effects of the inlet temperature t_{in} of the hydraulic fluid on the mean valve plate temperature rise $\Delta t_{m\,VP}$, where $\Delta t_{m\,VP}$ was defined as Δt_m =(t_α + t_β)/2 – t_{in}.

As the temperature t_{in} decreased, the temperature rise $\Delta t_{m\,VP}$ was higher because the viscosity was higher at the lower temperature, which yielded the higher viscous dissipation in the film of the bearing and sealing part.

Figure 12 illustrates the effect of the rotational speed N on the temperature rise $\Delta t_{m\,VP}$ of the valve plate. From comparison to Fig. 7 of the swash-plate temperature rise, it is readily apparent that the speed

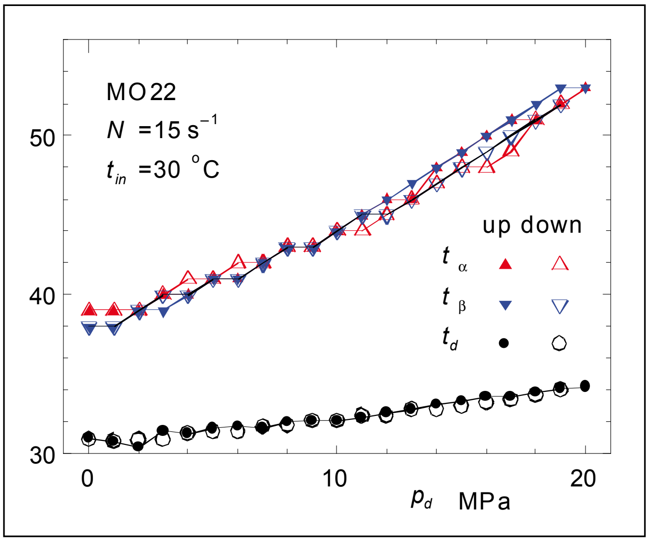

Figure 10. *Valve plate temperatures t_α, t_β and discharge oil temperature t_d (rotating cylinder-block type, MO22, N=15 s^{-1}, t_{in}=30 °C)*

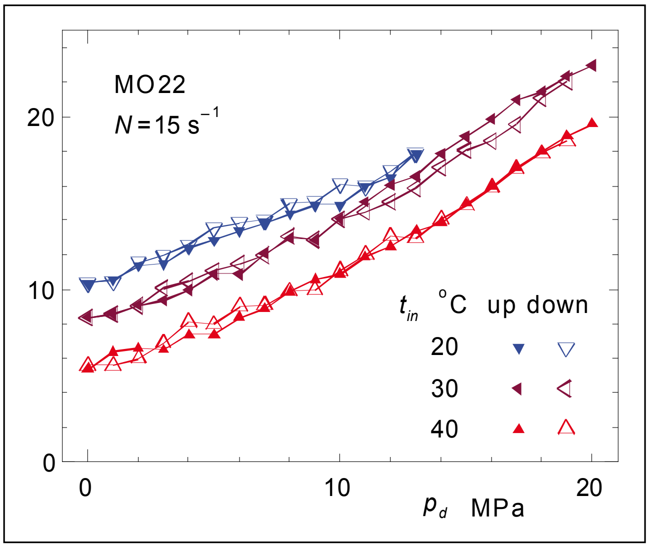

Figure 11. *Effect of inlet oil temperature t_{in} on valve-plate temperature rise $\Delta t_{m\ VP}$ (rotating cylinder-block type, MO22, N=15 s^{-1})*

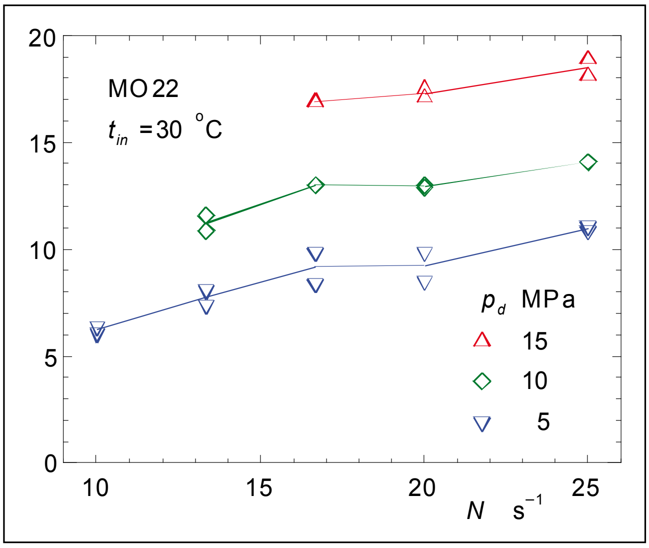

Figure 12. *Effect of rotational speed N on valve-plate temperature rise $\Delta t_{m\ VP}$ (rotating cylinder-block type, MO22, t_{in}=30 °C)*

N less affected the rise $\Delta t_{m\ VP}$ than $\Delta t_{m\ SP}$.

The slippers run on the swash plate and would operate in lightly contacting mixed lubrication because the hydrodynamic action and hydrostatic action was able to support the load effectively, while the sliding parts between the valve plate and the cylinder block were strongly contacted and operated perfectly in mixed lubrication.

4 Concluding Remarks

Using both the rotating cylinder-block type and rotating swash-plate type axial piston pumps, the temperatures of all three main sliding parts between the swash plate and the slipper, the cylinder block and the pistons, and the valve plate and the cylinder block were measured: the pump performance was evaluated. The viscosity grade of the hydraulic fluids, type of fluid, inlet fluid temperature, discharge pressure, rotational speed, and piston clearance were selected as parameters, and the thermal lubrication characteristics of the pumps were examined experimentally under field operating conditions. The conclusions of this experiment are summarized as the following:

As the discharge pressure increased, the temperature of the swash plate, cylinder block and the valve plate increased almost in direct relation. As the rotational speed increased, the temperature rises were dependent on operating conditions.

The swash plate temperature at the switching parts corresponding to discharge and suction increased greatly. The cylinder block temperature at the bottom dead center of the pistons increased markedly. The valve plate temperatures at both the switching parts were almost the same. The sliding part temperature was higher than the discharge oil temperature.

The temperature rise using the water–glycol fluid was noticeably smaller than the increases achieved using mineral oils. The temperature increases became large as the fluid viscosity increased and the inlet oil temperature decreased.

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Nomenclature

N	:	rotational speed
p_d	:	discharge pressure
Q_d	:	discharge flow rate
t	:	temperature
t_d	:	discharge oil temperture
t_{in}	:	inlet oil temperature
Δt	:	temperature rise = $t - t_{in}$
η	:	total efficiency
η_v	:	volumetric efficiency

Subscript

A, B, C, D, E : temperature measuring points on the swash plate
a, b, c, d, e : temperature measuring points in the cylinder block

CB	:	cylinder block
m	:	average
SP	:	swash plate
VP	:	valve plate
α, β	:	temperature measuring points on the valve plate
0	:	standard

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Merjenje temperature triboloških elementov v aksialni batni črpalki z nagibno ploščo

Razširjeni povzetek

Kot je že iz naslova razvidno, prispevek podaja in analizira predvsem meritve temperature znotraj aksialne batne črpalke z nagibno ploščo, *slika 4* pa prikazuje tudi volumetrični in skupni izkoristek merjene črpalke v odvisnosti od izstopnega tlaka. Ob izvajanju preizkusa z vrtilno frekvenco črpalke 1500 vrt./min je znašala temperatura mineralnega olja 30 ^{0}C na vstopu v črpalko. To je bilo olje po ISO VG 46. Strokovnjaki iz prakse se pogosto premalo zavedajo, da je skupni izkoristek tovrstnih črpalk približno do 60 bar, včasih celo do cca 100 bar, zelo slab. To je na sliki 4 dobro razvidno.

V raziskavi so bile merjene temperature na različnih mestih nagibne plošče (*slika 1*), bobna (cilindrskega bloka) (*slika 2*) in ventilsko-razdelilne plošče, to je plošče z vtočno in iztočno izvrtino (*slika 3*). Termopari so bili vgrajeni pod drsnimi površinami teh elementov na mestih, ki so označena na navedenih slikah. Kot hidravlične kapljevine so bile pri preizkusih uporabljene 4 vrste mineralnih hidravličnih olj, in sicer po ISO VG 22, 32, 46 in 68, ter vodni glikol VG 32. Vtočne temperature kapljevin v preizkušane črpalke so bile od 20 do 40 ^{0}C, najvišja vrtilna frekvenca pa 1700 vrt./min. Med izvajanjem meritev so tlak dvigali po 10 bar, začenši pri atmosferskem tlaku, pa vse do 200 bar in analogno zniževali v obratni smeri, kar je razvidno tudi iz podanih diagramov.

Na *sliki 6* velja posebej opozoriti na razmeroma visok dvig temperature za olje VG 68, ki se sicer redko uporablja v sistemih pogonsko-krmilne hidravlike. Vodni glikol VG 32 je v tem pogledu ugoden, ima pa nekaj drugih slabih lastnosti, ki jih številni strokovnjaki iz prakse dobro poznajo.

Na *sliki 8* velja opozoriti na razmeroma visok porast temperature v bobnu; to je za do 50 in celo skoraj do 60 ^{0}C, in to ob razmeroma nizkem tlaku do 180 bar in vrtilni frekvenci samo 1000 vrt./min ter vtočni temperaturi olja 30 ^{0}C. Ta je v praksi pogosto 20 do 30 ali celo 40 ^{0}C višja, višji pa so običajno tudi vrtljaji in tlaki.

V zaključkih prispevka je treba posebej poudariti navedbo avtorjev, da povečevanje tlaka na iztoku iz črpalke vpliva na skoraj sorazmerno zviševanje temperature nagibne plošče, bobna in ventilske plošče. Vpliv zviševanja vrtilne hitrosti na te temperature pa je odvisen od obratovalnih pogojev. Temperature zgoraj navedenih treh elementov so višje od temperatur kapljevine na iztoku iz črpalke (kar je tudi pričakovano, saj kapljevina odnaša s seboj toploto in ima za to dodane aditive).

Ključne besede: fluidna tehnika, tribologija, aksialna batna črpalka, temperatura, preizkus

Ocena in mnenje o vsebini članka z naslovom:
Temperature Measurement of Tribological Parts in Swash-Plate Type Axial Piston Pumps

V sodobni pogonsko-krmilni hidravliki (PKH) se vse več uporabljajo aksialne batne črpalke, ki, za razliko od večine ostalih tipov, zmorejo visoke tlake, večinoma do 350 bar. Večina krmilja PKH je prav tako razvita za ta nivo tlaka. Aksialne batne črpalke z nagibno ploščo so (po moji oceni) uporabljene pogosteje kot aksialne batne črpalke z nagibnim bobnom. Zato je takšen prispevek v naši reviji toliko bolj dobrodošel.

Žal pa so avtorji tega prispevka izvajali nekatere meritve pri parametrih, ki so nižji ali celo znatno nižji od tistih, ki nastopajo v zadnjih letih kot delovni parametri v sistemih PKH v industriji in gospodarstvu na sploh. Predvsem obravnavani najvišji tlak preskušanja 200 bar je znatno pod nivojem tlaka, ki nastopa v industrijskih strojih in postrojenjih; to je do 350 bar (v železarstvu v zadnjem desetletju večinoma 290 do 320 bar). Pri mobilnih strojih, pri katerih so črpalke gnane večinoma z dieselskimi motorji v neposrednem prenosu, so vrtilne frekvence tudi znatno nad 2000 vrt./min. Preizkusi v tem prispevku navajajo najvišje vrtljaje 1700 vrt./min, večina pa je izvedena z do 1500 vrt./min. Postavlja se vprašanje, ali lahko za oceno razmer znotraj črpalk v tovrstni industrijski praksi vrednosti oziroma parametre, ki so v tem prispevku izmerjeni, ekstrapoliramo tako glede »naših« dejanskih vrtljajev, tlakov, vtočnih temperatur kapljevin ipd.

Doc. dr.Jožef Pezdirnik, univ. dipl. inž.
Univerza v Ljubljani, Fakulteta za strojništvo