1. Introduction

Hydraulic pumps and motors are transformers between mechanical energy and pressure, which are the key components of fluid power systems (JHPS, 1989). The pumps and motors are positive-displacement machines, which roughly classified into three groups: vane-, piston-, and gear-types. The reliability and efficiency of the equipment are greatly dependent on the tribological behaviors of the sliding parts (Yamaguchi, 1986), particularly under severe operating conditions.

Hydraulic vane pumps are widely used in industry (ASTM, 2008) and can be broadly categorized into constant-volume and variable-volume pumps, as well as balanced-pressure and unbalanced-pressure pumps. The vanes are inserted into the slots of the rotor and the tips of the vanes slide on the inner surface of the cam ring as the rotor moves. The output of the pumps can be changed by altering the geometry of the displacement chambers, which consist of vanes, a cam ring, side-plates, and a rotor. Each chamber has bearing and sealing parts between the vane-tips and the cam ring-surface, between the vane side surfaces and the side-plates, and between the vane surfaces and the rotor slots.

These bearing and sealing parts, especially those between the vane tips and the cam ring, should be under an appropriate load in order to maintain the seal, and a number of mechanisms, such as single vanes, dual vanes, intra-vanes, and spring-loaded vanes, have been proposed in order to satisfy this requirement. Under a specific condition, especially in the case of single-vane pumps, the tip of the vane could detach from the inner surface of the cam ring because the forces became unbalanced, due potentially to, e.g., excessive pre-compression of the pump chamber at the entrapment, a decrease in the back pressure on the vanes, and a reduction in the centrifugal force of the vanes at low rotational speed. The tip subsequently and immediately reattaches to the cam ring, thus, violent and repeated detachment and reattachment occur, which reduces the efficiency and durability of the pump and leads to the
generation of vibration and noise and eventually catastrophic failure due to wear or seizure. Therefore, this violent
detachment and reattachment behavior should be avoided in order to ensure high degrees of reliability, durability, and
efficiency.


In recent years, vane tip detachment rarely occurs within specifications. However, the detachment may occur under special conditions and out of the specification. The detection of the sign and the development of the detectors are beneficial for users and important for manufacturers. In the present study, experiments were conducted in order to investigate vane detachment, cam ring vibration, sliding-part temperature, and pump performance. Furthermore, the relationship between the vane detachment and the temperature change is discussed and the possibility of a detection in a method other than measurement of noise (Ueno and Okajima, 1986), monitoring of flow rate (JHPS, 1989), and visualization of the vanes (Nishiumi and Maeda, 1993) is explored.

2. Nomenclature

\[ a \quad \text{Vibration acceleration of the cam ring} \]
\[ L_i \quad \text{Shaft input power} \]
\[ N \quad \text{Rotational speed} \]
\[ p_d \quad \text{Discharge pressure} \]
\[ Q_d \quad \text{Discharge flow rate} \]
\[ t \quad \text{Temperature} \]
\[ t_C \quad \text{Cam ring temperature} \]
\[ t_{in} \quad \text{Oil temperature at pump inlet} \]
\[ t_S \quad \text{Side-plate temperature} \]
\[ \Delta t \quad \text{Temperature difference, } t - t_{in} \]
\[ \eta \quad \text{Total efficiency} \]
\[ \eta_v \quad \text{Volumetric efficiency} \]
\[ \tau \quad \text{Time} \]

Subscripts

\[ C_i \quad \text{Temperature measurement points in the cam ring (i = 3, 5, 6, 8)} \]
\[ S_i \quad \text{Temperature measurement points in the side plate (i = 1, 3, 6, 8)} \]
\[ vd \quad \text{Vane detachment} \]

3. Experimental apparatus and test conditions

Figure 1 shows the hydraulic circuit of the tester (Kazama and Narita, 2011; Kazama, 2013, 2015). The test vane pump was mounted on the test bench, which consisted primarily of the hydraulic circuit, including valves, filters, an oil-cooler, and a reservoir. The power source included a three-phase induction motor (7.5 kW) and an electric inverter; the sensors included a torque sensor (5 N·m), a tachometer, a flow-rate meter (4 kl/h), a pressure gauge (Bourdon tube), a pressure sensor (strain gauge), and thermometer sensors (Pt100). The test oils were mineral oil-type hydraulic fluids with International Standards Organization Viscosity Grade (ISO VG) of 32 (32.9 mm²/s at 40°C and 6.37 mm²/s at
100°C).

Figure 2 shows a schematic of the test pump core and the locations of the sensors. The pump was a compact, pressure-balanced, hydraulic vane pump with a theoretical displacement of 9.4 ml/rev, a maximum discharge pressure of 21 MPa, and an allowable rotational speed of 12.5 to 30 s⁻¹, which was designed specifically for this test. The cam-ring curves were formed in accordance with the combination of circular and polynomial curves, where the major axis was 49 mm and the minor axis was 46 mm. The discharge and suction ports (two each) were arranged because of the balanced-pressure pump; the discharge ports were located in the region between bottom dead center (B.D.C) and top dead center (T.D.C) of the vanes inserted into the rotor and the suction ports were located in the region between T.D.C. and B.D.C. The angles of the suction ports were 20–70° and 200–250° and the angles of the discharge ports were 113–157° and 293–337°. The pump had 12 vanes (height: 8 mm, width: 22 mm, and thickness: 1.8 mm), which were inserted front to back into the slits (rotated 180° from their original position) of the rotors (diameter: 45.5 mm) so as to allow the vanes to detach easily. The rotor with the vanes was rotated counter-clockwise.

One piezoelectric acceleration pickup (frequency range in ±5%: 1 to 10 kHz, resonant frequency: 70 kHz and over) was affixed to the cam ring close to the suction port. Four thermocouples (K-type) were embedded in the cam ring, and four additional thermocouples were embedded in the side-plate of the test pump, as shown in Fig. 2, where the non-through holes were bored from the back surfaces of the ring and the plate and the distance between the bottom face of the holes and the sliding surfaces was about 1.5 mm. On the basis of the lower left T.D.C in Fig. 2 the position angles of the thermocouples C7, C6, C5, and C8 were 45°, 75°, 150°, and 345°; the angles of S3, S2, S1, and S4 were 75°, 150°, 195°, and 345°, respectively. The angle of the acceleration pickup was 315°. In other words, the locations of the thermocouples in the cam ring and the side-plate just after the discharge port are labeled C3 and S3, respectively. Similarly, the locations just after the suction port are C6 and S6 and the locations at the middle of the discharge port are C8 and S8. Additionally, the location in the cam ring at the middle of the suction port is C5 and that in the side-plate just before the suction port is S1.

The temperature at the inlet port of the pump, $t_{in}$, was maintained at 40, 50, or 60°C. The value of $t_{in}$ was set by means of a warm-up operation. For a test condition of the speed $N$ between 6.6 and 15 s⁻¹, $p_d$ was increased from atmospheric pressure to the maximum discharge pressure of 10 MPa in 1 MPa increments. The test conditions of the speed and the pressure were rather lower than real operating conditions because of the vanes to be detached and the limitation of the experimental environment. If vane detachment occurred, the experiment was terminated for safety considerations. The value of $N$ was adjusted continuously with an electric inverter, and $t_{in}$ was controlled manually using an oil cooler during the test. The speed $N$ and the temperature $t_{in}$ were maintained within ±0.05 s⁻¹ and ±1°C, respectively, before detachment. At each pressure setting, the values of $a$ and $t$ were recorded by a logger, and the discharge flow-rate was measured and the shaft torque was monitored.
4. Results and discussion

Figure 3 shows the pump performance results, including $Q_d$, $L_t$, $\eta_v$, and $\eta$. Figures 4a and 4b show the temperatures of the cam ring and the side-plate, respectively, measured during the experiment. The values of $t_{in}$ and $N$ were set to 60$^\circ$C and 10 s$^{-1}$, respectively, where the vanes detached from the cam ring at $p_d$ values of 5 to 6 MPa as the pressure was increased.

As $p_d$ was increased, $Q_d$ decreased and $L_t$ increased, while $\eta_v$ and $\eta$ decreased monotonically. The efficiencies $\eta_v$ and $\eta$ of the test pump were lower than those of a commercial pump because the test pump was modified. In response to $p_d$, the temperature differences of the cam ring, $\Delta t_C$, and the side-plate, $\Delta t_S$, from the inlet oil temperature, $t_{in}$, increased. The changes in temperatures $t_C$ and $t_S$ are indicated by $\Delta t_C$ and $\Delta t_S$, respectively, where $\Delta t$ is defined by $\Delta t = t - t_{in}$, because $\Delta t$ is more clear and precise to evaluate the temperature changes (Kazama, 2013, 2015).

The change in temperature of the cam ring at C5, $\Delta t_{C5}$, was the greatest, followed by $\Delta t_{C6}$, $\Delta t_{C3}$, and $\Delta t_{C8}$, as shown in Fig. 4a. Furthermore, as shown in Fig. 4a, the change in temperature of the side-plate at S3, $\Delta t_{S3}$, was the greatest, followed by $\Delta t_{S8}$, $\Delta t_{S1}$, and $\Delta t_{S6}$. As indicated by Figs. 4a and 4b, $\Delta t_C$ was larger than $\Delta t_S$ over the entire range of $p_d$. For the cam ring, $\Delta t_{C6}$ near the suction port was larger than $\Delta t_{C3}$ and $\Delta t_{C8}$ near the discharge port. In contrast, for the side-plate, $\Delta t_{S3}$ and $\Delta t_{S8}$ near the suction port were larger than $\Delta t_{S6}$ near the discharge port. This is because, near the suction ports, the cam ring was heated by friction between the vane tip and the cam ring surface as a result of high contact pressure, whereas the side-plates were cooled by suction oil at low temperature.

When vane detachment began to occur, the vibration of the cam ring and the noise of the pump became significant, and the flow rate became zero. This occurred because the vanes repeatedly jumped from and impacted with the cam ring and the pump could not discharge oil owing to the large gap between the vane tips and the cam ring surface. The experiment was terminated to avoid severe wear and seizure of the pump and out of safety concerns. Therefore, with the exception of $\Delta t_{C5}$ (which was measured as a reference), no data were obtained for $p_d \geq 6$ MPa.
Fig. 3  Pump efficiency under the vane-detachment condition ($N = 10$ s$^{-1}$ and $t_{in} = 60^\circ$C)

Fig. 4a  Temperature difference, $\Delta t_C$, of the cam ring under the vane-detachment condition ($N = 10$ s$^{-1}$ and $t_{in} = 60^\circ$C)
Fig. 4b  Temperature difference, $\Delta t_S$, of the side-plate under the vane-detachment condition ($N = 10$ s$^{-1}$ and $t_{in} = 60^\circ$C)

Figure 5 plots the pressure, $p_{d, vd}$, when the vanes became detached as a function of $N$ for various values of $t_{in}$. The pressure $p_{d, vd}$ was defined by the pressure when the large metallic noise generated abruptly and the discharge flow rate dropped suddenly during a rise in the discharge pressure. The pump could not be operated at $N < 6.7$ s$^{-1}$ because of the limitations of the electric inverter and motor, and vane detachment did not occur at $N > 15$ s$^{-1}$ for $p_d = 10$ MPa and $t_{in} = 40^\circ$C. Moreover, the experiment could not be performed at $t_{in} > 60^\circ$C because of the specifications of the apparatus and sensors, and detachment did not occur at $t_{in} < 40^\circ$C for $p_d = 10$ MPa and $N = 10$ s$^{-1}$. The data in Fig. 5 are approximate because the pump operation under the vane detachment conditions was unstable. The pressure $p_{d, vd}$ decreased as $N$ decreased and $t_{in}$ increased, where $p_{d, vd}$ was approximately inversely proportional to $N$, as shown in Fig. 5, although the dispersion of $p_{d, vd}$ was about 1 MPa. That is, the vanes easily detached under low rotational speeds and high oil temperatures.

Fig. 5  Discharge pressure, $p_{d, vd}$, plotted with respect to rotational speed, $N$, and oil temperature, $t_{in}$
Vane detachment is principally caused by unbalanced forces, where the forces pulling the vanes away from the cam ring exceeds the forces pushing the vanes toward the cam ring. Such forces include the hydrodynamic pressure at the vane tip considering lubrication, the fluid pressure in the chambers, the back pressure acting on the vane ends, the centrifugal force acting on the vane body, and the friction acting on the vane surfaces. The hydrodynamic pressure includes the EHL effect; the fluid pressure includes pre-compression in the chambers, and the back pressure includes time-lag. The vane tips do not follow the cam ring completely because the cam ring curve was not created by circular curves but also polynomial curves. Regarding vane detachment close to the suction ports, a decrease in the centrifugal force and a reduction in viscous damping would greatly affect the detachment and the vibration. In other words, as the rotational speed decreases, the centrifugal force of the vanes essentially decreases in proportion to $N^2$. As the oil temperature increases, the viscosity decreases and the damping effect may decrease.

Figure 6a shows a plot of $a$ vs. $p_d$ for $t_{in} = 40–60^\circ$C and $N = 10–15$ s$^{-1}$. During the period in which the vanes did not detach, the acceleration increased slightly as $p_d$ increased. However, when the vanes began to detach, the acceleration rapidly increased, while metallic impacting noise was generated. As the pump inlet oil temperature increased, the pressure at which the vanes detached decreased. Note that the acceleration could not be recorded for the case in which $t_{in} = 60^\circ$C and $N = 10$ s$^{-1}$ because the test was terminated, and the data for $a$ for the cases in which $t_{in} = 50^\circ$C and $N = 10$ s$^{-1}$, $t_{in} = 50^\circ$C and $N = 15$ s$^{-1}$, and $t_{in} = 40^\circ$C and $N = 10$ s$^{-1}$ were used as reference data because the pump operation was unstable under these conditions. Heat generation mainly owing to solid friction and repeatable detachment of the vanes are schematically illustrated in Fig. 6b.

![Fig. 6a Vibration acceleration, $a$, of the cam ring for various experimental conditions](image-url)
Figure 7 shows the time evolution of \( a \), Fig. 8 shows the temperatures of the cam ring, \( t_c \), and Fig. 9 shows the discharge pressure \( p_d \) and the input torque \( T \), during the period of closing the valve at the discharge port of the pump approaching the condition of vane detachment. Around \( \tau = 0 \), the valve was controlled manually by closing the valve slightly in order to set \( p_d \) to 8 to 9 MPa. Before detachment (approximately, \( \tau < 0 \) s in Figs. 7 through 9), the temperatures \( t_{C5} \) (at the middle of the suction port), \( t_{C6} \) (just after the suction port), \( t_{C3} \) (just after the discharge port), \( t_{C8} \) (at the middle of the discharge port) were low as shown in Fig. 8 and the change in the acceleration \( a \) was small in Fig. 7. Just after detachment (\( \tau \approx 1 \) to 2 s), \( t_{C6} \) and \( t_{C5} \) decreased for an instant, while \( t_{C3} \) and \( t_{C8} \) exhibited no change. Subsequently, all of \( t_{C3} \), \( t_{C6} \), \( t_{C3} \) and \( t_{C8} \) increased markedly (\( \tau > 2 \) s). At the same time, \( a \) increased, and the vibration and noise became large (\( \tau > 0 \) s). Concurrently, in Fig. 9 the pressure \( p_d \) decreased gradually and the torque \( T \) increased to some extent.

This behavior can be explained by the contact situation between the vane tip and the cam ring surface. Namely, during the initial stage of vane detachment, jumping of the vanes contributes to the decrease in frictional heating due to contact and to the increase in oil cooling by intermittent leaking, resulting in a decrease in \( t_{C6} \). Although \( t_{C6} \) decreased, this decrease was relatively small because the vane jumping occurred near the suction ports rather than near the discharge ports. After the start of vane detachment, however, the vanes began to repeatedly and violently jump and collide with the cam ring, which resulted in heat generation due primarily to solid-contact friction. The heat increased the oil temperature and thus decreased the viscous damping effect, resulting in substantial increases in vibration and temperature. Moreover, the decrease in \( p_d \) may be caused by rupture of the sealing between the vane tip and the cam ring surface; the increase in \( T \) may also be caused by friction of solid contact between the tip and the surface.
Fig. 7  Variation in acceleration, $a$, of the cam ring ($N = 15 \text{ s}^{-1}, t_{in} = 50^\circ\text{C}$)

Fig. 8  Variations in temperatures, $t_{c3}$, $t_{c5}$, $t_{c6}$, $t_{c8}$, of the cam ring ($N = 15 \text{ s}^{-1}, t_{in} = 50^\circ\text{C}$)
Fig. 9  Variations in discharge pressure $p_d$ and input torque $T$ ($N = 15$ s$^{-1}$, $t_{in} = 50^\circ$C)

5. Conclusion

Focusing on vane tip detachment, both vibration and temperature were measured using a test vane pump, which was modified to accommodate an accelerometer and thermo-couples, in terms of the discharge pressure, the rotational speed, and the oil temperature as experimental parameters. The conclusions are summarized below.

i) In the case of a higher oil temperature and a lower rotational speed, the detachment between the vane tip and the cam ring occurred under a lower discharge pressure, although the detachment occurred at high discharge pressure.

ii) When vane detachment began to occur as the discharge pressure increased, the vanes repeatedly and violently detached from and attached to the cam ring, and then the vibration and noise became large. Just after the start of vane detachment, the temperature of the cam ring decreased instantaneously and transiently. Subsequently, the temperature increased noticeably.

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