Dynamic high-frequency behaviour of the swash plate in a variable displacement axial piston pump

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Abstract

In variable displacement axial piston pumps and motors, the rotation of the barrel causes a strong vibrating torque load on the variable displacement components, that is, on the swash plate. In most analysis, this vibrating torque is counteracted by a mechanical end stop, thereby forcing a constant swash plate angle. However, in real pump and motor operation, the swash plate is not held mechanically. Instead, the position of the swash plate is defined by a hydraulic control system. The vibrating torque load on the swash block is compensated by the control piston and cylinder, in which the pressure changes dynamically, as a reflection of the dynamic torque load created by the rotating barrel. The variation of the control pressure demands the swash plate to oscillate around the average swash plate position, which is defined by the control system of the pump (or motor). In this study, the vibrating movement of the swash plate is measured in a variable displacement floating cup pump. The floating cup principle features two swash plates, which are operated out of phase. This has given the opportunity to connect the control systems of both swash plates by means of a restriction. The experiments have shown that the swash plate vibration has a significant effect on the displacement of the individual pistons, while commutating in and around the top and bottom dead centres. The study shows that the swash plate vibration needs to be included in the analysis of variable displacement axial piston pumps and motors.

Keywords

Axial piston pump, variable displacement pump, floating cup principle, dynamic swash plate position, control system

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Introduction

The essence of a variable displacement pump is that the displacement is variable and not constant. Although this is a rather trivial statement, the swash angle has usually been treated as a constant in previous studies, both in simulations and in experimental research. General exceptions are the investigations[1–13] that focus on the dynamic control of variable displacement pumps. Yet, even in these studies, the effects of the dynamic lateral torque load on the movement of the swash plate around the y-axis (see Figure 1) are often disregarded. Instead, the analysis is performed by taking the average torque load for one shaft revolution.

However, in 2000, Dobchuk et al.[9,10] performed an analysis in which he made a model that was 'capable of resolving swashplate displacement fluctuations caused by individual pistons entering and leaving the supply port of the pump'. Dobchuk also performed measurements on a John Deere AL75305 nine-piston variable displacement axial piston pump, in which he applied a rotational potentiometer to measure the swash plate rotation. It is questionable whether this sensor was capable of capturing the dynamic swash plate movements with frequencies of several kilohertz for the higher harmonics. Dobchuk also mentioned a problem with the rotational potentiometer: 'The rotational potentiometer would flip between windings, thereby creating a discrete output with step levels greater than the displacement to be measured'. In addition, Dobchuk measured the oil pressure in the control cylinder under various operating conditions. The measurements show a strong variation of this pressure level.

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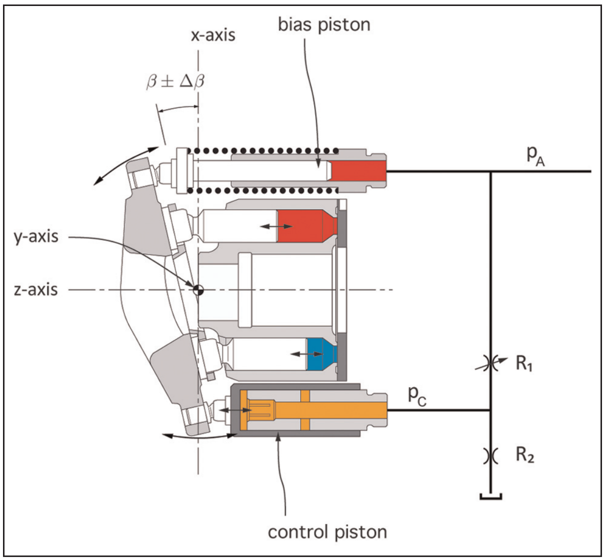

Figure 1. Control of a variable displacement slipper-type pump.

The frequency of the pressure variation corresponds to the rotational speed of the pump times the number of pistons of the pump. The amplitude correlates with the dynamic torque load on the swash plate.

The dynamic torque load was also investigated by Bahr et al.[12] In this study, the dynamic load and movement of the swash plate of a 40 cm^3/rev, nine-piston pump, running at 1450 r/min is analysed by means of simulations. The study indicated that the swash plate makes a vibrating movement around its average value. At a set swash angle of 7.5° and a pump pressure of 30 MPa, the swash plate would make a vibrating movement between 6.4° and 8.1°: 'Fourier analysis for the swash plate forced vibration shows that it contains the same frequencies as in the lateral moment'.

Indeed, if the swash plate is vibrating to such an extent, then this movement has important consequences for the characteristics, performance and behaviour of the pump. The vibrational movement of the swash plate results in a oscillating movement of the pistons of the rotating group (see Figure 1), which is superimposed on the general sinusoidal movement. The effect of the swash plate vibration is largest at and around the top and bottom dead centres where the commutation occurs and will therefore have a significant – if not dominating – effect on the compression and expansion of the oil columns. As such, the swash plate vibration might strongly affect the noise and pulsation levels.

So far, this vibration has not been taken into account in the design and optimisation of pumps. An inclusion of these phenomena would strongly increase the complexity of pump modelling and testing. For testing, it would no longer be tolerable to lock the swash angle mechanically to a certain position. Nevertheless, it would be desirable to have a controllable and reproducible test condition, and therefore, a control is needed to define this test condition. For simulation models, the challenge is to include the interaction between the pressure variation in the pump cylinders during commutation and the dynamic movement of the swash block into the model. The dynamic torque load of the swash plate is dominated by the commutation. As such, the commutation has a strong influence on the dynamic movement of the swash plate. Yet, the dynamic movement of the swash plate also results in a dynamic movement of the pistons and therefore strongly influences the commutation around the top and bottom dead centres. The vice versa relation between commutation effects and dynamic swash plate movement will add to the complexity of modelling of hydrostatic machines.

Floating cup variable displacement pump

In this study, the high-frequency, dynamic behaviour of the swash plate and its control actuators has been measured in a 28 cm^3 variable displacement floating cup pump (Figure 2).

Although the floating cup principle belongs to the family of axial piston pumps, the principle is quite different from conventional slipper-type and bent axis designs:[14]

- The pump has two rings of 12 pistons, arranged in a back-to-back configuration.
- The left and right sides of the pump are operated out of phase.
- The pistons are press fitted into the rotor.
- Each piston has its own cup-like cylinder, which is supported by, and floating on a disc.
- The pump has two barrels and two swash plates.
- Each swash plate can be rotated to vary the displacement of the pump.
- The maximum swash angle is limited to 8°.
- Each swash plate is controlled by means of one bias piston and two control pistons or actuators.
- The actuators create a pure torque on the swash plate.
- The friction between the swash plate and the swash plate bearing and between the pistons and the cylinders is lower in comparison to the slipper-type pumps.

The small amount of friction of the swash plate bearing, the relatively large opening area of the barrel ports and the low friction between the pistons and cylinders of the rotating group result in much less damping of the swash plate system than of conventional slipper-type pumps. Furthermore, the number of pistons per barrel is higher (12 per barrel for the floating cup against 9 per barrel for a typical slipper-type pump), which increases the excitation frequency, whereas the even number of pistons reduces the excitation frequency.

In this study, a new valve is introduced to control the rotational position of each of the swash plates at a

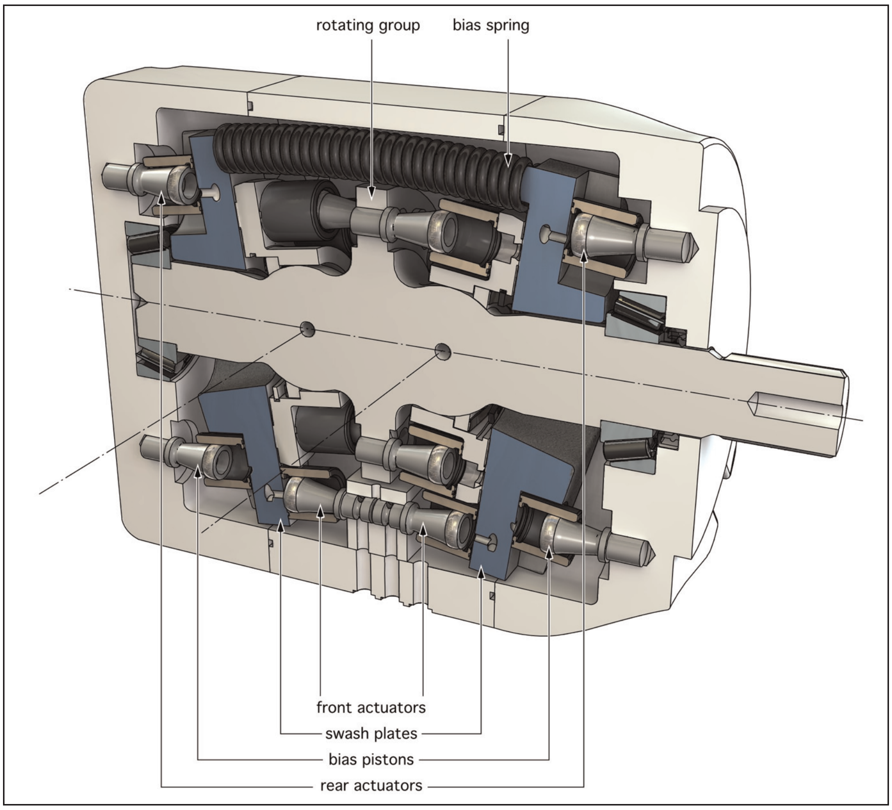

Figure 2. Cross section of a floating cup variable displacement pump (FCVP).

certain predefined value, while leaving the swash plates free to vibrate around this predefined swash plate angle. The movement of the swash plates is measured by means of inductive sensors. Furthermore, the pressure in the control cylinders is measured by means of piezo-resistive sensors.

A fundamental difference between conventional pumps and the floating cup pump is that the latter has two swash plates, which are driven out of phase. The control systems of the two swash plates of the floating cup variable displacement pump (FCVP) can be combined by means of a hydraulic connection between both systems. This study describes the effects of the resistance in the connecting line.

Control of slipper-type pumps

Although there are a large variety of controls for axial piston pumps and motors, most systems are a variation of the design that is displayed in Figure 1. A bias piston and spring push the swash plate to the maximum angle. The control piston counteracts the torque from the bias piston and spring and rotates the swash plate to a smaller angle. The relationship between the control pressure and the swash angle depends on the size of the bias and control pistons as well as on the arm length of these pistons with respect to the y-axis, that is, the axis of rotation of the swash plate. The control circuit itself can be simplified and represented by means of two restrictions in series (R_1 and R_2 in Figure 1). The two restrictions de facto create a simple restrictive pressure divider. In this simplified representation, the variable restriction R_1 symbolises the control valve of the pump.

The rotation of the barrel with the ring of pistons on top of the swash plate creates a strong and dynamic variation of the torque around the y-axis.[2,6,9,11–13] Dobchuk[9] has proven that the torque variation is counteracted by the control cylinder, that is, by the variation of the pressure in this cylinder. At a pump pressure of 120 bar and a rotational speed of 1800 r/min, the pressure in the control cylinder varied between 20 and 70 bar at a frequency of around 270 Hz, which corresponds to the rotational speed (30 rev/s) times the number of pistons (nine).

To create the pressure variation in the control cylinder, the control piston needs to move, thereby forcing oil through restriction R_2. However, to avoid cavitation in the control system, the orifice of this restriction cannot be chosen too small. Since the control pressure is about half the pump pressure, the variable resistance R_1 is about equal to R_2. The two relatively mild restrictions R_1 and R_2 create a substantial leakage flow. For a small 28 cm^3 pump, this leakage flow can be as large as 4.5 L/min.[15] At a rotational speed of 1500 r/min and a 50% displacement, this leakage equals to 19% of the flow output and thus in a 19% reduction of the total efficiency. Although the control valve represents a significant loss in the operation of variable displacement

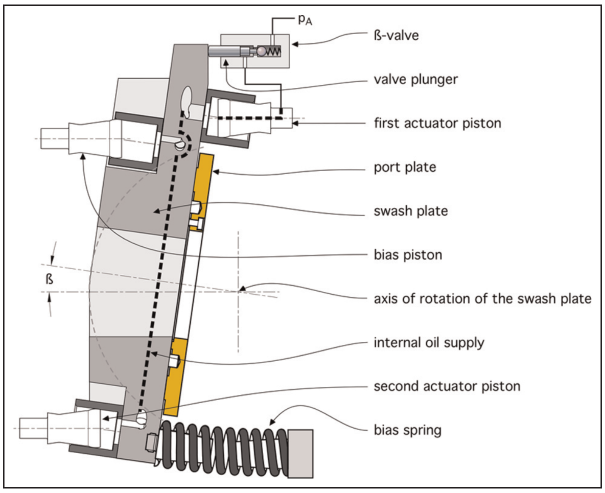

Figure 3. Design principle of the swash angle control at predefined swash plate angles.

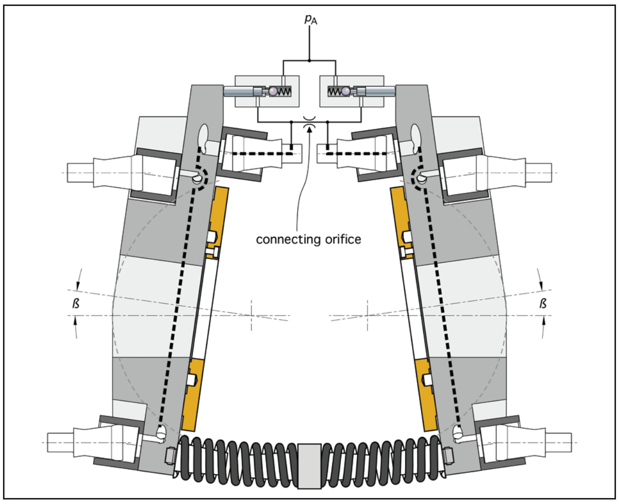

Figure 5. Connecting both actuator systems by means of an orifice.

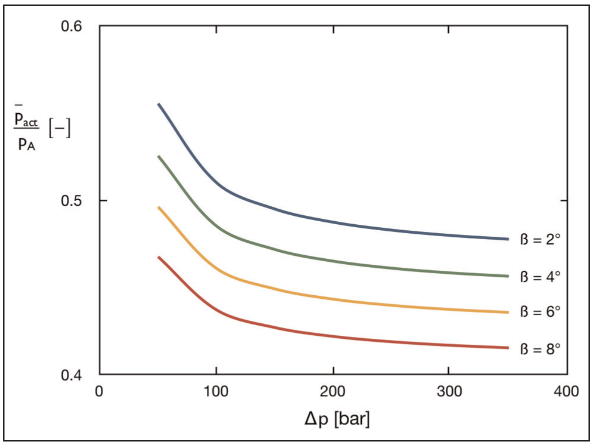

Figure 4. Calculated average pressure in the actuator system (normalised against the pump output pressure) and swash plate angle for different pump pressures, assuming a bias spring force of 45 N.

pumps, the detrimental effect on the pump efficiency is seldom mentioned. The standards for efficiency measurements (ISO4409[16] and SAEJ745[17]) even do not demand the measurement of these losses.

Control system of the FCVP

This study introduces a method to operate the pump at a certain predefined, average, swash plate angle, while leaving the swash plate free to respond to the varying torque load and vibrate around the predefined swash plate angle. The solution is applied in a floating cup pump, but it could also be used in other pump designs. In order to avoid using a mechanical end stop, the two so-called β-valves have been built into the pump, one for each swash plate. Figure 3 shows one of the two swash plates of the FCVP, including the β-valve, the bias piston and spring and the two actuator pistons. Together with the bias spring, the bias piston (which is always connected to the high pressure side of the pump) swivels the swash plate to a larger angle. This is counteracted by the pair of actuator pistons.

The β-valve defines a maximum displacement. As soon as the rotation of the swash plate pushes the valve plunger to the point where the check valve opens, oil is supplied to the actuator system, and the swash plate is rotated to a smaller angle. The swash plate can be controlled between 0° and a certain maximum swash plate angle, which is defined by the length of the valve plunger. Different predefined maximum swash plate angles can simply be realised by means of different valve plungers with different lengths.

The β-valve supplies oil to the control actuators when the swash angle is too large. This increases the pressure in the actuator cylinders, which results in a reduction of the swash angle β. For the 28 cm^3 prototype, the calculated static relationship is shown in Figure 4, taking into account a bias spring force of 45 N. The relationship is calculated on the basis of a simple model for the commutation and expansion in the top and bottom dead centres. In reality, the relationship changes due to carry-over effects[7] and changes with the rotational speed.

One of the most important design characteristics of the floating cup is the double row of pistons, mounted back-to-back on a single rotor. The pistons on the left side of the rotor are positioned in between the pistons on the right side, thereby creating an out-of-phase operation of the two halves of the pump. As a consequence, the pressure variation of the control pistons of the left side of the pump is also out of phase with the pressure profile of the right side. This creates the opportunity to connect and combine the control systems of both swash plates (see Figure 5).

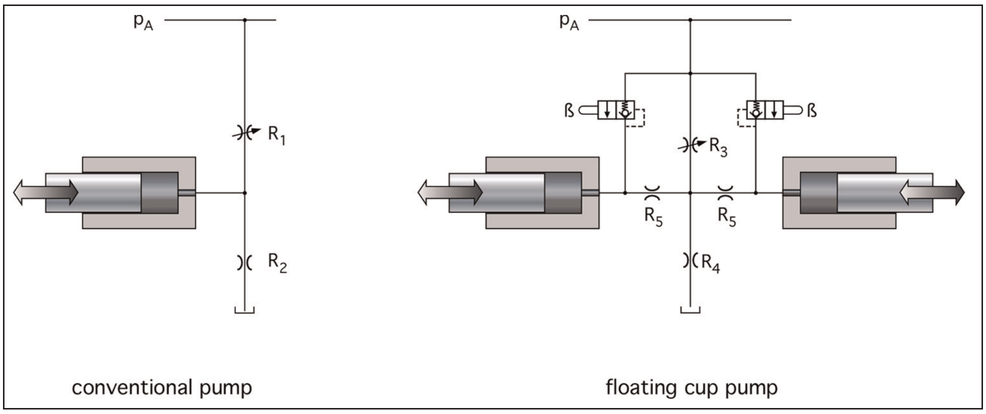

Figure 6. Simplified control circuits of (a) a conventional slipper-type pump and (b) the new floating cup pump with two swash plates.

However, without a resistance in the connecting line, the two actuator systems would be integrated to one single system. This undamped, combined system is undefined and unstable since the operation of one of the two β-valves influences both swash plate positions: the swash plates will be drifting and a proper control of the pump is impossible. Furthermore, the actuator system will no longer be able to create a pressure variation since the two swash plates move completely out of phase: Oil is simply transferred from one actuator system to the other. Without a pressure variation in the actuators, the varying torque load on the swash plates will not be counteracted by the pressure variation in the actuator system. This results in very large movements of the swash plates.

To create a stable control and effective operation of the actuator systems, a resistance needs to be applied between both actuator systems. In this study, various restrictions, that is, various orifices with different orifice diameters have been tested.

To enable the integration of a hydraulic control of the swash plate position, the orifice that connects the two actuator systems is split into two orifices in series (R_5 in Figure 6). The other two resistances R_3 and R_4 fulfil the same pressure split function as R_1 and R_2 in the original pump control. However, in the new design, the resistance R_4 can be chosen much larger than the resistance R_2 of the original pump. In the original circuit, the resistance R_2 cannot be too large, since this would result in large pressure variations in the control actuators, eventually causing cavitation. In the new control system for the floating cup pump, the pressure variation in the control actuators is determined by the value of resistances R_5 and no longer by the resistances R_3 and R_4 of the general pump control. Because of this, the new design allows much smaller orifices for R_3 and R_4, which strongly reduces the volumetric losses of the control system. In this study, however, the general pump control is overruled, and the orifices R_3 and R_4 have been closed altogether. As a result, the pump will be operated at the maximum swash plate angle, which is defined by the β-valves.

Measurement of the rotational position of the swash plates

One of the main objectives of this study was to measure the movement of the swash plate as a response to the varying torque load. To fulfil the demands of an accurate and dynamic measurement, two inductive sensors have been mounted, one for each swash plate (Appendix 2 gives a specification of all sensors). Figure 7 shows the position of the inductive sensor relative to the swash plate.

The inductive sensor measures the distance to a curved target area on the swash plate. The curvature of the target area is made as such that an almost linear relationship is realised between the swash plate angle and the output voltage of the sensor. The dimensions and position of the target area allow a measurement over the entire operating range of the pump between 0° and 8°.

Swash plate angle and control pressure

In a first experiment, the connecting line between the two actuator systems has been closed off completely (i.e. $R_5 = \infty$; see Figure 6). Both actuator systems are completely isolated as long as the β-valves are not operated. The vibrating movement of the swash plates results in a compression and expansion of the oil in the actuator system, without the possibility to supply or bleed extra oil to or from the actuator system (Figure 8).

The isolation of the actuator systems results in strong cavitation during the expansion stroke of the actuators. In the following compression, the pressure signal shows sharp peaks, possibly indicating microdieseling of combustable oil–air mixtures.

In a second experiment, the actuator systems are coupled and connected by means of two orifices in

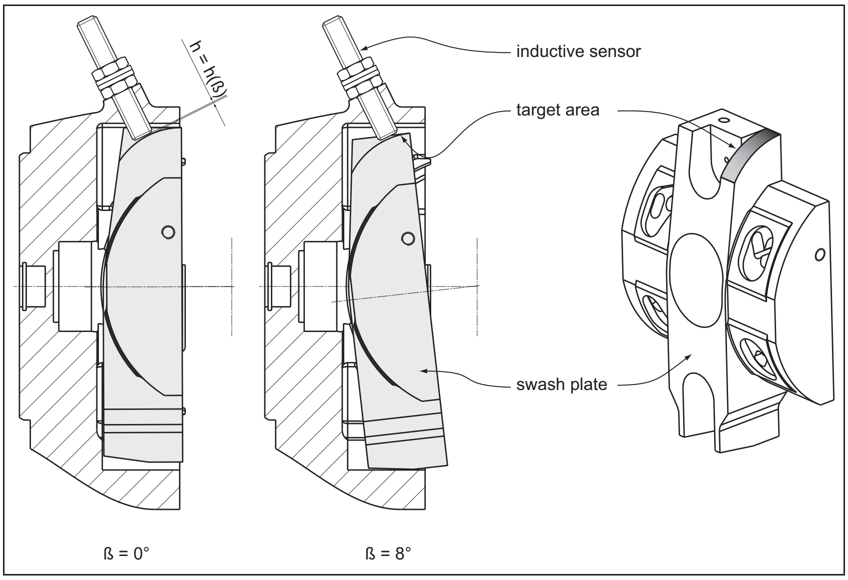

Figure 7. Cross section of part of the housing (at $\beta = 0°$ and $\beta = 8°$), showing the position of the inductive sensors and the curved target area on the swash plate. The gap height h varies as a function of the swash plate angle β.

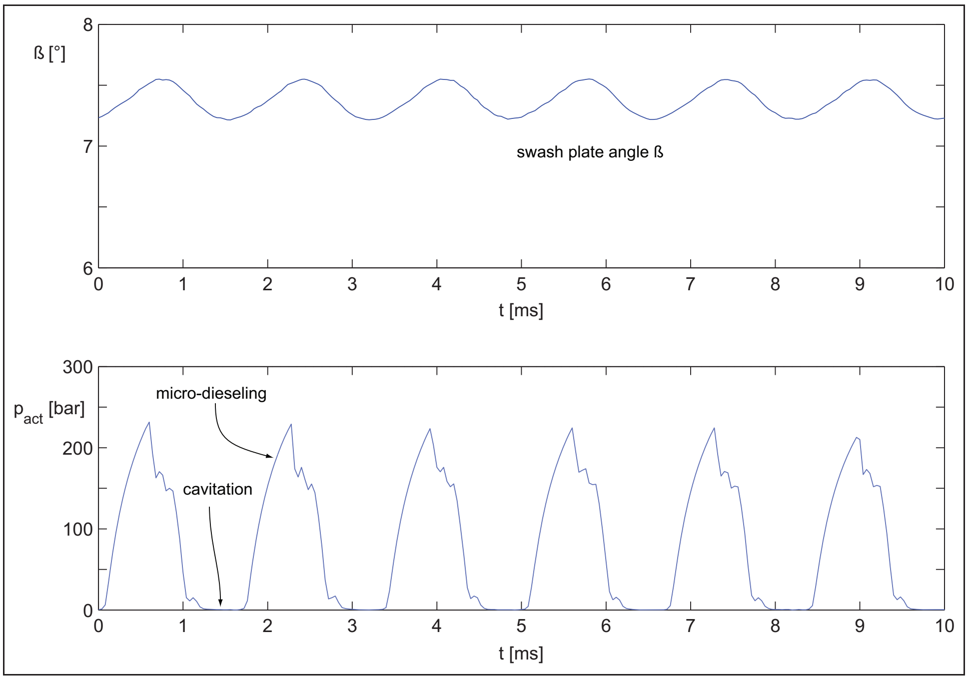

Figure 8. Operation of the pump with a blocked connection between the two actuator systems, showing (a) the measured swash plate movement and (b) actuator pressure of only one side of the pump ($n = 3000$ r/min, $p_A = 260$ bar, $T_{oil,1} = 50$ °C).

series (R_5 in Figure 6), each having a hole diameter of 1 mm. The diagrams (Figure 9) show the signals for both sides of the pump. The β-valves are designed and positioned as such that the internal check valves open at $\approx$ 7.45°. The peak-to-peak amplitude of the swash plate vibration is 0.07°. This is much smaller than in case of a complete separation of both actuator systems, for which the peak-to-peak amplitude was 0.33°. The larger amplitude for the decoupled system is probably due to the strong reduction of the stiffness of the actuator system when cavitation occurs.

In the floating cup design, the pistons on the left side of the rotor are positioned in between the pistons on the right side. As a result, the vibrations of the two sides of the pump are out of phase. This is clearly visible in the test results, as shown in Figure 9. Inside the pump, the flows of both sides are merged to a single flow. Being merged, the vibrations from both sides are

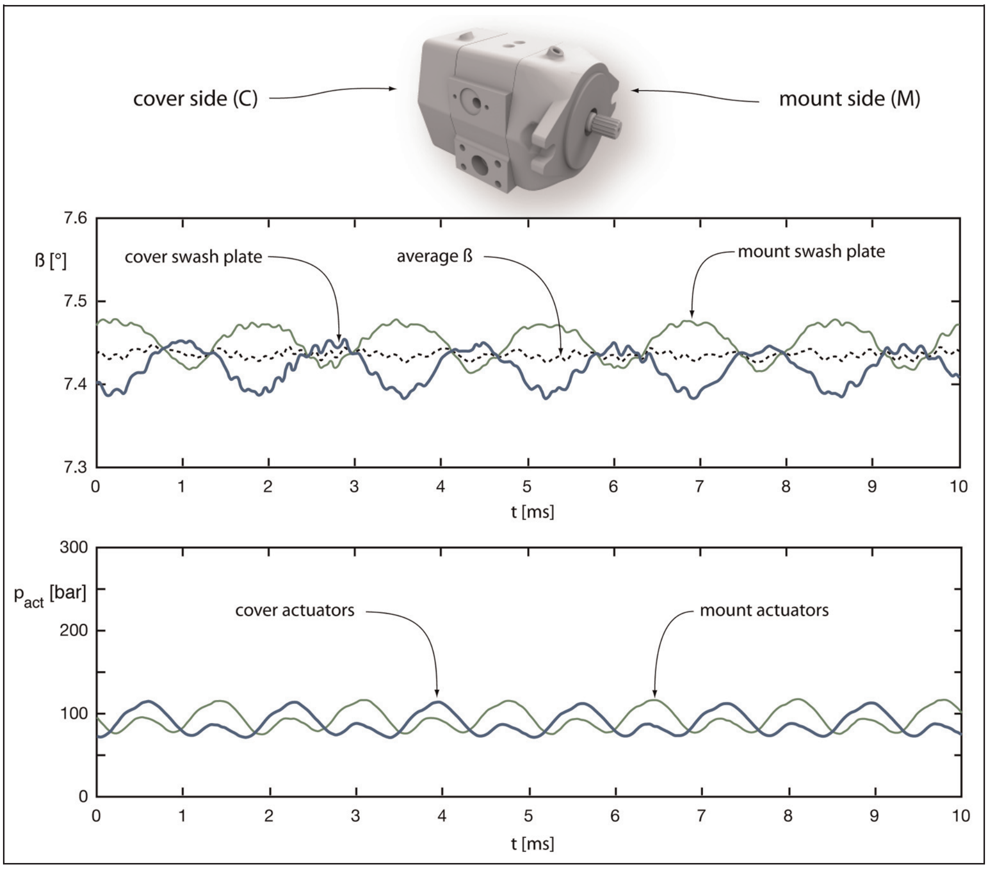

Figure 9. (a) The cover side and mount side of the pump, with (b) measured swash plate angle and (c) actuator pressure. The actuator systems are connected by means of two 1.0-mm-diameter orifices in series ($n = 3000$ r/min, $p_A = 300$ bar, $T_{oil,1} = 50$ °C).

almost completely cancelled (see the average swash plate angle in Figure 9(b)).

The test with the two 1.0-mm-diameter orifices also shows a strong reduction of the amplitude of the actuator pressure. The cavitation is completely avoided and, consequently, also the pressure spikes.

The swash plate vibration and the behaviour of the actuator system are dependent on the rotational speed. Figure 10 shows the swash plate angle and actuator pressure for three different operating speeds of the pump. Only the test results of the mount side of the pump are shown. As earlier, the test results of the cover side of the pump are similar but out of phase with the behaviour of the opposite side. The plots are normalised by dividing the time by the duration T of one pulse calculated as follows

$$T = \frac{60}{n \cdot \left(\frac{z}{2}\right)} = \frac{60}{n \cdot 12} = \frac{5}{n} \tag{1}$$

The measurements show that on a normalised timescale, the pressure traces are almost identical. On the other hand, the amplitude of the swash plate vibration is strongly affected by the rotational speed of the pump. Generally, the amplitude increases at lower operating speeds. The larger amplitude is a response of the system to the lower frequency at reduced rotational speeds. In itself, the lower frequency reduces the flow, which is generated by the actuators. This flow is sent through the orifices to create a pressure variation and will thus result in a smaller pressure amplitude. To compensate for the flow reduction due to the lower frequency, the amplitude of the swash plate vibration is increased in order to achieve the same flow through the orifices as at a higher operating speed.

The relationship between the swash angle amplitude and the operating speed has been further investigated (Figure 11). In these experiments, different orifices have been tested with an opening diameter of 0.7, 0.8 and 1.0 mm. As earlier, for each test, two equal orifices in series are applied in the connecting line between the two actuator systems.

As would have been expected, an increase of the resistance in the connecting line (i.e. a smaller orifice diameter) results in a reduction of the swash angle variation. The experiments also show a strong increase of the peak-to-peak amplitude at low operating speeds. An indirect effect of the larger amplitude is a change of the average swash angle. This can clearly be seen in Figure 12, which shows another measurement at various operating speeds. Because the β-valves control the maximum value of the swash plate position, the maximum value is approximately constant. Consequently, a

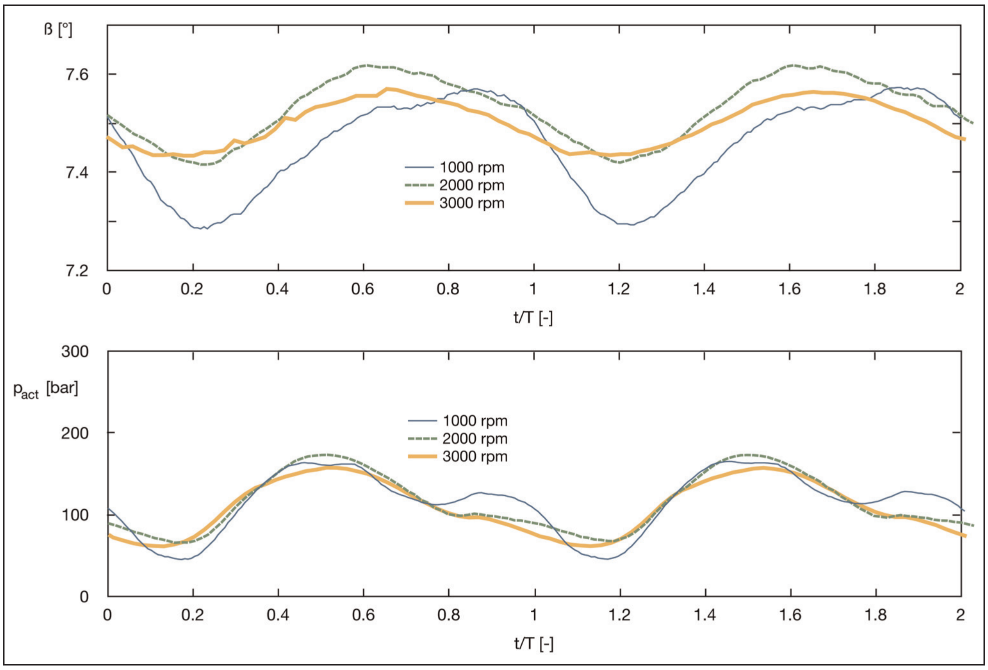

Figure 10. (a) Measured swash plate angle and (b) actuator pressure at the mount side of the pump for three different operating speeds. The actuator systems are connected by means of two 0.7-mm-diameter orifices in series ($n = 3000$ r/min, $p_A = 300$ bar, $T_{oil,1} = 50\,°C$).

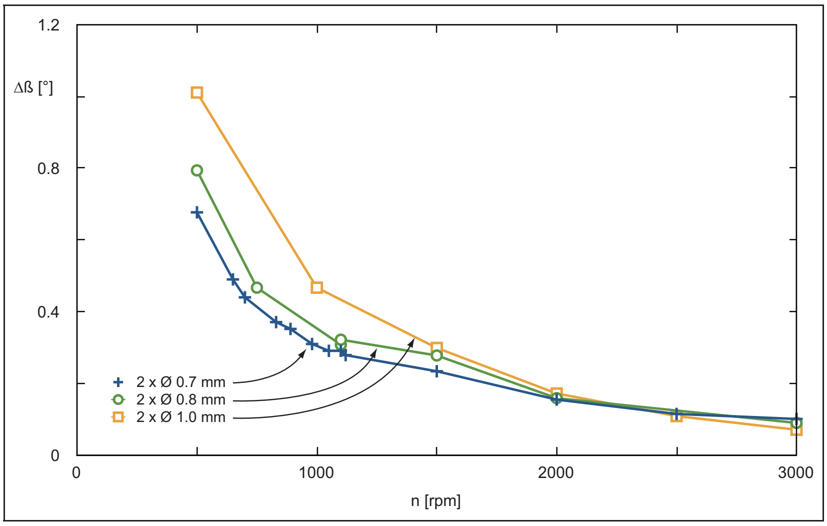

Figure 11. Measured peak-to-peak amplitude $\Delta\beta$ at different operating speeds and for different orifices in the connecting line between the two actuator systems ($p_A = 300$ bar, $T_{oil,1} = 50\,°C$).

larger amplitude of the swash plate vibration results in a lower average swash plate position. This is illustrated in Figure 12.

Influence on the cylinder movement

The measurements of the movement of the swash plate can be used to calculate the effect on this movement on the displacement of the individual pump cylinders. Figure 13 shows the position of an individual piston in its corresponding cylinder. The relatively small swash plate angle of 7.45° results in a short stroke, which is typical for a multi-piston pump. The dotted line would be the sinusoidal movement, in case the swash plate position would have been locked mechanically. The other line in Figure 13 is calculated on the basis of the actual measurement of the swash plate angle β. The measurement is performed at a rotational speed of the pump of 1000 r/min, a pump pressure of 300 bar and an oil temperature at the inlet of 50 °C.

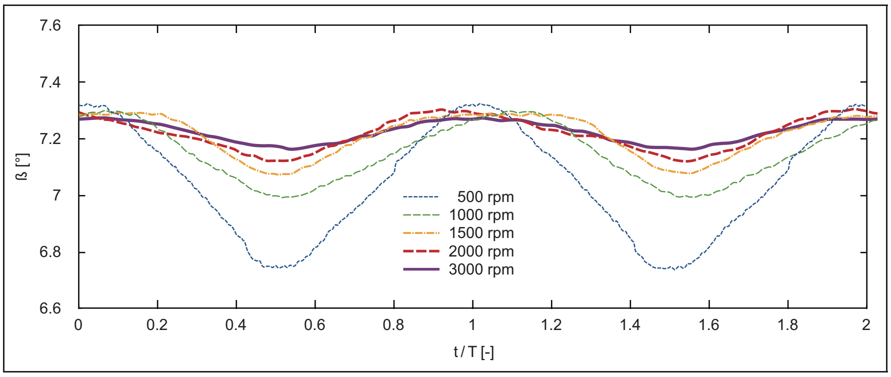

Figure 12. Measured swash angle of the mount side of the pump at different operating speeds (2×0.6 mm orifices, $p_A = 300$ bar, $T_{oil,1} = 50\,°C$).

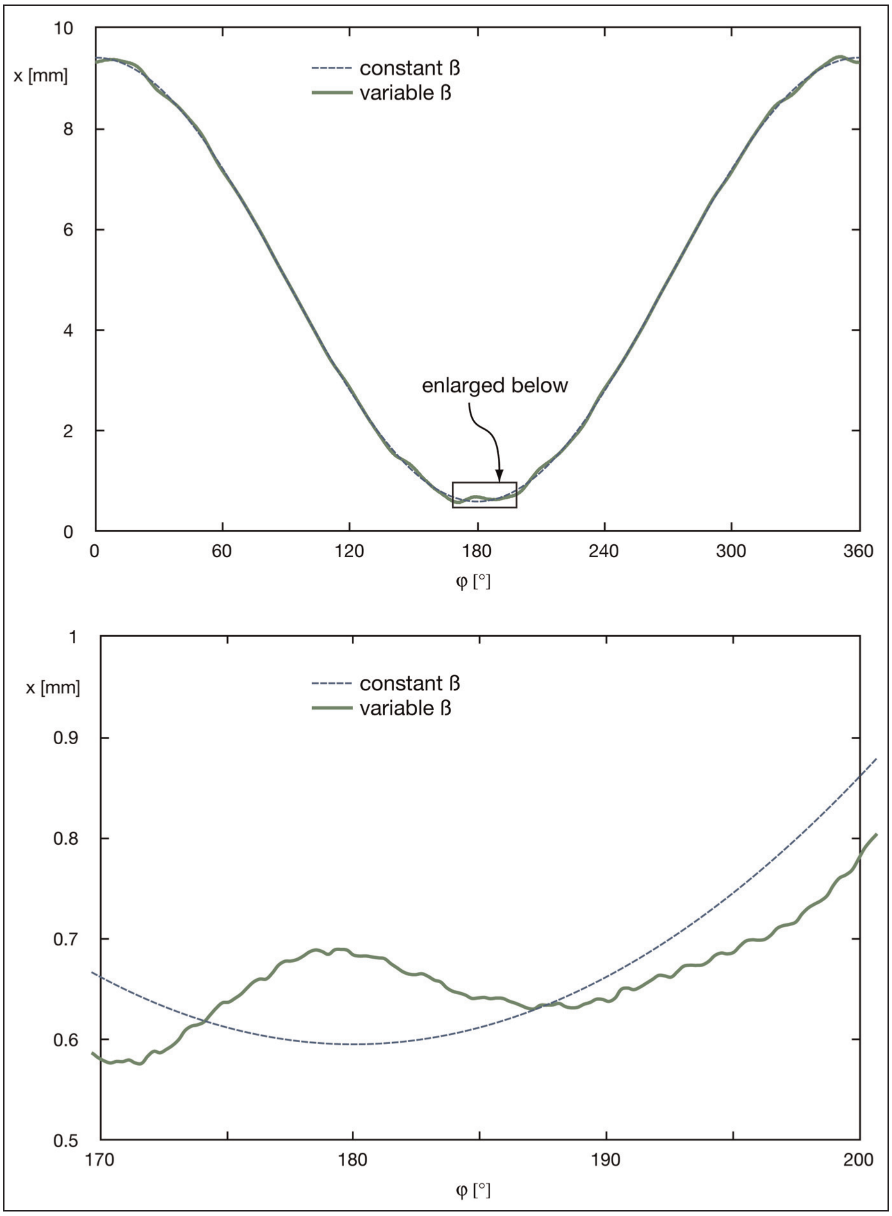

Figure 13. Comparison of the displacement of a single piston with a constant swash plate position β and a variable β: (a) the piston position x for an entire revolution and (b) enlarged area where the commutation occurs around the top dead centre ($n = 1000$ r/min, $p_A = 300$ bar, $T_L = 50\,°C$).

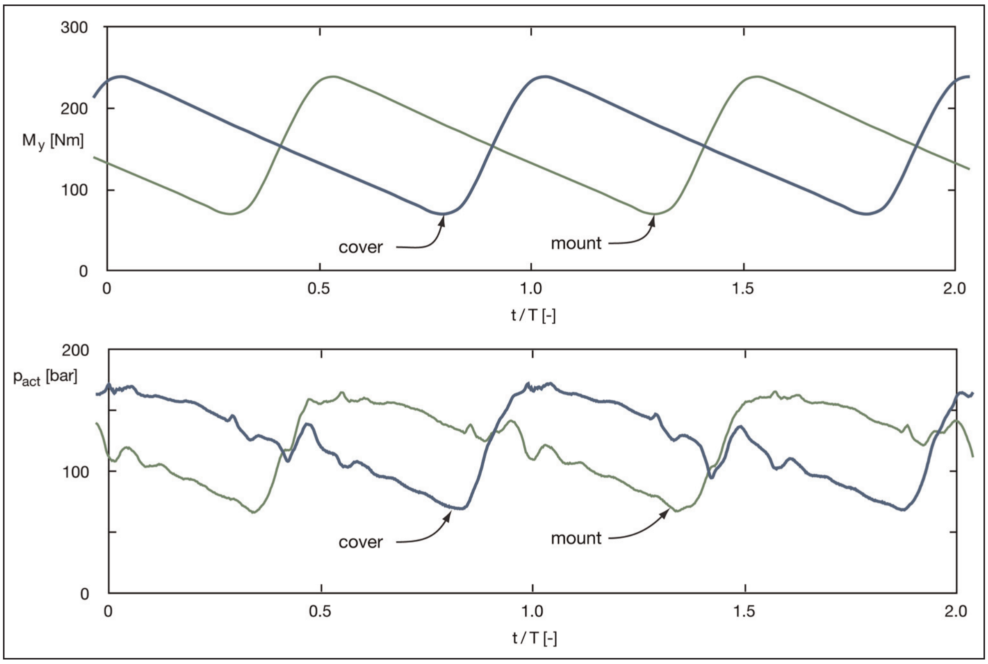

Figure 14. (a) Calculated lateral swash plate torque M_y (p_A = 300 bar) and (b) measured actuator pressure p_{act} (n = 223 r/min, p_A = 300 bar).

Critical for the behaviour of the pump is what happens during commutation, that is, around the top and bottom dead centres. This is also the area in which the effect of the swash plate movement is largest. This area has been enlarged in Figure 13(b). As can be seen, the movement of the swash plate has a significant effect on the displacement of the individual pistons. For this measurement, the displacement due to the swash plate vibration is about as large as the sinusoidal movement during the commutation. Since the amplitude of the swash plate vibration varies with the operating speed (see Figures 11 and 12), the influence of the swash plate vibration will be much larger at low operating speeds. At operating speeds of 3000 r/min, the amplitude of the swash plate vibration is only one-third compared to the vibration at 1000 r/min but even then, the change of the piston velocity will have a significant effect on the Δp in the relief grooves of the port plate. The effect of the vibrating swash block becomes even more important if the pump is running at a smaller displacement, which reduces the amplitude of the sinusoidal movement, whereas the vibration of the swash plate remains the same.

Analysis and conclusion

The driving factor behind the swash plate vibration is the lateral torque load. Figure 14 shows a comparison of the calculated torque load and the measured pressure variation in the actuator system.

The frequency of the rotational vibration corresponds to the rotational speed of the pump and the number of pistons, which is acting on each swash plate. The amplitude is dependent on the pump pressure as well as on the rotational speed, being largest at high pump pressures in combination with low rotational speeds. The measurements confirm the earlier analysis made by Dobchuk[9] and Bar et al.[12]

The vibrating movement of the swash plate has important consequences for the analysis and modelling of variable displacement pumps and motors. Although the measurements have been performed at a floating cup pump, the measurements and analysis performed by Dobchuk[9] and Bahr et al.[12] indicate that the swash plate of other types of axial piston pumps and motors also vibrates. Also in other types of variable displacement axial piston pumps, the torque variation caused by the rotating group of pistons exists as a fundamental vibrating load on the swash plate, which has to be counteracted by the pressure variation in the control cylinder, and therefore, by a movement of the swash plate. It is a fundamental mistake to treat the swash angle as a constant in the analysis, testing, design and development of variable displacement hydrostatic machines. Furthermore, the influence of the pump control should be included in standards for pump and motor testing.

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Declaration of conflicting interests

The author declares that there is no conflict of interest.

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Appendix 1

Notation

h	gap height inductive sensor
M_y	swash plate torque around the y-axis
n	rotational speed
p_{act}	dynamic pressure level in the actuator or control piston
p_A	pump pressure
p_C	control pressure
p_L	supply pressure
R	restriction
t	time
T	pulse duration
T_A	oil temperature at the pump outlet
T_L	oil temperature at the pump inlet
x	piston displacement
x, y, z	Cartesian coordinates
z	number of pistons of the pump
β	angular position of the swash plate
$\Delta\beta$	amplitude of the swash plate vibration
φ	rotational position of the pump axis

Appendix 2

Sensor specifications

Table 1. Pressure sensors.

Make	Type	Principle/description	Range (bar)	Output	Quantity
Kistler	4065A500	Piezoresistive	0–500	To amp.	p_C
	4618	Amplifier	0–500	0–10 V	
Trafag	8891	Thin film	0–60	0–10 V	p_L
Trafag	8891	Thin film	0–400	0–10 V	p_A

Table 2. Temperature sensors.

Make	Type	Principle/description	Range	Output	Quantity
Metatemp	MI-Pt100	Pt100 3 wire mm Pmax 600 bar	0 °C–100 °C	To transmitter	T_L and T_A
	SEM 203P	Transmitter, used for all temperature sensors	0 °C–100 °C	4–20 mA	

Table 3. Inductive position sensors.

Make	Type	Principle/description	Range	Output	Quantity
Balluff	BES 516-324-SA17-05	Inductive measuring type	0–1.2 mm	0–10 V	β
IFM	IF5598 IFA2002-FRKG/US	Inductive switching type	Switching distance of 2 mm	Pulse	n