MOVEMENT OF THE SWASH PLATE IN VARIABLE IN-LINE PUMPS AT DECREASED DISPLACEMENT SETTING ANGLE

Liselott Ericson
Fluid and Mechatronic Systems, Linköping University, 581 83 Linköping
liselott.ericson@liu.se

Abstract. *Emitted noise is one of the biggest drawbacks of using fluid power systems. By using variable pump/motor units the fluid power system architecture is more flexible and efficiency is drastically improved. However, the noise level at fragment displacement angles is not proportional to the power output. The noise level is rather increased with decreased setting angle. Studies by other researchers show increased flow pulsation at small displacement due to the controller's dynamic response.*
In this paper a simulation model to understand the pump/motor behaviour at variable displacement angles is presented. The model includes the important forces created due to pressure built up in the pump and acting on the swash plate. The control valve and actuator are modelled to enable investigations of the oscillations on the swash-plate. The open loop system is compared to the closed loop system, where internal forces from the pistons on the swash plate and pressure excitation are fed back by the controller.
The pulsating pressure in the control actuator has no major impact on the flow and force pulsations. The swash plate oscillations due to the internal piston forces, however, are significant and will affect the outlet flow pulsations. The pulsation is significant but will not have a major impact on valve plate designs for conventional pumps and motors.

Keywords: *Fluid power pump, in-line, swash plate, noise, flow pulsation*

1. INTRODUCTION

Hydraulic systems transfer energy by means of pressurised fluid and are widely used in both industry and mobile applications. The benefits compared to other domains such as electrical systems include, for example, high power density. Noise is one major drawback of hydraulic systems. Pulsating flow and forces created inside hydraulic pumps cause undesired noise. The flow pulsation is spread through the pipeline system to other part of the hydraulic system while the force pulsations are transmitted through connected mechanical structures.

In Harrison and Edge (2000), a good summary can be found of the noise reduction area in hydraulic pumps up until 2000. Harrison mentions among other things pressure relief grooves, pre-compression filter volumes and check valves. All these features are intended to reduce the rapid pressure build up at the commutation between the high and low pressure ports or at the commutation between the low and high pressure ports. The features work satisfactorily for fixed single-quadrant pumps but pumps/motors that work at more quadrants and/or have variable displacements, the usability of these features is limited. Nafz *et al.* (2012) have investigated actively controlled pre-compression angles. Some active or semi-active features are also mentioned in Harrison's summary for example rotating valve plates, but the features are assumed to be expensive to implement. Johansson *et al.* (2007) investigated a second inclination angle on the swash plate, the cross-angle, which is a major improvement in variable piston pump/motor units. The increasing number of studies of more advanced noise reduction features shows an increased interest in quiet, high-efficiency pumps and motors. Advanced and accurate simulation models are needed to enable satisfactory investigations of noise in pumps and motors.

Research on flow pulsations and force oscillations in variable displacement units is most often conducted with no discussion about the displacement control units such as actuators for the swash plate angle. The investigations are generally divided over different areas. Studies of the pump controller does not consider the flow pulsations and other similar dynamic affects. On the other hand, studies of flow pulsations is conducted with fixed swash plate angle even if the pump has variable displacement.

Pulsating load from the piston causes torque variation on the swash plate (Zeiger and Akers (1986) Kassem and Bahr (2000) Bahr *et al.* (2003) among others). Dobchuk *et al.* (2000) show that the control pressure is also pulsating due to the pressure variations in the outlet from the pump. In Achten *et al.* (2013), the swash plate angle oscillations are investigated in a floating cup pump design. The authors conclude that the additional oscillations caused by individual cylinders will affect noise and the efficiency of the pump. There are also studies where the oscillations of the swash plate may be used to reduce torque on the swash plate, in particular the driving torque. In Bahr *et al.* (2003), the swash plate angle is controlled to achieve constant power operation with fluctuated load pressure. The control valve is oscillated with the same frequency as the lateral moment. Mehta and Manring (2005) make a theoretical investigation of how the shaft torque pulsation can be eliminated ideally by adding a super imposed movement of the swash plate displacement.

In this paper a pressure controlled axial in-line pump with variable displacement is investigated. The control valve and actuator are modelled to enable investigations of the oscillations on the swash-plate. The open loop system is compared to the closed loop system, where internal forces from the pistons on swash plate and pressure excitation are fed back by the controller. The changes in force and flow pulsations are analysed when the controller is modelled. The paper focuses on the controller and swash plate movement interactions with the flow pulsation and pressure oscillations at the outlet of the pump.

2. FLOW PULSATIONS AND FORCES

The noise generated in hydraulic pumps can be rated using pulsating flow and forces. These characteristics are highly dependent on pressure level, displacement angle and rotational speed. The pulsating flow created inside the pump can be divided into two parts: kinematic and compressible flow pulsations. Kinematic flow ripple is created due to the limited number of pumping elements and the different piston's linear motion. The ith piston's position is calculated as shown in Eq. 1 and its derivative is shown in Eq. 2. This creates a sinusoidal movement of the piston and thereby the kinematic part of the flow pulsations.

$$x_{p,i} = R_b \tan(\alpha_e)(1 - \cos(\phi_i)) \tag{1}$$

$$v_{p,i} = \dot{\phi}_i R_b \tan\alpha_e \sin(\phi_i) \tag{2}$$

where is defined as

$$\tan(\alpha_e) = \varepsilon \tan(\alpha) \tag{3}$$

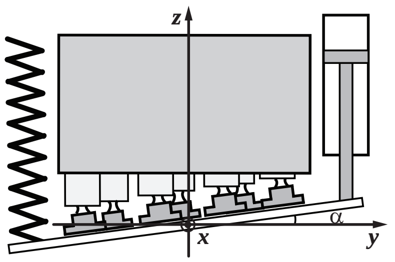
The flow pulsation also contains a compressible part which appears every time a cylinder connects to the high or low pressure port. This part is created by compressibility effects in the pumping chambers. The pulsation is mainly dependent on the cylinder volume, Eq. 4, which should be compressed or decompressed depending on the pressure difference and the bulk modulus, Eq. 5.

$$V_{p,i} = x_{p,i} A_p \tag{4}$$

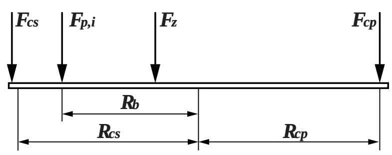
$$q_{c,i} = \frac{V_{p,i}}{\beta_e} \frac{dp_{c,i}}{dt} \tag{5}$$

The compressible part can be reduced by using pressure relief grooves or similar features. The kinematic part is more difficult to reduce; more pistons are one method but rather expensive. However, in conventional axial piston pumps/motors, the compressible part dominates.

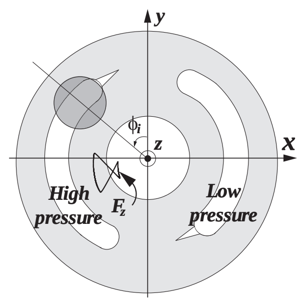
The pulsating forces on the swash plate will also create noise from mechanical vibrations. A principle sketch of the rotating group and the swash plate is shown in Fig. 1(a). The forces and distances of the action point are shown in Fig. 1(b). F_z is the corresponding force from all pistons acting on the swash-plate/valve plate, illustrated in Fig. 1(c).



(a) Principle sketch of the pump's rotating parts and control actuator.



(b) Important forces on the swash plate.



(c) Valve plate configuration.

Figure 1. Principle sketch of the simulated pump.

The moments created by the total axial piston force and acting on the swash plate are calculated according to Eq. 6 - 7 (Ivantysyn and Ivantysynova (2003)), the coordinate system according to Fig. 1(a). The driving torque on the axis is calculated as according to Eq. 8.

$$M_x = \frac{1}{\cos^2\alpha_e} \sum_{i=1}^{z} (p_i A_p R_b \cos(\phi_i)) \tag{6}$$

$$M_y = \sum_{i=1}^{z} (p_i A_p R_b \sin(\phi_i)) \tag{7}$$

$$M_z = \sum_{i=1}^{z} (p_i A_p R_b tan(\alpha_e) \sin(\phi_i)) \tag{8}$$

The pump's driving torque and the moment M_x gives the moment acting on the bearing system named M_b according to Eq. 9.

$$M_b = \sqrt{M_x^2 + M_z^2} \tag{9}$$

These moments are assumed to affect the noise in the pumps as well in the hydraulic systems.

3. SIMULATION MODEL

This section describes the models considered in the paper. The pump model is composed of two interconnected systems; internal pump model and the pressure controller. The model is briefly explained followed by the different set-ups considered. The simulation model is implemented in the simulation program Hopsan, Eriksson *et al.* (2010).

The rotational parts in the pump are simulated as shown in Fig. 2, Werndin *et al.* (2001). The pump model is distributed to individual components, i.e. each cylinder in the pump, two orifices per cylinder representing opening to the high and low pressure port respectively, etc. In this way, the system is kept close to reality. Originally, the model was only used for fixed angle on the swash plate, i.e. a setting angle is set and is kept constant and independent of the pump controller.

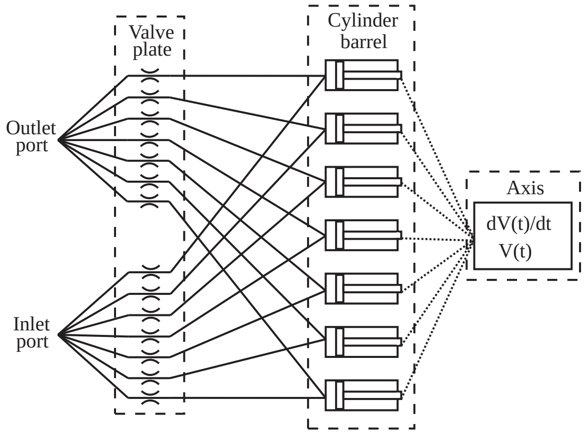


Figure 2. The distributed pump model in Hopsan.

The pressure-controlled pump is modelled with a single-stage control valve and in principle modelled as shown in Fig. 3. The controller consists of a simple a single-stage open centre valve. The actuator is a double-acting asymmetric cylinder. The system is modelled very simply with a volume and an orifice. The dynamic response of such a controller was analysed in for example Palmberg J-O. (1985). The research was made with a radial piston pump but the same analysis has been made for in-line pumps by the same research group. The flow pulsations were not considered in these articles.

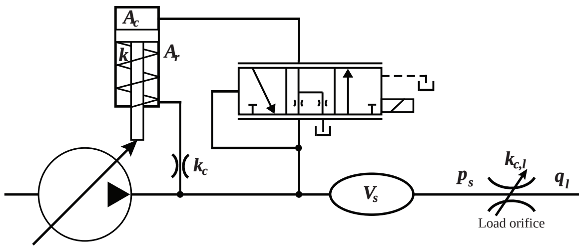
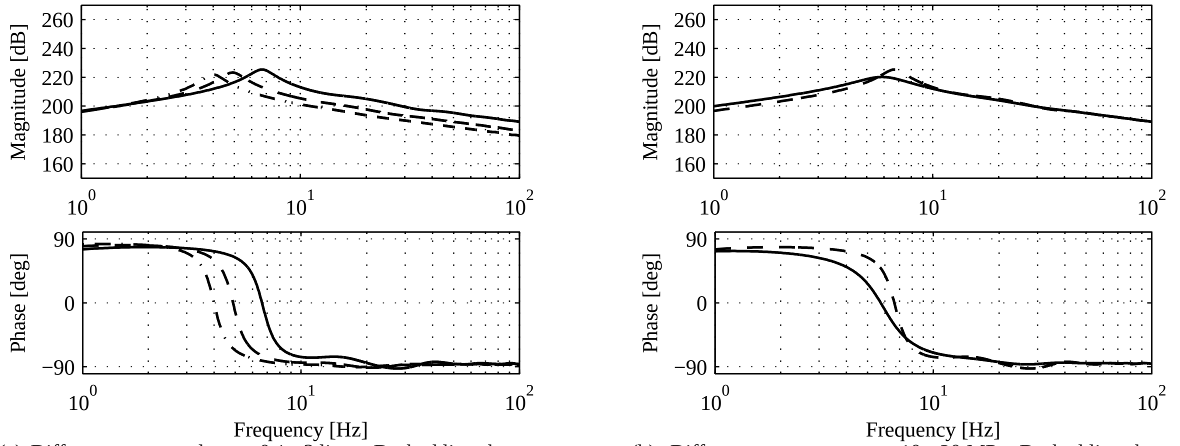


Figure 3. Principle sketch of the pump controller and load valve.

The control system is highly non-linear and the frequency analysis of the closed loop system is made numerically in the simulation program. Figure 4 shows the transfer function of the closed loop system, $\frac{p_s}{q_l}$, of the system shown in Fig. 3. The pump is only modelled as a constant flow depending on the setting angle, i.e. flow pulsations are not modelled. Frequency analysis is made by a load flow disturbance. Figure 4 shows frequency responses with different system parameters. Increased volume will decrease the system response and increase the damping. Increased pressure means increased resonance response.

The harmonic frequencies from the pump pressure ripple are multiples of the pump speed and the number of pistons as Eq. 10 shows.

$$\omega_h = z n_p h \qquad \text{where} \qquad h = 1, 2, 3, \ldots \tag{10}$$



(a) Different system volumes, 0.1 - 3 liters. Dashed lines have a bigger system volume than solid lines.

(b) Different system pressure, 10 - 20 MPa. Dashed lines have a higher system pressure than solid lines.

Figure 4. Frequency response for the closed loop system shown in Fig. 3.

A seven-piston pump will have to run at less than 100 rpm to excite frequencies which may interact with the control and system resonance frequency. The pump considered in this paper is designed for a higher minimum continuous operating speed. The frequency response of the source flow pulsation is shown in Fig. 5.

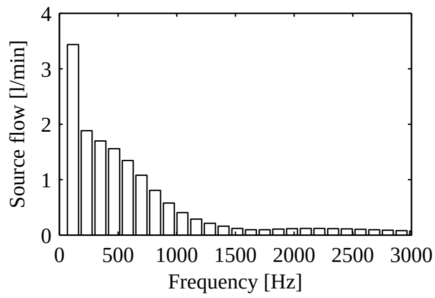
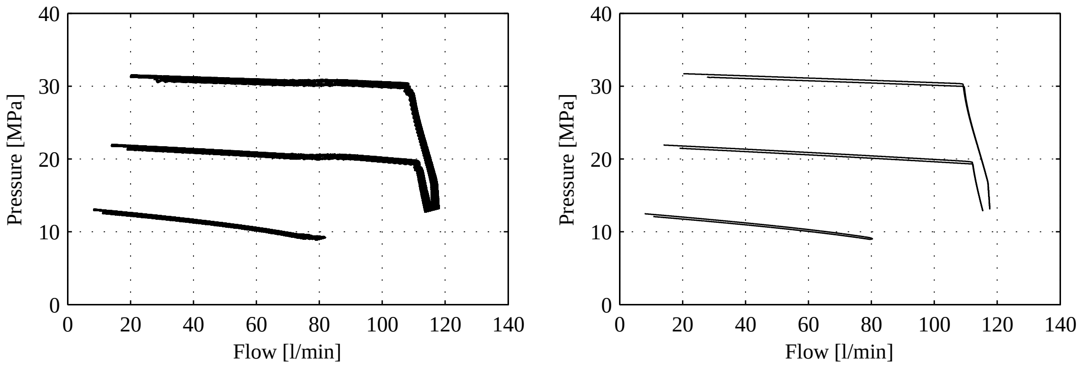


Figure 5. Harmonic frequencies at a rotational speed of 1000 rpm which gives a fundamental frequency of $\approx$ 118 Hz.

Figure 6(a) shows the steady-state characteristics when the full pump model is simulated. The thicker line is caused by the pressure ripple created by the pump flow pulsations. The slope of the horizontal line is due to the flow forces and spring coefficients. Figure 6(b) shows the same system without the flow and force pulsations created by the pump. The parameter values are shown in Tab. 1.



(a) Full model of the pump and controller.

(b) Simplified pressure control pump

Figure 6. Static flow-pressure characteristics of the pressure-controlled pump.

3.1 Experiment cases

To analyse the swash-plate oscillations due to controller and pressure pulsations, three different set-ups of the simulation model are used: Cases A, B and C.

Case A is shown in Fig. 7. The system is used to analyse the noise disturbance independently of the external system and controller. With this model technique, the change inside the pump will be exclusively tested for the internal behaviour. This technique is referred to as Case A in the paper and is used as a reference system. The method is used in many noise investigation of pumps and motors and is useful for making tests independently of the external system, Edge and Johnston (1990). The pressure at the valve plate is kept constant.

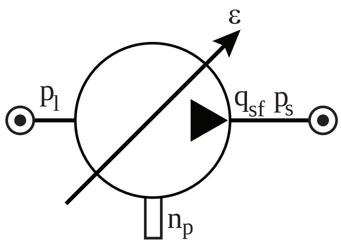


Figure 7. Test set-up for Case A. The pump icon illustrates the pump in Fig. 2.

Case C uses the test set-up shown in Fig. 3, i.e. a fully controlled pump model. The port plate is connected to the system volume. The load orifice controls the load flow. The volume connects to the control system. The swash plate setting angle is controlled by the control actuator, which means there will be an additional oscillation on the swash plate. Also, the pressure in the volume will not be constant as in Case A. In Case C, a full pump model is simulated.

Case B (reduced pump model) differs from Case C in that the piston forces are not acting on the control actuator, i.e. an open loop system is modelled. This means that the oscillations on the swash-plate are caused by the controller itself. The different test set-up allows investigations of what affects the oscillations of the swash plate.

In Tab. 1, values of the main parameters of the controller's dynamic response are stated. The simple model parameter is referred to simulation when the piston forces are not considered for the swash plate movements.

Table 1. Parameter data for pump control.

Parameter	Parameter explanation	Value
A_c	Area of control cylinder Case C	$4.5 \cdot 10^{-4}$ m^2
A_c	Area of control cylinder Cases A, B	$3 \cdot 10^{-4}$ m^2
A_r	Area of control cylinder Cases A, B, C	$5 \cdot 10^{-4}$ m^2
A_v	Area of control valve	$5.03 \cdot 10^{-5}$ m^2
K_c	Pressure flow coefficient Case C	$1.2 \cdot 10^{-12}$ m^5/Ns
K_c	Pressure flow coefficient Case A, B	$2.9 \cdot 10^{-12}$ m^5/Ns
k	Spring coefficient Case C	230 kNm
k	Spring coefficient Cases A, B	130 kNm

The pump simulation parameter is shown in Tab. 2. System pressure and rotational speed are varied to test how the oscillations will vary. The valve plate is simulated with pressure relief grooves at the connection to both high pressure port and the low pressure port, illustrated in Fig. 1(c). The valve plate is typically designed for pump operation.

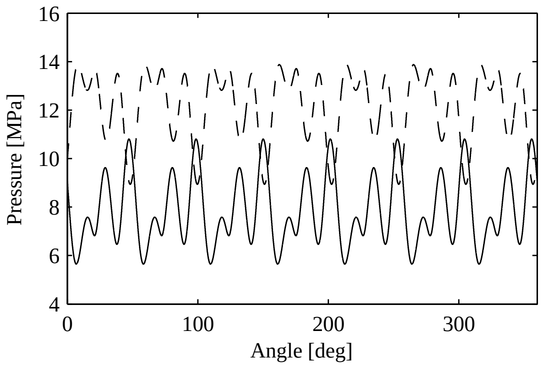
Table 2. Parameter data for the pump model.

Parameter	**Parameter explanation**	**Value**
D_p	Pump displacement	60 cm^3/rev
z	Number of pistons	7
α	Maximum displacement angle	16 degrees
p_s	System pressure	10 - 30 MPa
n_p	Rotational speed	200 - 2000 rpm
p_l	Inlet pressure	0.5 MPa

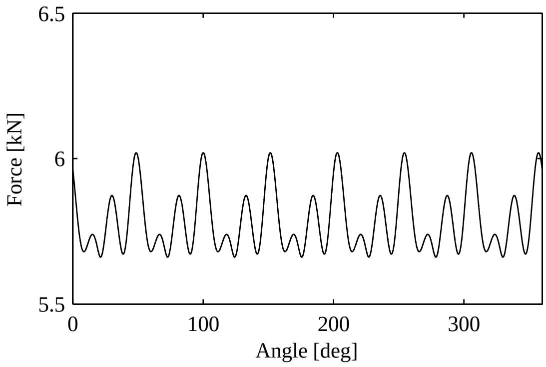
4. RESULTS

In Case B, the force on the pump pistons is disabled, i.e. no force from the pump pistons to the control actuator. The pressure variations in the control cylinder can be directly traced back to the system pressure variations. The oscillations of the swash plate are very small. In the worst tested operation condition, small rotational speeds and small setting angles, the swash plate oscillation is less than 0.2 % of the average setting angle. The tested pressure control does not significantly affect the flow and torque pulsations.

Case C simulates a complete pump model including flow and force pulsation as well as pressure control system. The swash plate will be affected by the force from the pistons as well as the control actuator. Figure 8(a) shows the pressure variations on each side of the actuator. The pressure pulsations are amplified by the force on the piston, Fig. 8(b).



(a) Pressure in the volume of the control actuator. Solid line is the system side and dashed line is the valve side.

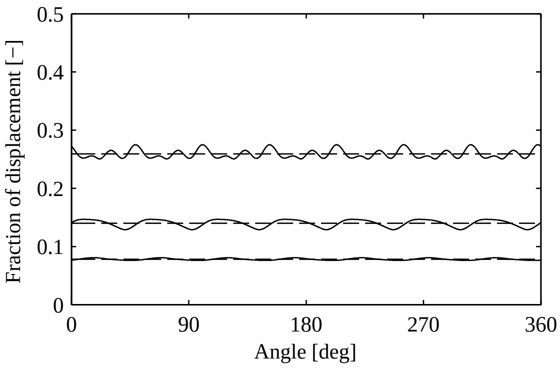


(b) Force at the control actuator connection to the swash plate

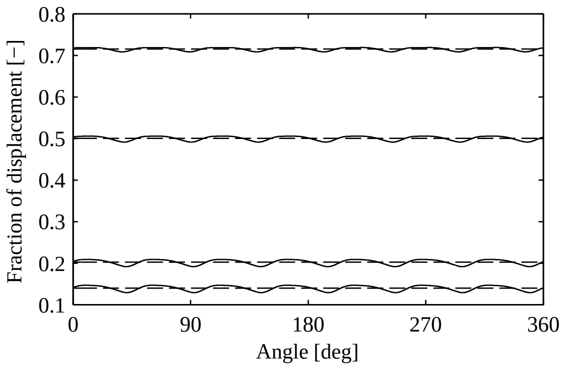
Figure 8. Pressure in the volumes of the actuators and the force on the piston acting on the swash plate.

Case A in Fig. 7 is used as a reference system in the following results. The system uses a fixed setting angle and constant pressure in the high pressure port. Both the setting angle and the pressure are calculated by Root Mean Square from the result for Case C.

Figures 9 and 10 show the oscillations on the swash plate at different setting angle, pressure and rotational speed. There is a clearly change in behaviour at approximately 500 rpm. The amplitude of the oscillations increases at lower rotational speed, smaller displacement angle and lower pressure level.

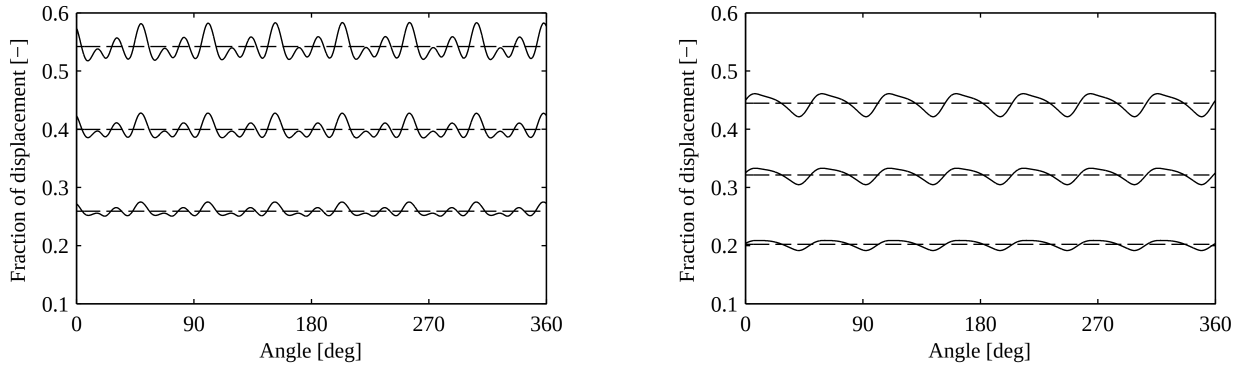


(a) $\approx$ 10 MPa at 500 rpm (upper lines), 1000 rpm (middle lines) and 2000 rpm (lower lines)



(b) 1000 rpm and $\approx$ 10 MPa at different setting angles

Figure 9. Setting angle for Cases A (dashed lines) and C (solid lines) under different operating conditions. Figure 9(a) shows same pressure at different speeds and Fig. 9(b) shows same pressure and rotational speed at different setting angles.

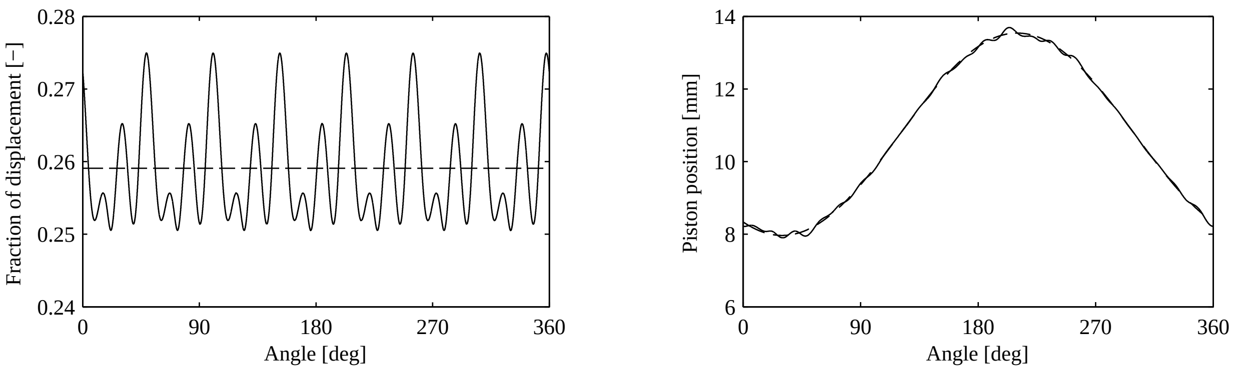


(a) 500 rpm at ≈ 10 MPa (lower lines), ≈ 20 MPa (middle lines) and ≈ 30 MPa (upper lines)

(b) 1000 rpm at ≈ 10 MPa (lower lines), ≈ 20 MPa (middle lines) and ≈ 30 MPa (upper lines)

Figure 10. Setting angle for Cases A (dashed lines) and C (solid lines) at different operating conditions. Figure 10(a) shows rotational speed 500 rpm and Fig. 10(b) 1000 rpm at different system pressures.

The setting angle of the swash plate will influence the piston position and hence the cylinder volume. Figure 11 shows the setting angle and piston position at small setting angles. The setting angle oscillates with an amplitude of 10 %. The piston position has noticeably changed. The oscillation will be reduced at larger angels but the volume at top dead centre is smaller which will make the oscillations more apparent.



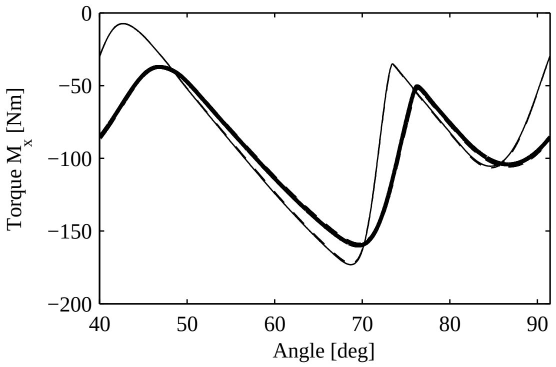
(a) Fraction of swash plate angle

(b) Piston position

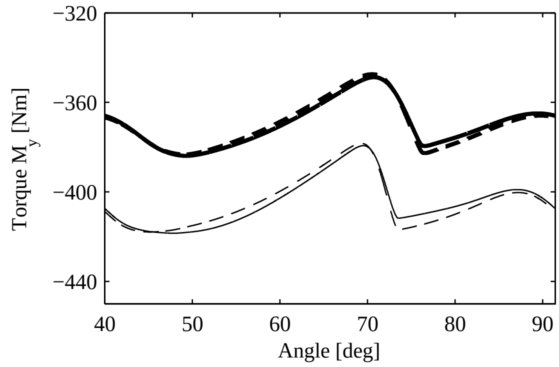
Figure 11. Case A is dashed lines and case C is solid lines. The operating condition is 500 rpm and ≈ 12 MPa.

To view the moment and flow pulsations, operating conditions ≈ 10 MPa at 500 rpm ε = 0.26 and ≈ 10 MPa at 2000 rpm ε = 0.28 have been chosen. Figure 12 shows the two cases and all the moments considered, i.e. M_x, M_y, M_z and M_b. The only significant change occurs at driving torque M_z compared to case A for low rotational speeds. The low rotational speeds make it possible for the pressure fluctuations to break through on the force and flow pulsations. The additional oscillations are not apparent in Case B, where piston forces are not connected to the controller.

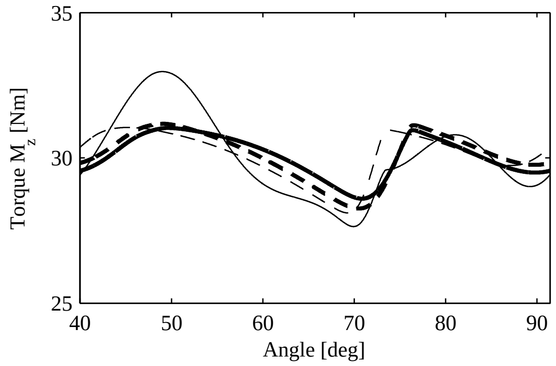
Figure 13 shows system pressure and flow pulsations at 500 rpm and Fig. 14 at 2000 rpm. The kinematic flow pulsation is changed in Case C compared to Case A. The compressible pulsations are reduced in Case C at 2000 rpm and no significant changes at 500 rpm. No significant trends can be seen to explain the behaviour.



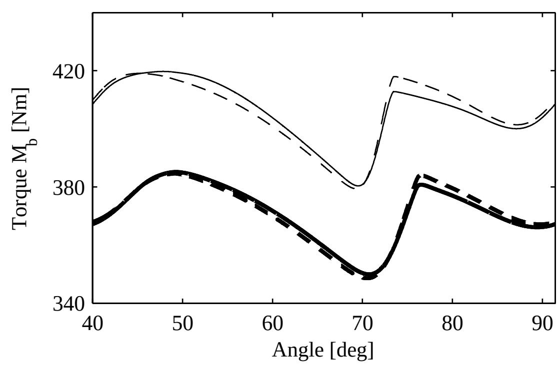
(a) M_x, Eq. 6



(b) M_y, Eq. 7

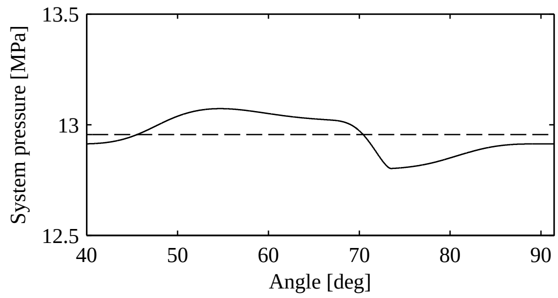


(c) M_z, Eq. 8

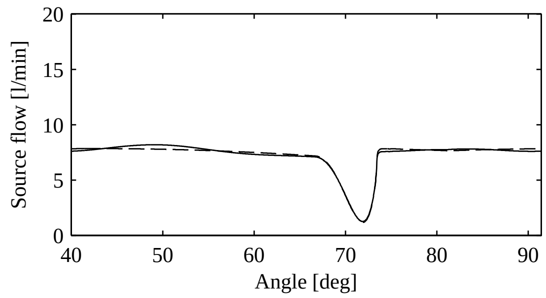


(d) M_b, Eq. 9

Figure 12. Moment around x-, y-, and z-axes and M_b. Solid lines are Case C and dashed lines are Case A. Thicker lines show operating condition $p_s \approx$ = 12 MPa at 500 rpm and setting angle ε = 0.26 and thinner lines $p_s \approx$ = 13 MPa at 2000 rpm and setting angle ε = 0.28

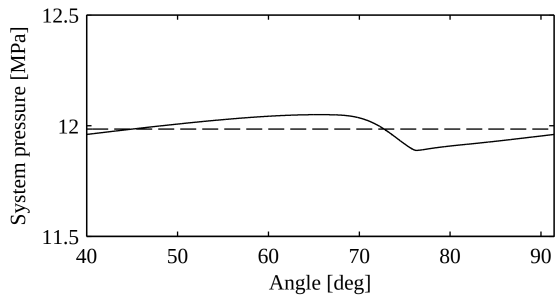


(a) System pressure

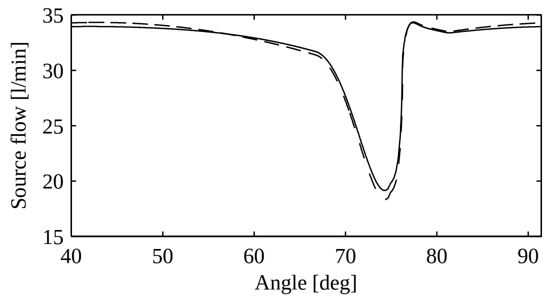


(b) Source flow

Figure 13. System pressure and source flow at operating condition $p_s \approx$ = 13 MPa at 500 rpm and setting angle ε = 0.26. Solid lines are case C and dashed lines are case A.



(a) System pressure



(b) Source flow

Figure 14. System pressure and source flow at operating condition $p_s \approx$ = 12 MPa at 2000 rpm and setting angle ε = 0.28. Solid lines are Case C and dashed lines are Case A.

5. DISCUSSION AND CONCLUSIONS

Kinematic flow pulsations are changed and additional oscillation will appear especially at reduced speeds due to the piston forces acting on the swash plate. The oscillation caused by the controller, Case B, has very small amplitude and the amplitude is independent of the pressure, setting angle and speed. However, the behaviour changes at small rotational speeds, which may be caused by the resonance response of the controller. The amplitude is still very small. In Case C, the oscillations on the swash plate become bigger at small speeds and setting angles and at larger pressures. The behaviour changes drastically at $\approx$ 500 rpm.

It is unlikely that these results will lead to changes in valve plate design, e.g. pressure relief grooves etc., because the design at small displacement angle is not at the critical design limit. However the oscillations on the swash plate may have to be considered when designing cross-angle, Johansson *et al.* (2007), for variable pumps and motors. Also, when active noise reduction is considered, e.g. Nafz *et al.* (2012), where the design depends on the small displacement angles the oscillations may have an impact.

The simulated pump has a maximum displacement angle of 16 degrees and a fairly large dead volume which will make the design less sensitive to small changes in swash plate position and hence piston position. In pumps with smaller displacement angles and smaller dead volumes, the oscillations may have a greater impact and be more apparent.

It is a plausible assumption to reduce the kinematic flow pulsations and also torque variations with swash plate oscillations. However, using the swash plate oscillation to eliminate the compressible flow ripple is improbable.

6. NOMENCLATURE

A_c	Area of control actuator	m^2
A_r	Area of control actuator	m^2
A_v	Area of control valve	m^2
A_p	Piston area	m^2
D_p	Displacement	m^3/rev
i, h	Integer	-
K_c	Pressure flow coefficient	m^5/Ns
k	Spring coefficient	Nm
M_b	Moment on bearing	Nm
M_x	Moment on swash plate	Nm
M_y	Moment on swash plate	Nm
M_z	Driving torque	Nm
n_p	Rotational speed	rev/s
p_c	Cylinder pressure	Pa

p_l	Inlet pressure	Pa
p_s	System pressure	Pa
q_c	Compressible flow	m^3/s
R_b	Barrel radius	m
V_p	Cylinder volume	m^3
v_p	Piston velocity	m/s
x_p	Piston displacement	m
z	Number of pistons	-
α	Swash plate angle of inclination	rad
α_e	Effective swash plate angle	rad
β_e	Effective bulk modulus	Pa
ε	Fraction of max swash plate angle	-
ϕ	Piston angular position	rad
ω_h	Harmonic frequency	Hz

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