Swash Plate Oscillations due to Piston Forces in Variable In-line Pumps

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This study investigates the oscillations of swash plates caused by piston forces acting on the swash plate. Earlier investigations of variable axial piston pumps assume a fixed swash plate angle, i.e. the swash plate is fixed at different displacement angles. Under normal operating conditions, the swash plate is controlled by a hydraulic actuator which affects the swash plate. The presented models are able to separate different losses caused by the swash plate oscillations and the controller. The results show oscillations on the swash plate which affect both efficiency and flow pulsation and hence the noise level.

Keywords: Fluid power pump/motor, efficiency, noise, flow pulsations
Target audience: Fluid power pump/motor developer, mobile hydraulics developer

1 Introduction

Variable pump/motor units increase the overall efficiency of fluid power systems. The fluid power architecture is more flexible and efficiency is drastically improved. However, the variable pump/motor is less efficient compared to equivalent fixed units. This is a big challenge for fluid power in new products. Another aspect is the noise creation in variable pump/motor units. The noise level at fragment displacement angles is not proportional to the power output but rather increases with decreasing setting angles. Investigations of noise in fluid power pump/motor units are most often performed with a fixed swash plate angle, i.e. the swash plate is fixed at different displacement angles. The pressure is kept constant at the pump outlet; this gives a good description of the internal features in the pump independently of the external system. This method is used in many investigations of pump noise and is useful for making tests independently of the external system /1/.

The characteristics which are considered to be the main noise source are flow pulsations, force and moment created at the commutation between the high and low pressure valve plate ports. The pulsations are created by the limited number of pumping elements and the limited oil stiffness. The behaviour of the pulsations depends on how the computation is designed. In /2/, a good summary can be found of the noise reduction field 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 and vice versa. These features work satisfactorily for fixed single-quadrant pumps but their effectiveness is limited in pumps/motors that work at more quadrants and/or have variable displacements. After 2000, the focus of research in the noise area has been more from the method point of view and how already invented features can be optimised. Some of the recent research is in the adaptive and semi-active areas, e.g. /3/ investigated actively controlled pre-compression angles and /4/ adaptive pre-compression angle for variable pumps. The increasing number of studies of more advanced noise reduction features shows increased interest in quiet, high-efficiency pumps and motors and also the possibility to include more advanced features.

Pulsating forces from the pistons cause torque variations on the swash plate, /5/, /6/, /7/ among others. /8/ shows that the control pressure also pulsates due to the pressure variations in the outlet from the pump. The main impact on the oscillations is the forces on the swash plate. In /9/, the swash plate oscillations are investigated in a floating cup pump design and measurements were performed in /10/. 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 /7/, the swash plate angle is theoretically controlled to achieve constant power operation with fluctuating load pressure. The control valve is oscillated with the same frequency as the lateral moment. Reference /11/ makes a theoretical investigation of how the shaft torque pulsation can be eliminated ideally by adding a super-imposed movement of the swash plate displacement.

This paper investigates a pressure-controlled axial in-line pump with variable displacement. The control valve and actuator are modelled to allow the oscillations of the swash-plate to be studied. 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 paper shows that the swash plate oscillations due to the pulsating forces on the swash plate impact on the flow pulsations with different valve plate configurations. The paper also looks at how the pump's efficiency changes when pre-compression angle and pressure relief grooves are used.

2 Simulation Models

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 forces on the swash plate will also create additional oscillations on the swash plate. The axial piston force, i.e. the sum of the product of cylinder pressure and piston area, and its point of action are shown in Figure 1. The graph shows a zero-lapped valve plate and a valve plate with pressure relief grooves.

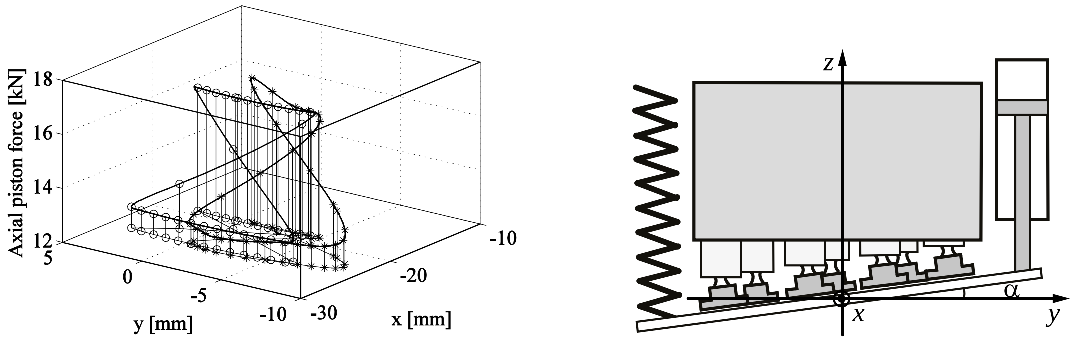

Figure 1: Right: The axial piston force and its point of action for a zero-lapped valve plate (circle) and pressure relief grooves (stars). Left: Principle sketch of the pump and its actuator.

The point of action and the force create the moments around the x and y axes. The moment around the z axis is the driving torque. The force on the actuator is dependent on the moment around the x axis. A principle sketch of the pump's rotating parts and displacement control system is shown in Figure 1.

The simulation model is implemented in the simulation program Hopsan. The pressure-controlled pump consists of a comprehensive pump model, which gives a detailed view of the pulsating flow and forces. The controller consists of a single-stage control valve and a double-acting asymmetric cylinder. The principle model is shown in Figure 2.

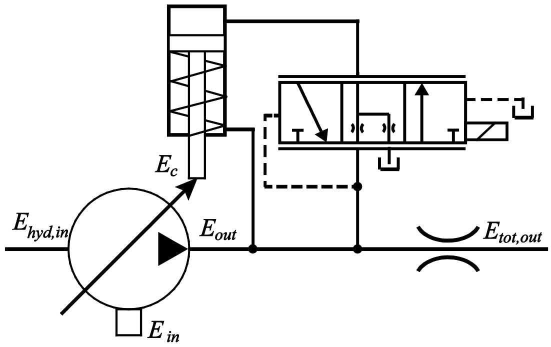

Figure 2: The pump controller including the index for energies.

2.1 Test setups

Two simulation models that work with different principles are used in the paper.

Closed-loop pump model: This model contains the complete pump simulation model where the swash plate angle is controlled by a controller. For comparison of flow pulsations and efficiencies, it is important to equalise the pressure and swash plate setting angle at each operational condition. The displacement is controlled by minor changes in the load valve. Due to the control flow-pressure characteristic, the outlet pressure will change depending on the delivered flow. The pressure is kept at a certain level for all flows by varying the reference pressure in the controller. The difference between E_{out} and $E_{tot,out}$ in Figure 2 is the energy losses for the flow which the controller is using. The controller is not validated by measurements but can be seen as a typical controller for pressure controlled pumps. The control valve configuration has not a major impact on the swash plate oscillations. The valve is too slow to interact with the oscillations at the simulated rotational speeds but the control actuator is nonetheless included in the mass-spring system created by the inertia of the swash plate, the connected spring and the pump cylinders. The resonance frequency of the mass-spring system may not be accurate in the model but the simulations do however give a good picture of the force interaction.

Open-loop pump model: The swash plate is fixed and the inlet and outlet pressures are constant. The outlet pressure is set to the Root Mean Square (RMS) of the closed-loop pump model's outlet pressure and the displacement is set to the controlled average value of the closed-loop pump model. In this model, the energies in Figure 2 are $E_c = 0$ because $v = 0$. The pressure behaviour is different for the two considered models; the open-loop pressure is constant while the pressure for the closed-loop pump is dependent on the external system, i.e. volume and load orifice.

The investigation is made with a 60 cm^3/rev sized in-line pump with the maximum displacement angle at α_{max}=16°. The fraction of displacement, ε, is calculated as equation (1).

$$\varepsilon = \frac{\tan(\alpha_e)}{\tan(\alpha_{\max})} \tag{1}$$

To investigate the swash plate oscillations' dependency on valve plate designs, different designs are used. The valve plates used are: zero-lapped valve plate, which implies that the commutation between high and low pressure port and vice versa is made instantly in the case of fixed swash plate; pre-compression valve plate with different sizes of the pre-compression zone and also optimised pre- compression angle for different operational conditions; pressure relief groove valve plate with pressure relief grooves at the opening to the low and high pressure ports respectively. The pre-compression angle and pressure relief groove result in a similar behaviour.

2.2 Energy and efficiency

Different efficiency models have been developed by various researches, beginning with /12/ where the models are functions of for example pressure, displacement angle and rotational speed. The models have been refined over the years. However, in this study not all losses in the pump are modelled as in the mentioned efficiency models. The losses concerned are the oil's compression and viscous friction at the swash plate. The actuator and control valve have leakage as well as friction. The idea behind the efficiency calculations in this paper is to investigate the difference between a fixed and an oscillating swash plate. By stripping the model of losses that are not of interest, a clearer view can be obtained of the real dissimilarities.

The energy consumed due to the oscillation of the swash plate during one rotation is calculated as equation (2).

$$E_c = \int_0^{1/n_p} F_c v_c dt \tag{2}$$

The energy, E_c, depends on the force on the actuator, F_c, and the speed of the actuator, v_c, at which the swash plate oscillates.

Efficiency is calculated as equation (3). The power varies during the rotation and the energy during one rotation is therefore used as the measure of energy consumption and also the efficiency of the pump.

$$\eta = \frac{P_{out}}{P_{in}} = \frac{\int_0^{1/n_p} P_{out} dt}{\int_0^{1/n_p} P_{in} dt} = \frac{E_{out}}{E_{in}} \tag{3}$$

where η is efficiency and P power. The efficiencies considered in this paper is termed η_{sf}, equation (4) and η_{tot}, equation (5). The notation can be found in Figure 2.

$$\eta_{sf} = \frac{E_{out}}{E_{in} + E_{hyd,in} + E_c} \tag{4}$$

$$\eta_{tot} = \frac{E_{out,tot}}{E_{in} + E_{hyd,in}} \tag{5}$$

Equation (4) shows the efficiency, η_{sf}, without the losses due to the controller flow. This value is compared with the efficiency for the open-loop pump model. The value is used to consider the losses caused by the oscillations of the swash plate. The total efficiency of the variable pump is η_{tot}, equation (5). This value includes the hydraulic losses in the controller.

3 Results

The inlet pressure in all figures throughout the paper is 0.5 MPa. The axial piston force and its point of action affect the swash plate's movement. Figure 3 and Figure 4 shows the swash plate oscillations at two different speeds and fractions of displacement. The amplitude of the swash plate oscillations due to the axial piston force is significantly dependent on the rotational speed. At high speeds, the system dynamics reduce the oscillation, which are therefor smaller at high rotational speed. The oscillations are more dependent on rotational speed than on the displacement angle.

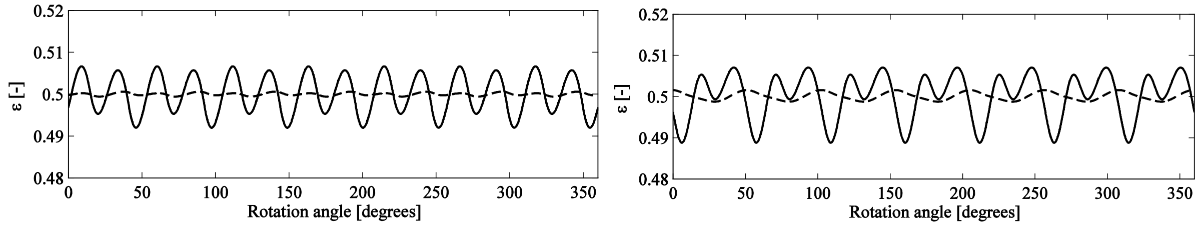

Figure 3: Fraction of displacement with oscillating swash plate at a rotational speed of 1000 rpm (solid line) and 3000 rpm (dashed line). The left graph shows results for zero-lapped valve plate and the right graph shows results for pressure relief groove valve plate. Operating condition is p_p = 20 MPa and ε = 0.5.

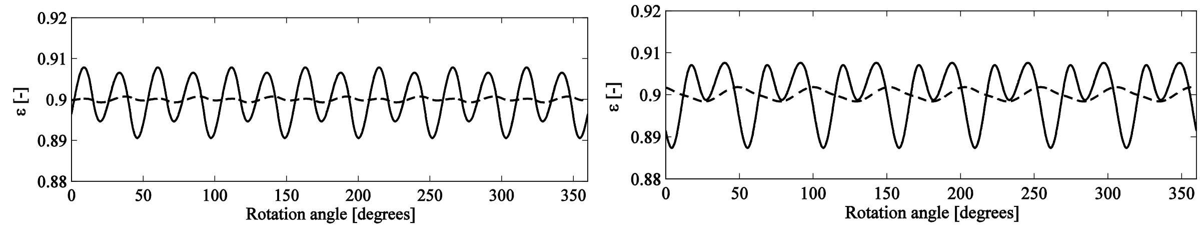

Figure 4: Fraction of displacement with oscillating swash plate at a rotational speed of 1000 rpm (solid line) and 3000 rpm (dashed line). The left graph shows zero-lapped valve plate and the right graph shows pressure relief groove valve plate. Operating condition is p_p = 20 MPa and ε = 0.9.

Figure 5 shows the peak to peak value of the fraction of displacement, i.e. $\Delta\varepsilon = \max(\varepsilon) - \min(\varepsilon)$, for a zero-lapped valve plate and increased pre-compression angle. The figure clearly shows the oscillations' speed dependency. The increased amplitude at 1300 rpm is due to the resonance frequency of the spring-mass system of the swash-plate. The relevant spring cumulate consists of the mechanical and hydraulic springs in the actuator and cylinders.

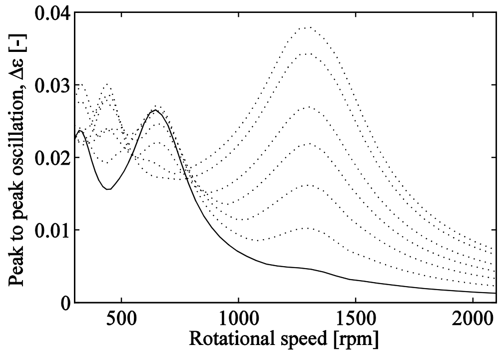

Figure 5: The peak to peak value of the fraction of displacement, i.e. $\Delta\varepsilon = \max(\varepsilon) - \min(\varepsilon)$, at different pre-compression angles. The solid line represents the zero-lapped valve plate and the dotted lines increasing pre-compression angle. Operating condition is p_p = 20 MPa and ε = 0.9.

The resonance frequency of the spring-mass system is about f_r = 150 Hz, which coincides with the pump's fundamental frequency of $f_p = n_p z$ where z is the number of pistons, i.e. z = 7 in this paper, at approximately n_p = 1300 rpm. The fundamental frequency, f_p, becomes increasingly dominant with increased pre-compression angle. The actuator force in frequency domain is shown in Figure 6. The fundamental amplitude is increased when pressure relief groove or pre-compression valve plates are used. This gives rise to the increased oscillation amplitude in Figure 5. The other peaks in Figure 5 are the harmonics.

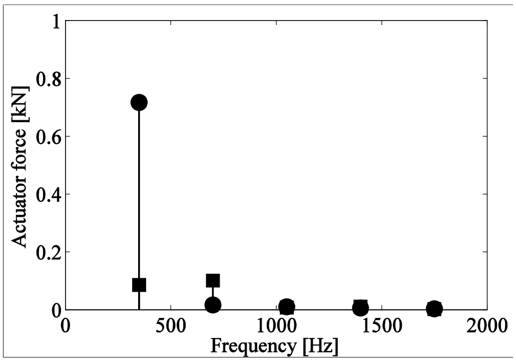

Figure 6: Force on the actuator in the frequency domain with zero-lapped valve plate (square) and pre-compression angle (point). Operating condition is n_p = 1000 rpm, p_p = 20 MPa and ε = 0.5.

The oscillations of the swash plate position will affect the motion of the piston and also flow and forces. Figure 7 and Figure 8 show a piston's axial position and its velocity with the closed-loop pump model with a freely oscillating swash plate and the open-loop model with a fixed swash plate. The two different figures show results for zero-lapped valve plate respective pressure relief groove valve plate.

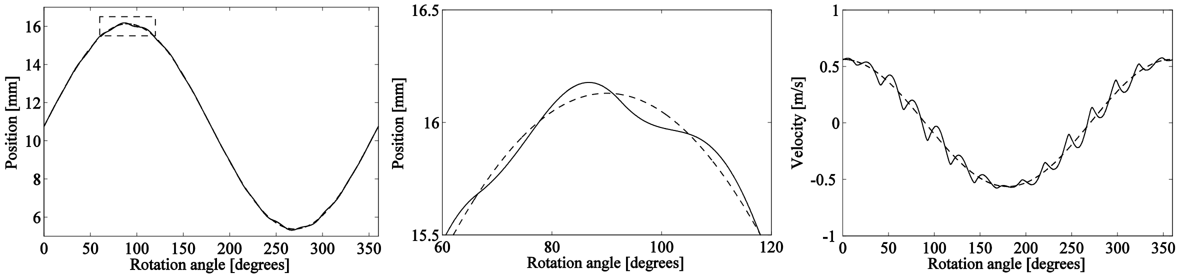

Figure 7: Piston position and piston velocity (most right figure) with fixed swash plate (dashed line) and oscillating swash plate (solid line). The valve plate is zero-lapped. The middle figure shows the position at top dead centre (the rectangle in the left figure). Operating condition is n_p = 1000 rpm, p_p = 20 MPa and ε = 0.5.

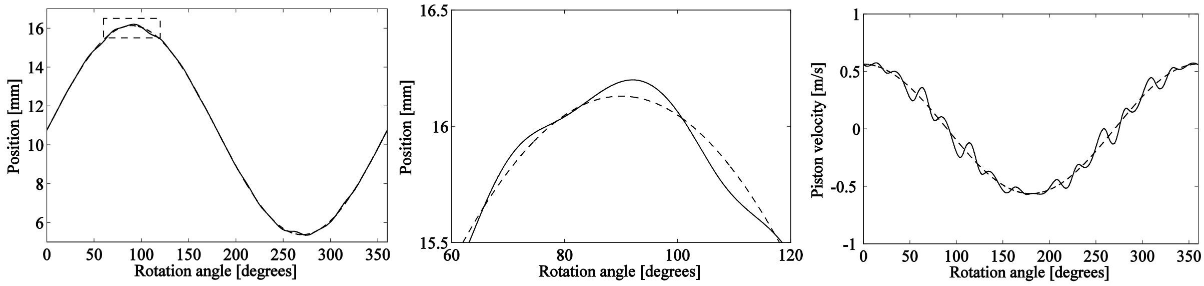

Figure 8: Piston position and piston velocity (most right figure) with fixed swash plate (dashed line) and oscillating swash plate (solid line). Pressure relief groove valve plate is used. The middle figure shows the position at top dead centre (the rectangle in the left figure). Operating condition is n_p = 1000 rpm, p_p = 20 MPa and ε = 0.5.

The oscillations cause flow changes as can be seen in Figure 9 for zero-lapped valve plate and pressure relief groove valve plate. In Figure 10, the flow pulsations are shown for optimised pre-compression angle valve plate at two different speeds. The variation between the fixed and oscillating swash plate is partially caused by the different modelling technique of the external system, i.e. constant and oscillating outlet pressure. However, the difference is significant. The compression part of the flow pulsations decreases with an oscillating swash plate. This has to be taken into account when designing valve plates. The optimised pre-compression angle with oscillating valve plate is for lower speed about 10-20 % smaller than fixed swash plate.

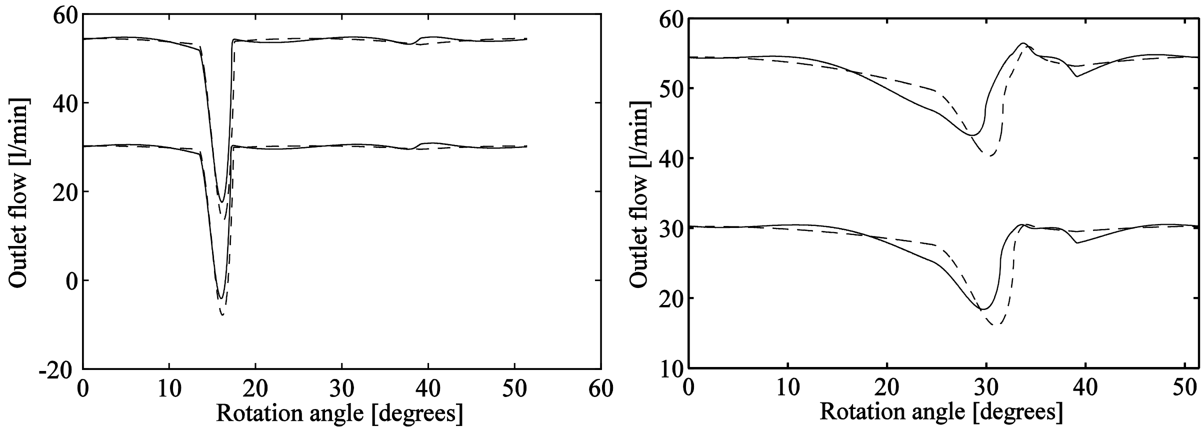

Figure 9: Outlet flow for zero-lapped valve plate (left) and pressure relief groove valve plate (right). The oscillating swash plate is compared with a fixed swash plate using solid lines and dashed lines respectively. The fraction of displacement is $\varepsilon = 0.9$ for the upper lines and $\varepsilon = 0.5$ for the lower lines. Operating condition is n_p = 1000 rpm and p_p = 20 MPa.

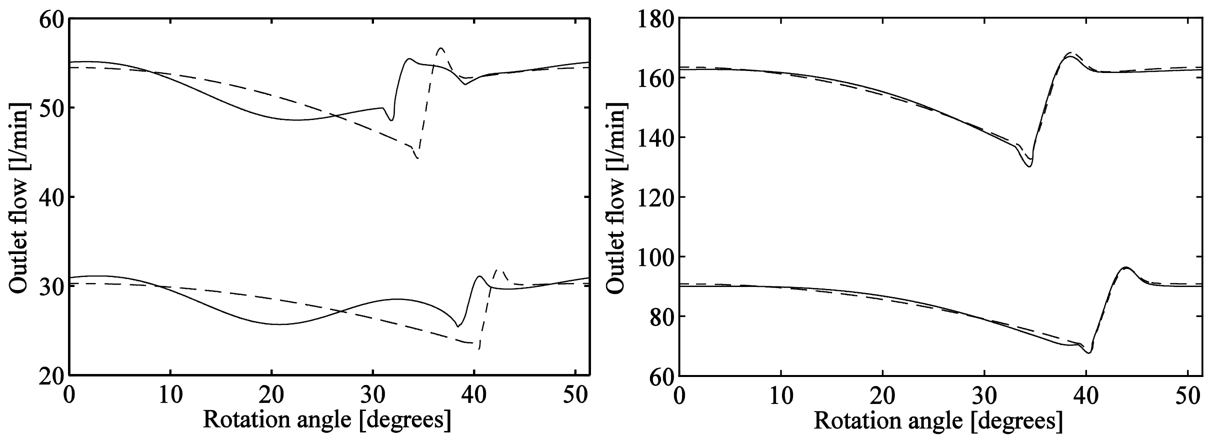

Figure 10: Outlet flow at 1000 rpm to the left and 3000 rpm to the right with optimised pre-compression angle at the connection to the high-pressure port. The oscillating swash plate is compared with a fixed swash plate using solid lines and dashed lines respectively. The fraction of displacement is $\varepsilon = 0.9$ for the upper lines and $\varepsilon = 0.5$ for the lower lines. The outlet pressure is p_p = 20 MPa.

Figure 11 shows the efficiency η_{sf} and η_{tot} at fraction of displacement $\varepsilon = 0.5$ with optimised pre-compression angles at different pressures. This means that the pre-compression angles are changed at every pressure level; when the pre-compression angle is optimised in normal circumstances the lowest pressure has to be selected as design point. Figure 11 also shows results for zero-lapped valve plate. Zero-lapped valve plate with the oscillating swash plate has higher efficiency when η_{sf} is considered compared to fixed swash plate while reduced when optimised pre-compression angles are considered. The difference in efficiency between fixed and oscillating swash plate is reduced at higher speed. The control flow is considerably reducing the efficiency. At higher rotational speed the total efficiency, η_{tot}, is less prominent compared to lower rotational speed. The flow to the control valve is fairly independent to the rotational speed.

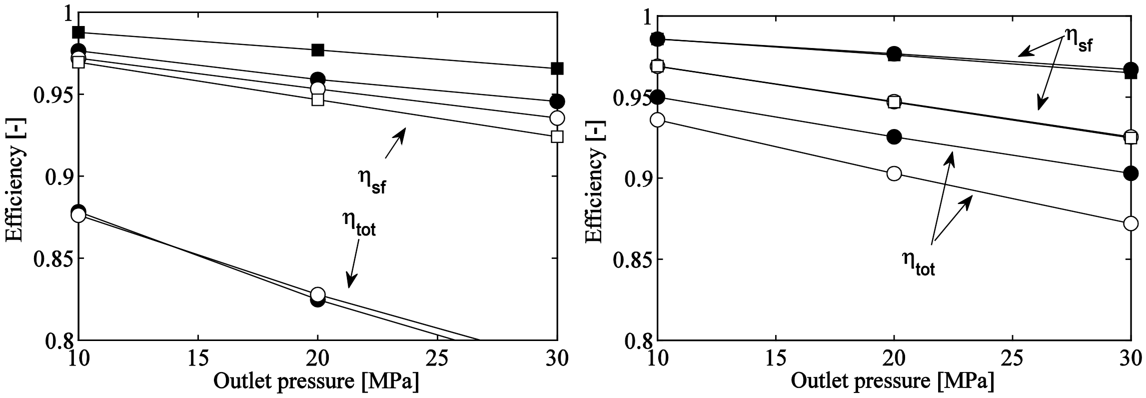

Figure 11: The efficiency simulations of the pump at 1000 rpm to the left and 3000 rpm to the right. Filled markers show the efficiency for optimised pre-compression angles while empty markers show the efficiency for zero-lapped valve plate. Circles show results for oscillating swash plate and squares show results for fixed swash plate. The fraction of displacement is $\varepsilon = 0.5$.

4 Discussion and Conclusion

The oscillation of the swash plate is significant. The amplitude of the oscillations is dependent on the mass-spring system's resonance frequency. Consider that the value of the resonance frequency in this simulation model is not validated by measurements. However, when designing a pump controller the oscillating forces on the swash plate should be considered. A change of valve plate design will also change the interaction between the forces in the pump and hence the actuator force.

The flow pulsations show large variations between fixed and oscillating swash plates. The oscillation will change the valve plate design if the main aim of the design is to minimise the flow pulsation. The oscillations reduce the compressible flow pulsation due to the direction of movement, i.e. the oscillation creates a pre-compression of the cylinder volume. The optimised pre-compression angle will be slightly smaller if the valve plate is optimised with oscillating swash plate. Smaller displacement angles and dead volumes may increase the flow pulsation changes due to the swash plate oscillations. The results may be affected due to the different modelling technique for the fixed and oscillating swash plate models.

The oscillation will affect the efficiency of the pump. If the control flow is not considered results shows both an increase and a decrease of efficiency depending how the valve plate configuration is made. The control flow is reducing the efficiency a lot. The continued work will include experimental verification and validation of the model. Also, a more comprehensive model of the pumps internal behaviour is needed to increase the understanding of the losses in the pump.

Nomenclature

Variable	***Description***	***Unit***
E_{in}	Energy on the shaft	[J]
$E_{hyd,in}$	Energy at the inlet port	[J]
E_{out}	Energy at the outlet port before the controller port	[J]
$E_{out,tot}$	Energy at the outlet port after the controller port	[J]
F_c	Force	[N]
f_p	Frequency	[1/s]

n_p	Rotational speed	[rev/s]
p_p	Outlet pressure	[Pa]
v_c	Velocity	[m/s]
α_{max}	Maximum displacement angle	[rad]
α_e	Effective displacement angle	[rad]
ε	Fraction of displacement	[-]
η_{sf}	Efficiency excluding the control flow	[-]
η_{tot}	Efficiency including the control flow	[-]
ω	Rotational speed	[1/s]

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