Swash plate oscillation in a variable displacement floating cup pump

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Abstract

The oscillation of a swash plate has been studied in a variable displacement floating cup pump. Servo valves have been designed and built into the pump. These so called ß-valves define a certain operating angle, while leaving the swash plate free to oscillate, in the same way as during real pump operation. Measurements have been performed in a wide range of operating conditions. A simulation model has been built to get a better understanding of the dynamic behaviour. The study has proven the large impact of the swivel torque of the swash plate on the pump behaviour. The swash plate oscillation creates an additional displacement of the pump, especially during commutation, which is often larger than the original sinusoidal movement. The dynamic oscillation of the swash plate therefore dominates the commutation phenomena. It has also been proven that the overall efficiency is substantially reduced if the swash plate is free to oscillate, like in real pump operation. Although this study is performed on a floating cup pump, the authors of this paper believe that these effects also occur in conventional types of pumps and motors. It is recommended that a similar study should be performed on slipper type and bent axis pumps and motors.

Keywords: Variable displacement pump, floating cup principle, oscillation of the swash plate

1 Introduction

When analysing variable displacement pumps or motors, it is often convenient to set the displacement at a constant value. In for instance a slipper type pump, this condition can be realised by locking the swash plate position by means of a screw or bolt. This way, a certain swash angle and pump displacement can be defined and reproduced during experiments. In simulations, the same 'locked' position is simply realised by defining the angular position of the swash plate as a constant. These procedures and assumptions are so common that they are often not even mentioned in publications.

This paper discusses the validity of these procedures and assumptions. Evidently, in real operation, a variable displacement pump does not have a constant displacement. In most cases the swash plate position is controlled dynamically by means of hydraulic cylinders. These cylinders not only set the swash angle but also act as shock absorbers for counteracting the oscillating torque load of the pistons on the swash plate. From previous literature [1-6] it is already known that the oscillating torque load results in an oscillating rotation of the swash plate. However, the vibration of the swash plate also results in an extra movement of the pistons of the rotating group, which is superimposed on the general, sinusoidal movement [1, 2].

The extra oscillating displacement is largest in the top and bottom dead centres [7]. In these commutation zones, the base sinusoidal movement is minimal. The oscillating movement of the swash plate and the resulting movement of the pistons could therefore have a larger effect on the commutation than the base sinusoidal movement of the pistons. This is of importance for the design and dimensioning of the port plates, in particular of the silencing grooves.

This paper investigates the oscillating movement of the swash plates in a 28 cc variable displacement floating cup pump. A special servo valve has been introduced which controls the pressure level in the actuator cylinders and, more or less, defines a constant and reproducible operating condition, while keeping the swash plates free to vibrate. A simulation model has been developed which describes the vice versa relationship between the commutation and the swash plate movement. The results from the simulation have been compared to measurements in a wide range of operating conditions. Finally the effect of the swash plate oscillation on the piston movement, the commutation and the efficiency will be discussed.

2 Variable displacement floating cup pump

In the floating cup principle [8, 9] the pistons are not free to move, but are press fitted into a rotor. Each piston has its own cup-like cylinder, which is supported by and floating on a tilted ‘barrel plate’. The floating cup principle is a multi-piston principle. Typically, 2 rings of 12 pistons are mounted side by side on a rotor, resulting in a total number of 24 pistons.

The rotation of the barrel plates and the cups is synchronised with the main shaft by means of a pin and slot mechanism. In the variable displacement design [10], the pump displacement is defined by the angular position of the swash plates (see Figures 1 and 2).

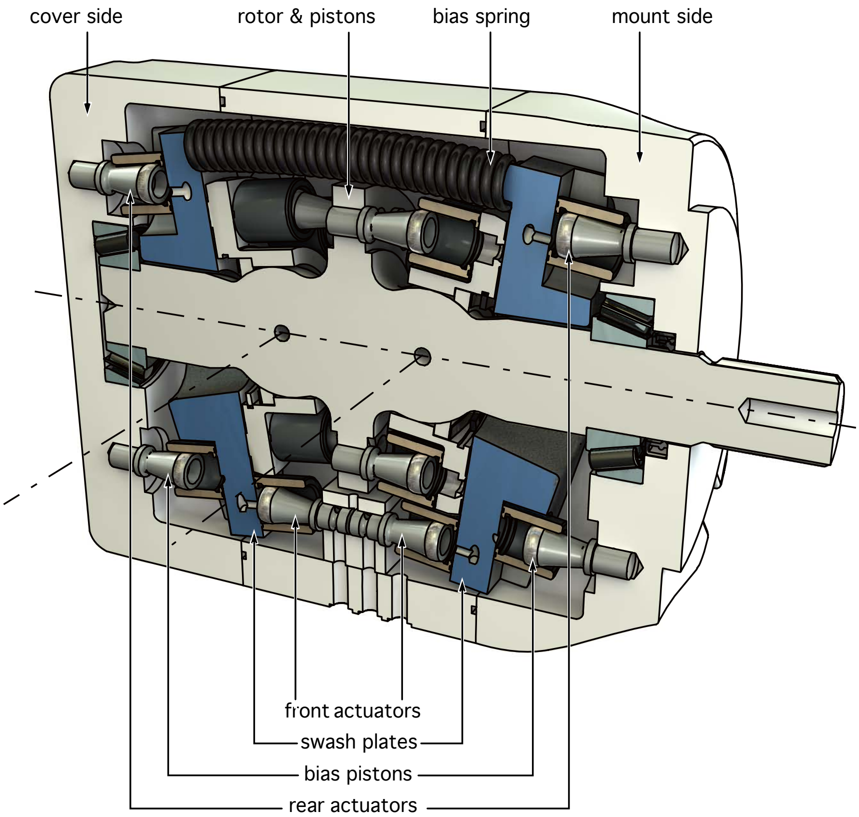


Fig. 1: Cross section of the variable displacement floating cup pump (FCVP28). The right side, where the pump is mounted, is called the mount side, the left side the cover side.

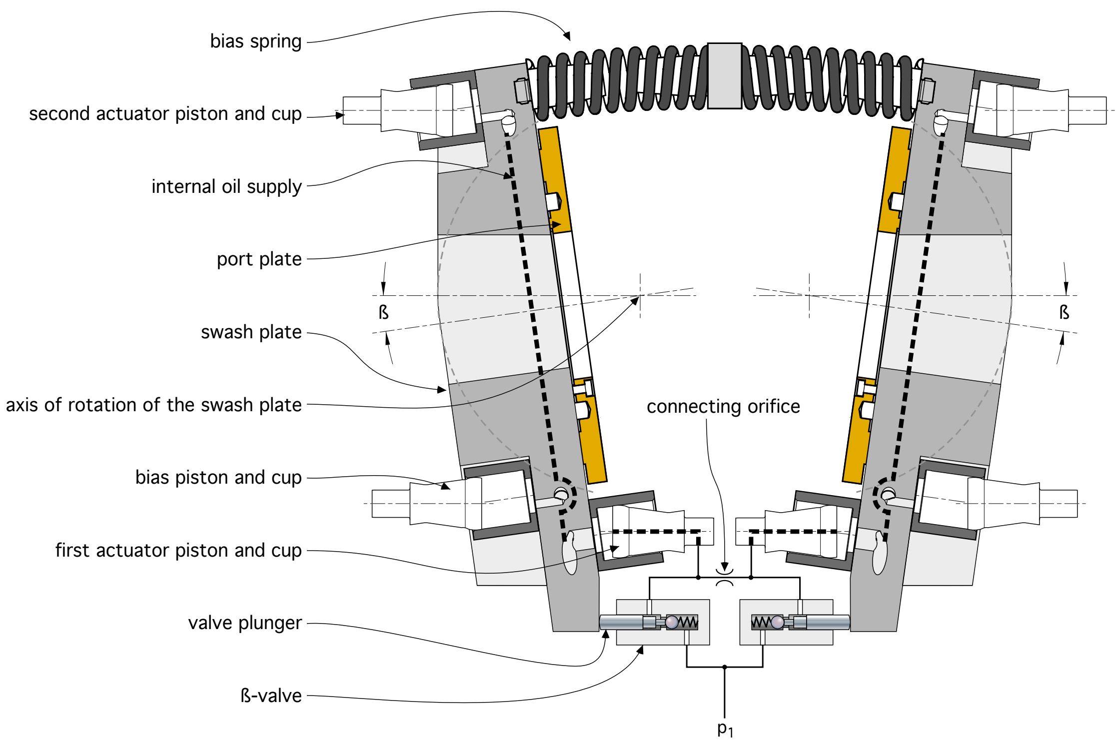


Fig 2: Cross section of the swash plates and actuator systems

Figure 2 shows a detailed cross section of the swash plates. A bias spring sets the swash angle to a maximum at the start of the pump operation. Furthermore, for each swash plate, one bias piston and two actuator pistons are used to control the swash angle *ß* i.e. the displacement of the pump. The bias and actuator pistons are the same as of the rotating group. Like in conventional slipper type pumps, the cylinder of the bias piston is always connected to the high-pressure side of the pump. The bias piston and one of the actuator pistons are positioned at the backside of the swash plate, thereby supporting and reducing the deformation of the swash plate.

Unlike conventional pumps, in which only one actuator piston is applied, each swash plate has two actuator pistons to control the angular position *ß* of the swash plate. The two actuators push in opposite directions (Figures 1, 2 and 12). As a result the two actuators together create a pure operating moment, thereby eliminating any axial force load on the swash plate and its hydrostatic bearings.

The average pressure level in the actuators determines the average position of the swash plate. A control valve sets the actuator pressure. Like in normal pumps, this control valve can be a pressure control, a flow control, a power control or a combination of these controls. For testing the pump at specific predefined and reproducible swash plate conditions, the normal procedure is to connect the actuators to the low-pressure side of the pump. The bias spring and the bias pistons will then rotate the swash plate to a larger angle until a mechanical end stop is hit.

The mechanical lock up not only defines the swash angle; it also stops the oscillation of the swash plate. This is unlike real pump operation in which the swash angle is controlled by means of hydraulic cylinders.

The aim of this study is to measure the pump performance and characteristics like in real operation. On the other hand, a system is needed which allows a clearly defined and reproducible swash angle. This is achieved by implementing small so-called ß-valves (see Figure 2). When the bias piston forces the swash plate to move to a larger angle, the ß-valve opens and supplies oil to the actuators. The pressure in the actuator cylinders increases, which counteracts the torque from the bias piston and cylinder. The ß-valve is operated by a pin, which pushes the check valve of the ß-valve to an open position. The length of the valve pin defines the angle at which the pump is operated. In this study 4 different pin pairs are used defining swash angles of about 2°, 4°, 6° and 8°.

The actuators are not only used to set the swash angle but also counteract the torque load of the rotating group on the swash plate. The strong and dynamic variation of the torque load results in an equally strong and dynamic variation of the pressure level in the actuators. This is the reason why the swash plates need to oscillate. The movement of the swash plate results in a displacement of the actuator volume, which is then used to change the pressure level in the actuators.

The variable displacement floating cup pump has two of these oscillating swash plates. Since the pistons of the left side of the rotor are positioned in between the pistons of the right side of the rotor, the two sides of the pump are operated in counter phase. This counter phase operation is used to connect the actuator systems on the left and right side of the pumps. Whenever the actuators of the left side of the pump make a delivery stroke, the actuators on the right side make a suction stroke, and vice versa. Via the connecting orifice, oil is simply transferred from one side to the other (Figure 2). The orifice in the connecting line (2 x Ø0.7 mm in series) determines the pressure variation in the actuators in relation to the rocking movement of the swash plates.

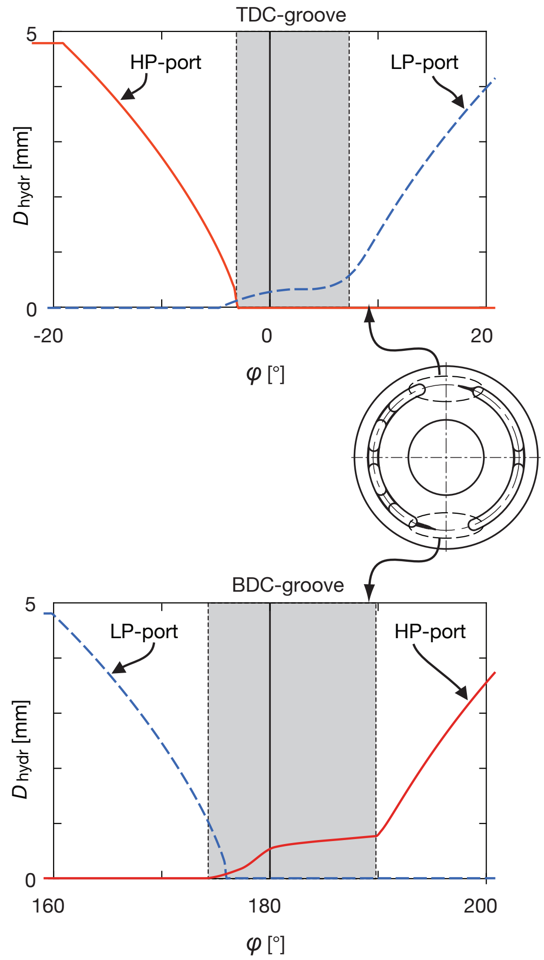


Fig. 3: Hydraulic diameter of the opening area between the ports of the barrel and the ports and silencing grooves of the port plate. The grey areas in the middle indicate the position of the silencing grooves

Like in conventional pumps and motors, silencing grooves are applied to soften the commutation in the top and bottom dead centres. Figure 3 shows the hydraulic diameter of the opening area between the ports of the barrel and the ports and grooves of the port plate. The dimensions and geometry of the silencing grooves have a strong effect on the torque load, which the barrel exerts on the swash plate. The design shown in figure 3 is optimised assuming a locked swash angle *ß*.

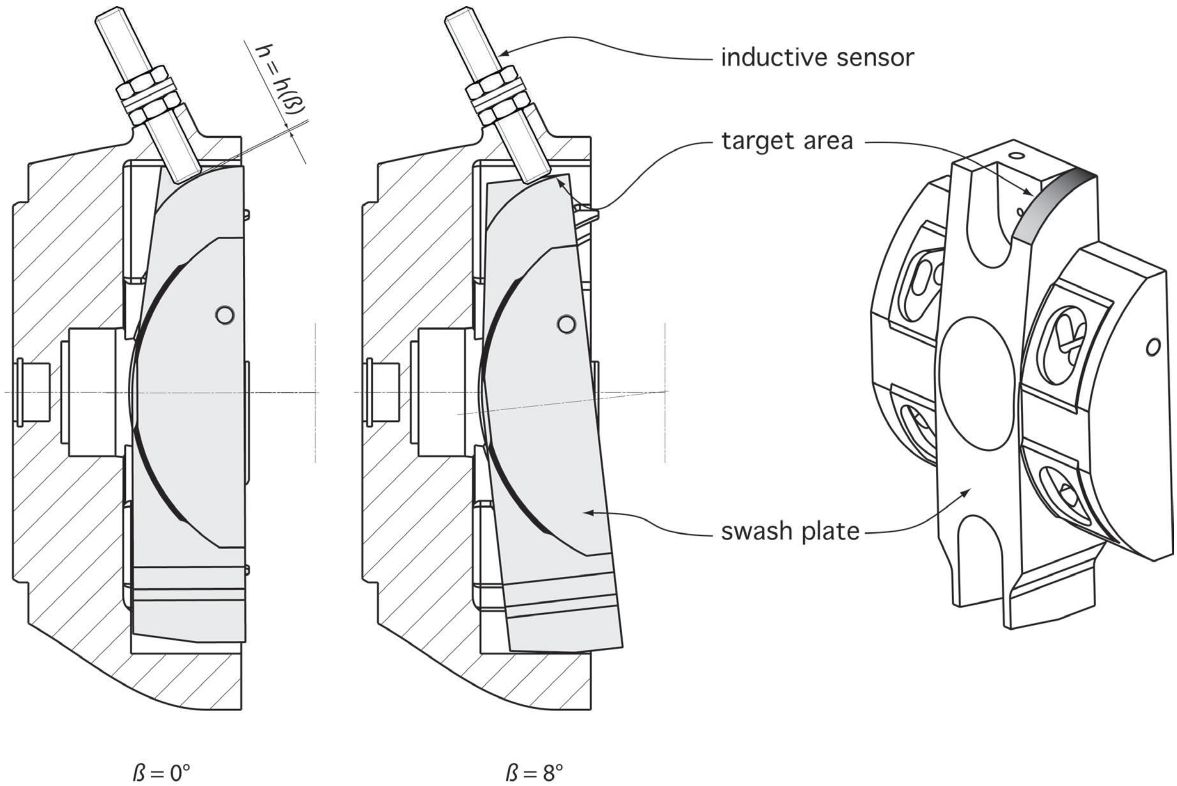


Fig. 4: Cross section of part of the housing (at ß = 0° and ß = 8°), showing the position of the inductive sensors and the curved target area on the swash plate. The gap height h varies as a function of the swash plate angle ß

Of this pump, the movement of the swash plates and the pressure in the actuator systems have been measured at 4 different swash angles (2°, 4°, 6° and 7.8°), 6 different operating speeds (500, 1000, 1500, 2000, 2500 and 3000 rpm) and 7 different pump pressures (50, 100, 150, 200, 250, 300 350 bar). The FCVP is designed for a maximum swash angle of 8°. This is the angle were the swash plates will be stopped mechanically. Since the objective of this study was to study the pump behaviour when the swash plates are oscillating, the maximum swash angle is reduced to about 7.8°, thereby pushing the swash angle away from the mechanical end stop at 8°.

The movement of the swash plates has been measured by means of inductive sensors (one for each swash plate), as is illustrated in Figure 4. The pressure in the actuator cylinders has been measured by means of fast response piezoresistive pressure sensors.

In all operating points, the efficiency of the pump has been measured at Eindhoven University of Technology, for two different conditions:

- the swash plates are locked mechanically and the swash plates can’t oscillate
- the swash angle is set by means of the ß-valves and the swash plates are free to oscillate

At the end of this paper the two sets of measurements are compared. The measured swash angle and pressure level of the actuators is compared to the results from a simulation model.

3 Simulation model

In most analysis of variable displacement axial piston pumps, the swash angle is considered to be a constant. The calculation of the pressure in each of the cylinders of the rotating group is a matter of combining the sinusoidal movement of each piston with a detailed model of the port plate geometry (the upper part of Figure 5).

The variation of the displacement volumes of the rotating group is also a function of the swash angle *ß*. If the swash angle varies during one revolution –as is postulated in this study– the variation of the displacement volumes is no longer sinusoidal. An extra movement is added to the sinus curve. The source of this extra movement is the torque load of the pistons of the rotating group acting on the swash plate. The rotational speed and the number of pistons rotating on the barrel determine the prime frequency of the variation of the torque load. In case of 12 pistons per barrel (as is the case in the floating cup pump) the swash plate torque will change periodically 12 times per revolution.

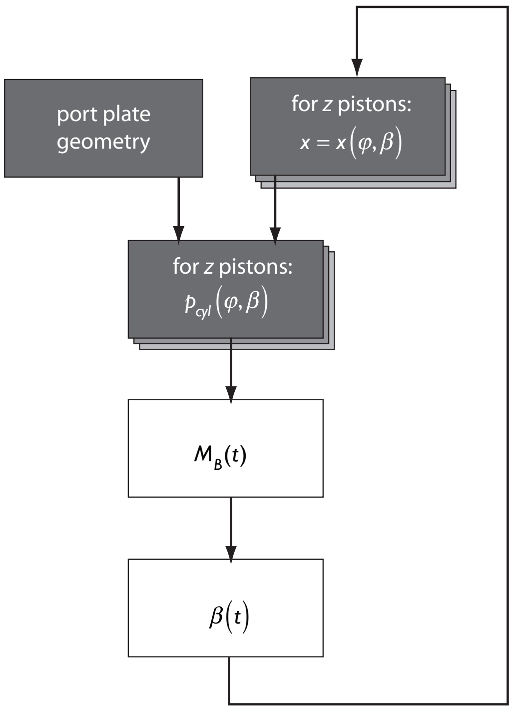


Fig 5: Model of the cylinder pressure assuming a constant swash angle β (the grey boxes). The white boxes show the extension of the model in case the swash plates are free to oscillate and create an additional oscillation of the displacement volumes of the rotating group.

The torque load causes a rocking movement of the swash plate and the swash angle β will no longer be constant but vary as a function of time. The actuators counteract this rocking movement. The amplitude of $\beta(t)$ is, amongst others, determined by the amplitude of the torque load on the swash plate, which in turn is strongly dependent on the commutation of the displacement volumes in the top and bottom dead centres. As a result, there is a circular dependency in which the commutation influences the oscillating movement of the swash plate, and the oscillating movement influences the commutation.

The interaction between swash plate oscillation and commutation has a significant effect on the behaviour of the pump. Figure 6 shows, for four different operating speeds and a pump pressure of 300 bar, a comparison of the calculated torque load of the barrel on the swash plate in case of a constant swash angle compared to the situation in which the swash plate is free to oscillate.

The oscillation of the swash plate softens the saw tooth shaped torque curve, which is typical for a situation in which the swash plate is not free to oscillate but is locked mechanically. This is also of importance for all studies in which the torque load on the swash plate is averaged during one revolution of the barrel. The oscillation of the swash plate creates a lower average value at low rotational speeds and a higher average torque load at high operating speeds.

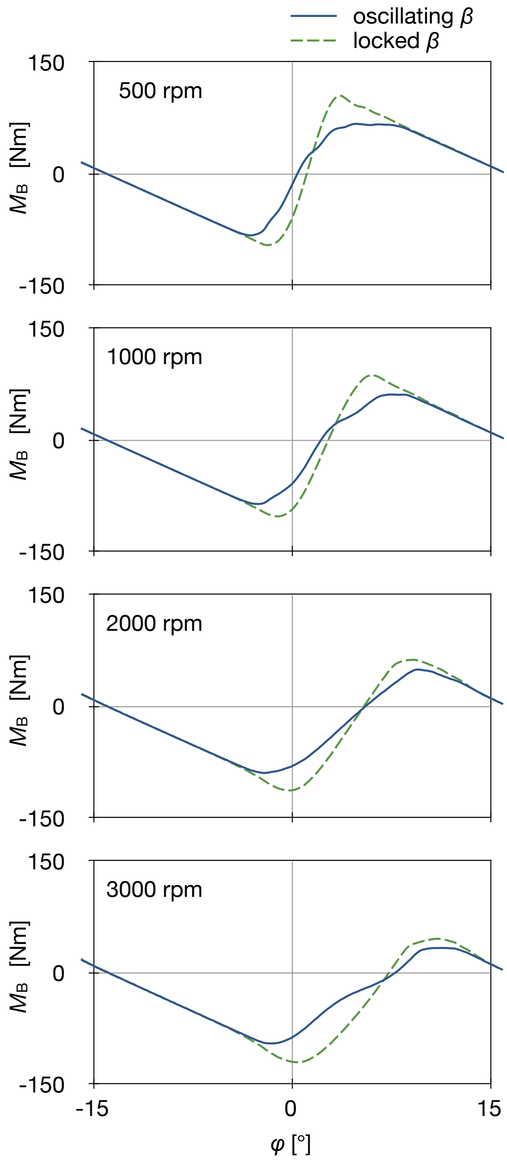


Fig. 6: Calculated torque load M_B of the barrel on the swash plate for a locked swash plate at an absolutely constant swash angle and a free oscillating swash plate (p_1 = 300 bar, p_0 = 10 bar, T_0 = 55°C, β = 7.8°)

4 Validation of the simulation model

The simulation model is validated by comparing the measured and calculated results of the swash angle and the pressure in the actuator system in a wide range of operating conditions. Figures 7 and 8 show the outcome for the maximum swash angle at a pump pressure of 300 bar. The pump was operated with a supply pressure of 10 bar and an oil temperature at the inlet of 50°C. The time scale shown on horizontal axis is made dimensionless by dividing the time by the characteristic period T of 1/12 of a revolution (12 being the number of pistons per barrel):

$$T = \frac{60}{12 \cdot n} \tag{1}$$

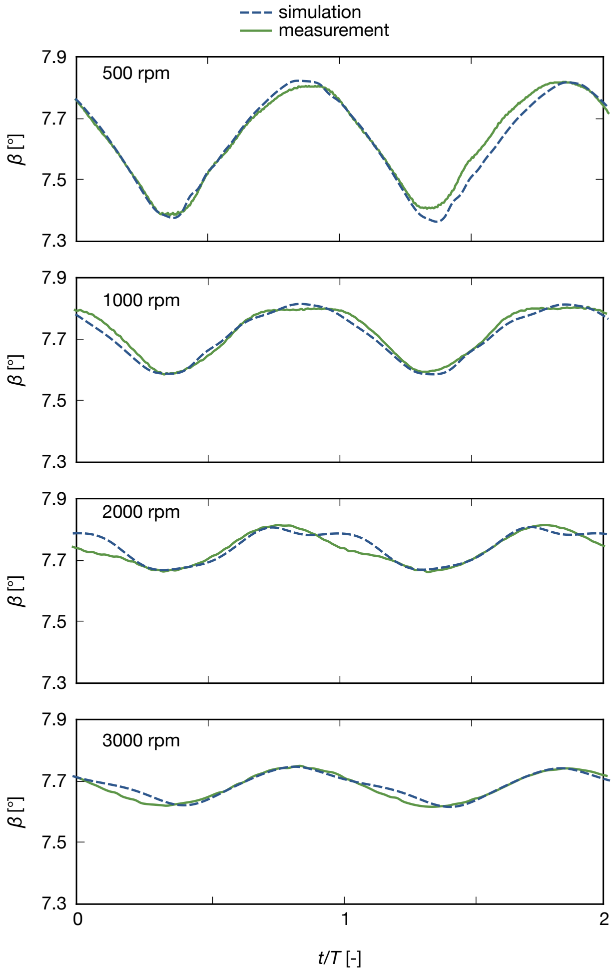


Fig. 7: Calculated and measured swash angle (p_1 = 300 bar, p_0 = 10 bar, T_0 = 50°C, β' = 7.8°).

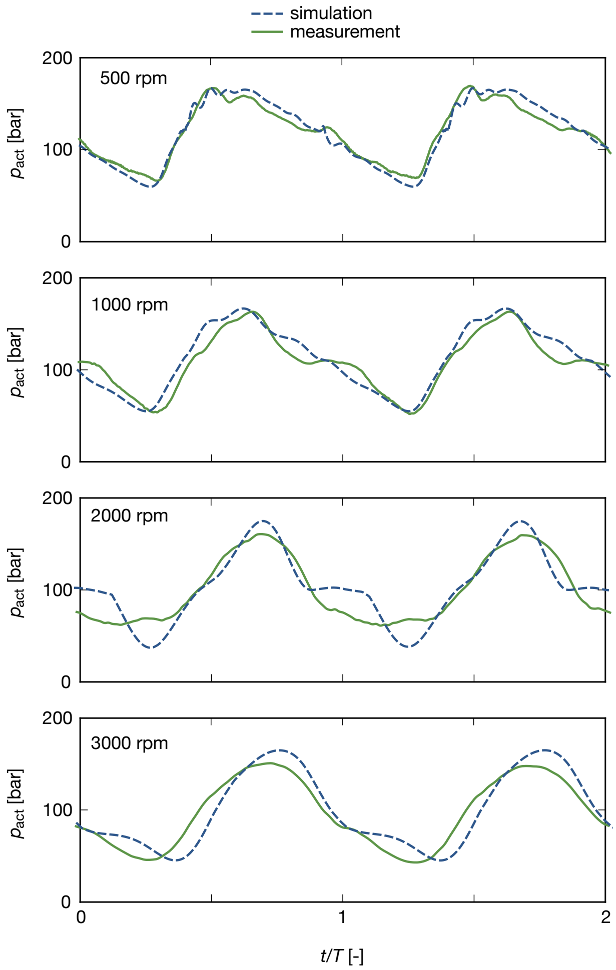


Fig 8: Calculated and measured pressure in the actuator system (p_1 = 300 bar, p_0 = 10 bar, T_0 = 50°C, β' = 7.8°)

There is a good correlation between the simulations and the measurements. The simulation model allows an adequate prediction of the dynamic behaviour of the swash plate and the pressure variation in the actuator system in a wide range of operating conditions. The model correctly calculates a larger amplitude of the swash plate oscillation at lower operating speeds of the pump. The most important difference between the simulation and measurement results occurs around 2000 rpm, where the simulation model forecasts a larger amplitude of the actuator pressure and a somewhat different swash angle variation.

5 Influence of pump pressure, operating speed and swash angle

The variable displacement floating cup pump has two swash plates, one on the mount side and one on the cover side. The pistons on the cover side are positioned in between the pistons on the mount side. As a result, the commutation, the torque load on the swash plates and the resulting dynamic behaviour of both actuator systems are all out of phase.

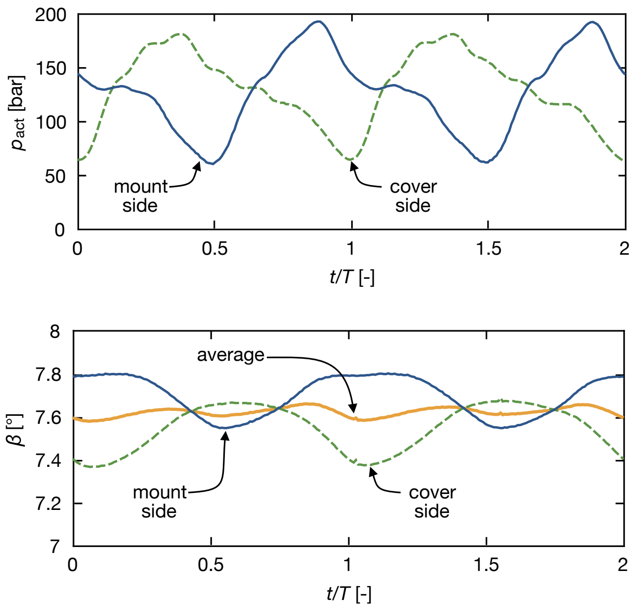


Fig. 9: Measured actuator pressure and swash angle for both sides of the pump (n = 1000 rpm, p_1 = 350 bar, p_0 = 10 bar, T_0 = 50°C, β' = 7.8°)

This can clearly be seen in the measurement of the actuator pressure and the swash angle (Figure 9). The experiment is performed for an opening position of the ß-valve, at *β'* = 7.8°. Although each swash plate is oscillating with an amplitude of about 0.3°, the average swash angle of the two sides of the pump is much more constant. This is an advantage of the mirrored out-of-phase operation of the floating cup principle.

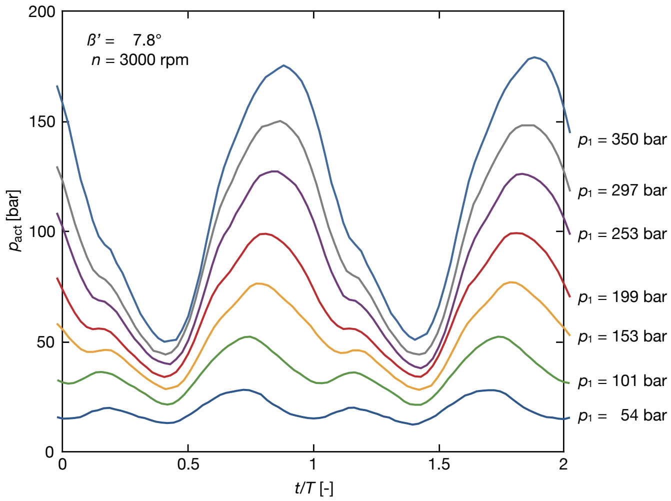
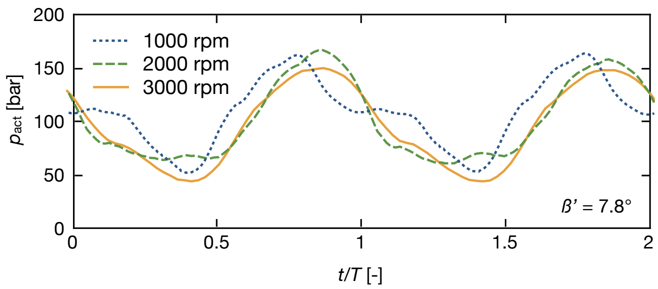
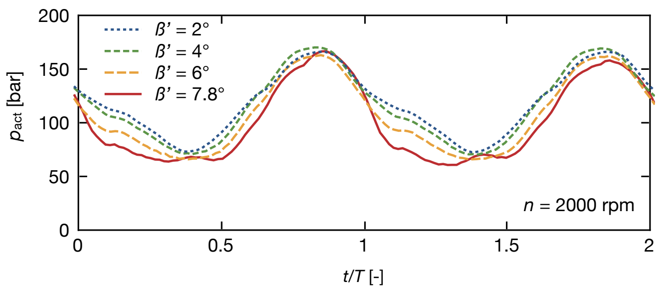


Fig. 10: Measured influence of the delivery pressure of the pump on the pressure in the actuator system (n = 3000 rpm, p_0 = 10 bar, T_0 = 50°C, β' = 7.8°)



a) Influence of the operating speed n



b) Influence of the swash angle ß

Fig. 11: Measured influence of the rotational speed (a) and the swash angle (b) on the pressure in the actuator system (p_1 = 300 bar, p_0 = 10 bar, T_0 = 50°C)

The pressure variation in the actuator system is strongly influenced by the delivery pressure of the pump (see Figure 10). This is to be expected: a higher pump pressure creates a larger torque load on the swash plate, which has to be counteracted by a higher pressure and pressure variation in the actuator system. The rotating pressurised cylinders of each barrel determine the torque load of the rotating group. Aside from the commutation, the hydrostatic forces from the barrel are to a much lesser extend related to the operating speed or the swash angle. As a result the amplitude of the actuator pressure is almost independent of the operating speed *n* and the swash angle *ß*, as can be seen in Figures 11a and 11b.

The relationship between the oscillating pressure in the actuator system and the hydrostatic force generated by the barrel can be explained by means of the forces shown in Figure 12. There are many forces acting on the swash plate, such as the force from the bias spring, the hydrostatic force from the ß-valve and the friction force in the swash plate bearing, but the most dominating forces are the hydrostatic force F_b from the barrel, the force F_{bias} from the bias cup, and the two actuator forces F_{act}. All of these forces create a moment around the swivel axis of the swash plate. The torque generated by the barrel fluctuates around zero, being alternating positive and negative. The bias cup adds an almost constant torque load and pushes the total moment of the barrel and the bias cup above the zero axis. The calculated effects of the operating pressure and speed on the torque load of the swash plate can be seen in Figure 13. Both the average torque and the torque variation have an almost linear relationship with the delivery pressure of the pump. The operating speed only has a mild influence on the torque load, most and for all because of the influence of the pump speed on the commutation.

The actuator system has to counteract the load from the barrel and the bias piston. There are three parameters that determine the torque created by the actuators:

- the effective arm length of the actuators
- the hydrostatic area of the actuator cups
- the pressure level in the actuators

Both the cup area and the arm length can be considered to be constant. Therefore, a variation of the torque created by the actuators can only be achieved by a variation of the pressure level in the actuator system. This variation needs to correspond with the strong and dynamic variation of the torque created by the bias cup and the barrel. Figure 15 shows the average and the amplitude of the measured values of the pressure in the actuator system. The measured values show the same trend as the calculated values for the torque load (Figure 14).

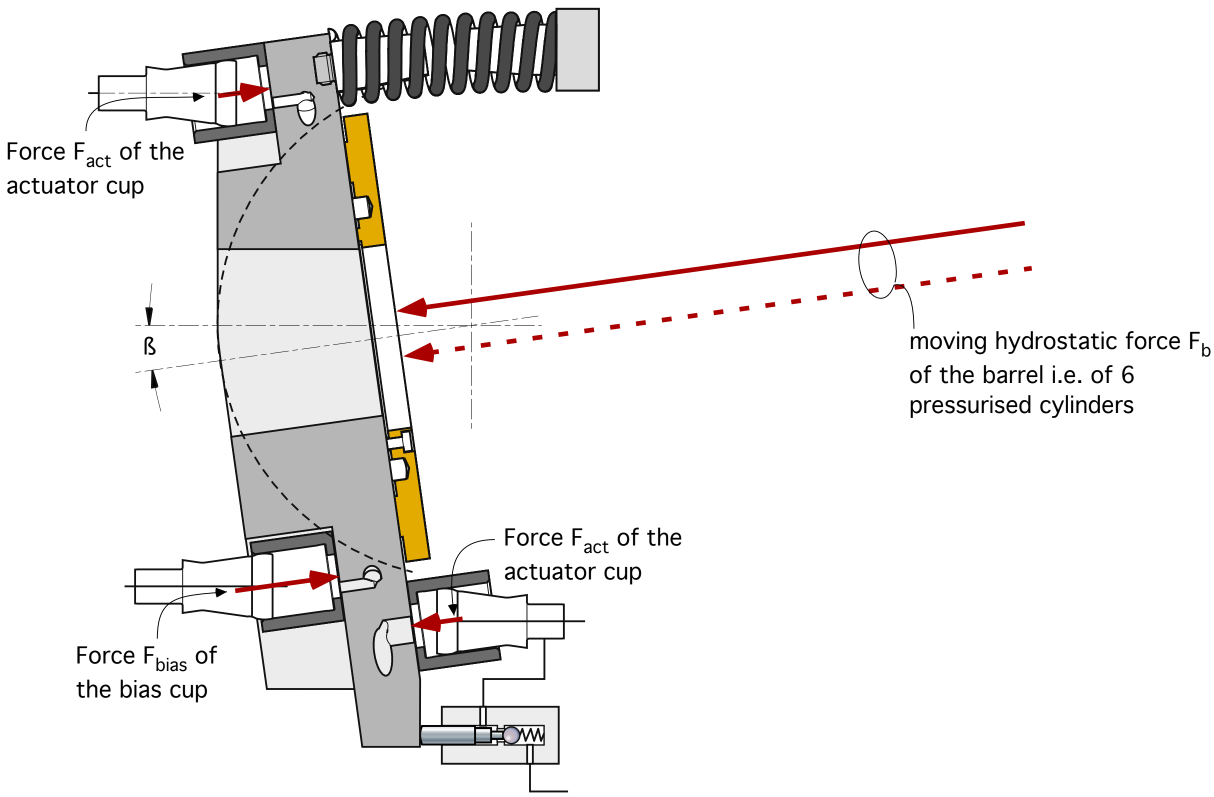


Fig. 12: Most important forces that determine the torque balance of the swash plate

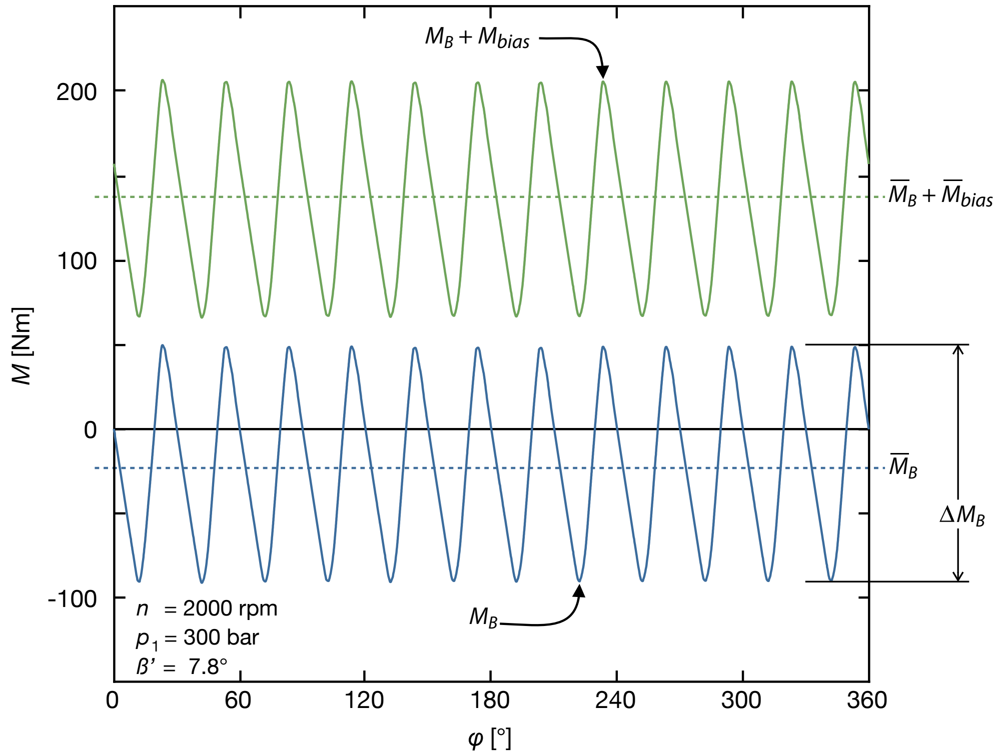
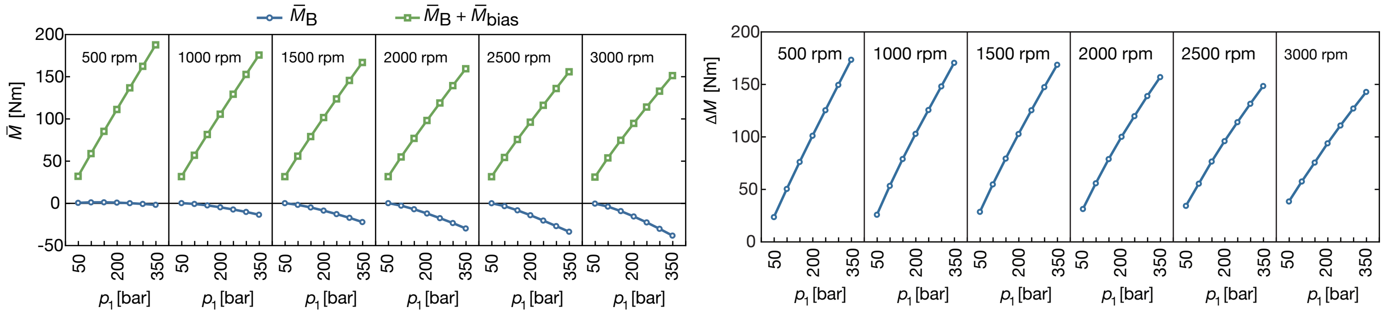
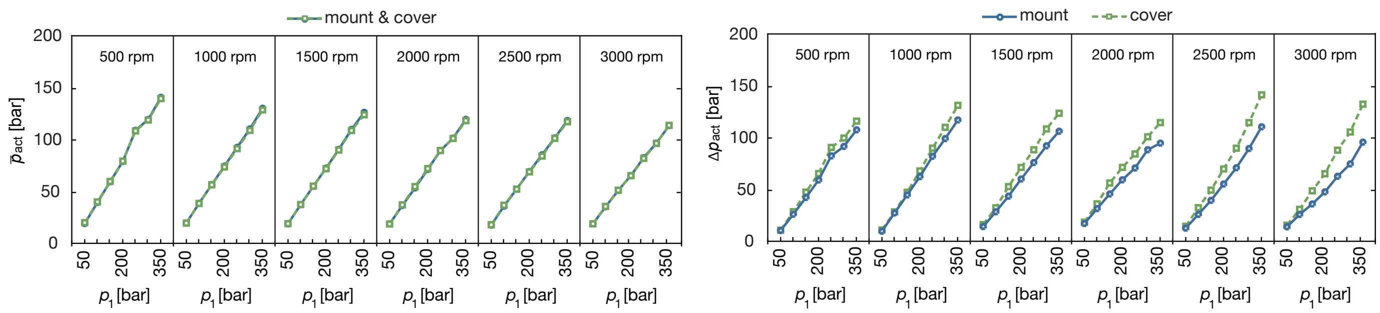


Fig. 13: Calculated torque load of the barrel M_B and of the barrel plus the bias cup (M_B + M_{bias}) for one barrel revolution (n = 2000 rpm, p_1 = 300 bar, p_0 = 10 bar, $ß'$ = 7.8°)



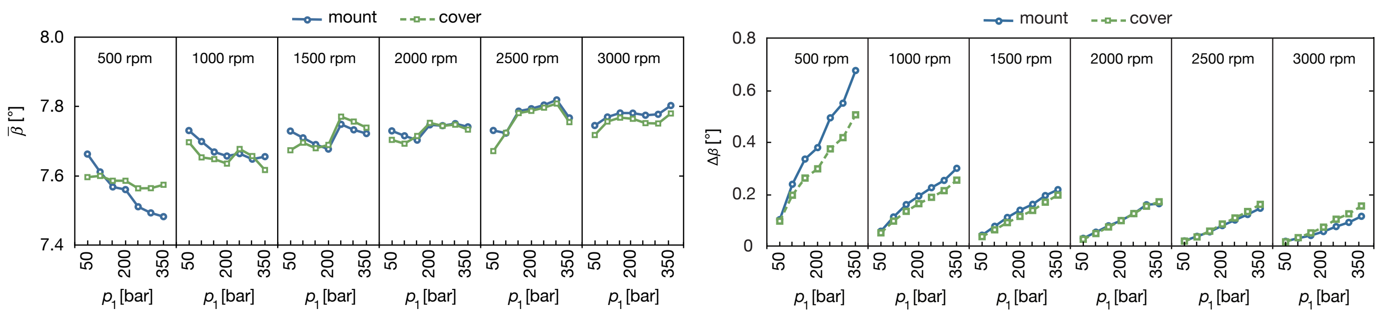
a) Average torque load *b) Amplitude of the torque load*

Fig. 14: Calculated effect of the operating speed and delivery pressure of the pump on the average torque load and the amplitude of the torque load on the swash plate (p_0 = 10 bar, ß' = 7.8°).



a) Average actuator pressure *b) Amplitude of the actuator pressure*

Fig. 15: Measured mean pressure and pressure amplitude of the actuator system for various operating pressures and speeds (p_0 = 10 bar, T_0 = 50°C, ß' = 7.8°)



a) Average swash angle ß *b) Amplitude of the swash angle ß*

Fig. 16: Measured influence of the delivery pressure and the operating speed on the average swash angle and the peak-to-peak amplitude of the swash angle (p_0 = 10 bar, T_0 = 50°C, ß' = 7.8°).

In order to generate a strong and dynamic variation of the pressure level in the actuator system, the swash plates have to oscillate, thereby forcing oil in and out of the actuator cups, through the centre restriction in the line that connects both actuator systems (see Figure 2). The Δp of the resistance is dependent on the flow, which is generated by the rocking movement of the swash plates. This flow is directly related to the operating speed of the pump. To compensate for the reduced frequency at lower speeds, the swash plate system responses by increasing the amplitude of the swash plate (Figure 16b), thereby creating again the desired flow through the connecting orifice. This effect can be seen in Figure 17, which shows the oscillation of the swash plate relative to the setting point of the ß-valve for three different operating speeds of the pump.

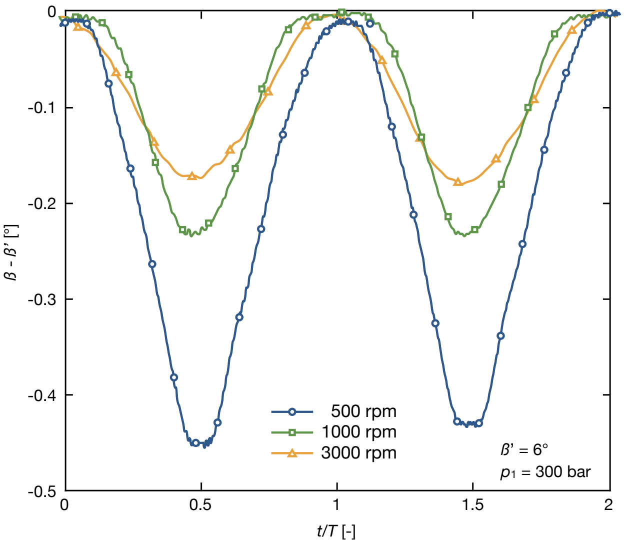


Fig. 17: Measured swash plate oscillation, relative to the opening position ß' of the ß-valve, for three different pump speeds (p_1 = 300 bar, p_0 = 10 bar, ß' = 6°, T_0 = 50°C)

The effect of the increased amplitude of the swash plate oscillation at reduced pump speeds can also be seen in Figure 16b. Since the ß-valves define the maximum value of the oscillation, the average swash angle (Figure 16a) reduces when the oscillation amplitude increases, i.e. at low operating speeds. This implies that the pump displacement, when operated with the ß-valves, is not a constant value, despite the mechanically defined opening position of the ß-valves.

6 Effects on the commutation

The oscillating movement of the swash plate also creates an additional movement of the barrel, and therefore affects the cylinder volume of each cup of the rotating group. The effect of the swash plate oscillation is largest in the top and bottom dead centres where the commutation occurs. Consequently, the swash plate oscillation also influences the compression and expansion of the oil in the cups. Figures 18 and 19 show the results of the simulation for two pump speeds (500 and 3000 rpm) and for two different operating conditions:

- Assuming a fixed swash plate, locked at a constant swash angle of *ß* = 7.8°
- Simulating a real pump operation with an oscillating swash plate, having an opening position of the ß-vale of *ß'* = 7.8°

Both simulations have been performed for a delivery pressure of 300 bar. The grey area in the middle of each diagram shows the position of the silencing grooves (see Figure 3).

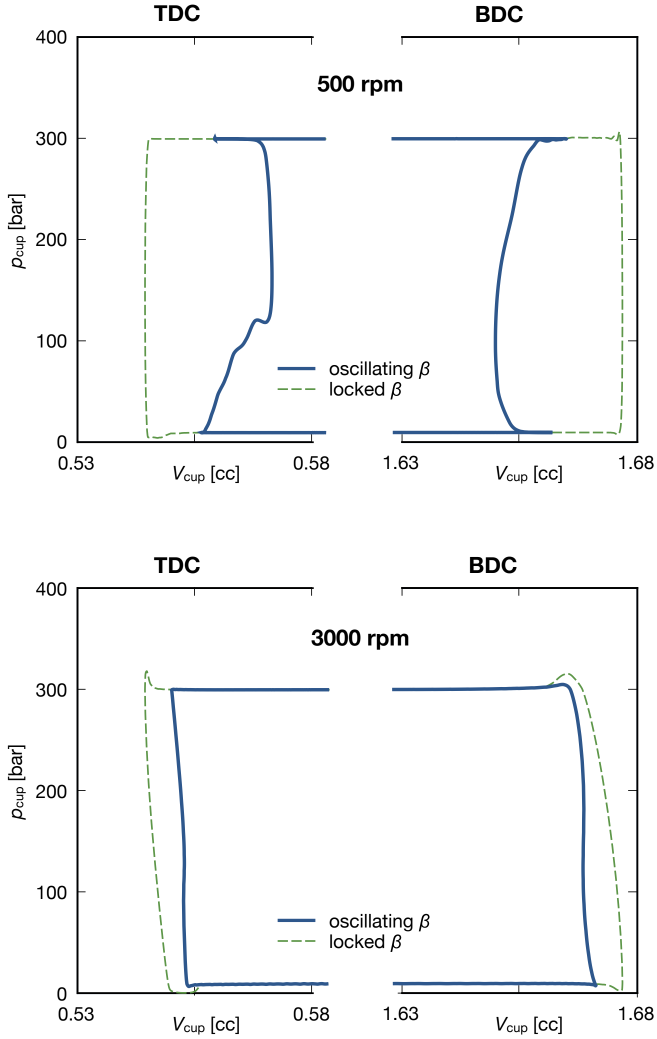
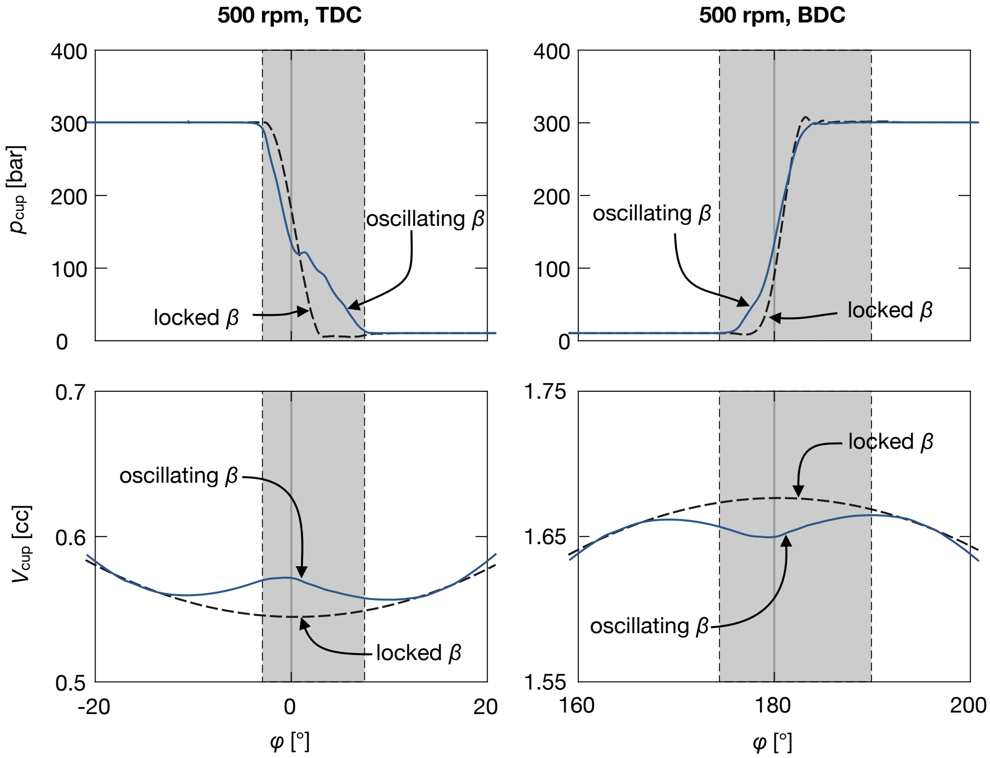
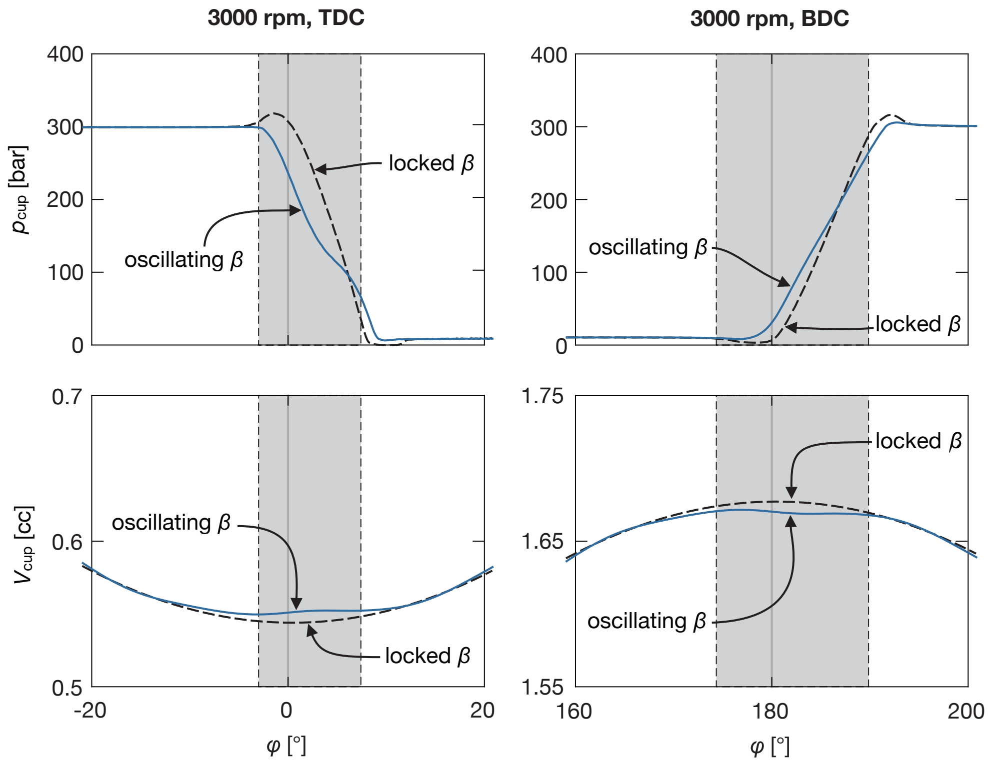


Fig. 18: Calculated effect of the swash plate oscillation on the pV-diagram in and around the top and bottom dead centres at two different pump speeds (p_1 = 300 bar, p_0 = 10 bar, ß' = 7.8°, T_0 = 50°C)

In the areas where the commutation occurs, the additional displacement, which is created by the swash plate oscillation, is in most operating points larger than the base sinusoidal movement. The results shown in Figures 18 and 19 are calculated for a situation in which the pump is operated at full displacement. At smaller displacements, the sinusoidal movement will be further reduced, whereas the swash plate oscillation is hardly affected by the swash angle. As a result the effect of the swash plate oscillation is even larger at smaller swash angles. Generally, the effect of the swash plate oscillation is largest at low operating speeds, high pump pressures and small swash angles. The influence of the swash plate oscillation should be taken into account in the design of the port plates and the silencing grooves. This has not yet been considered in the design of the port plates shown in this paper.

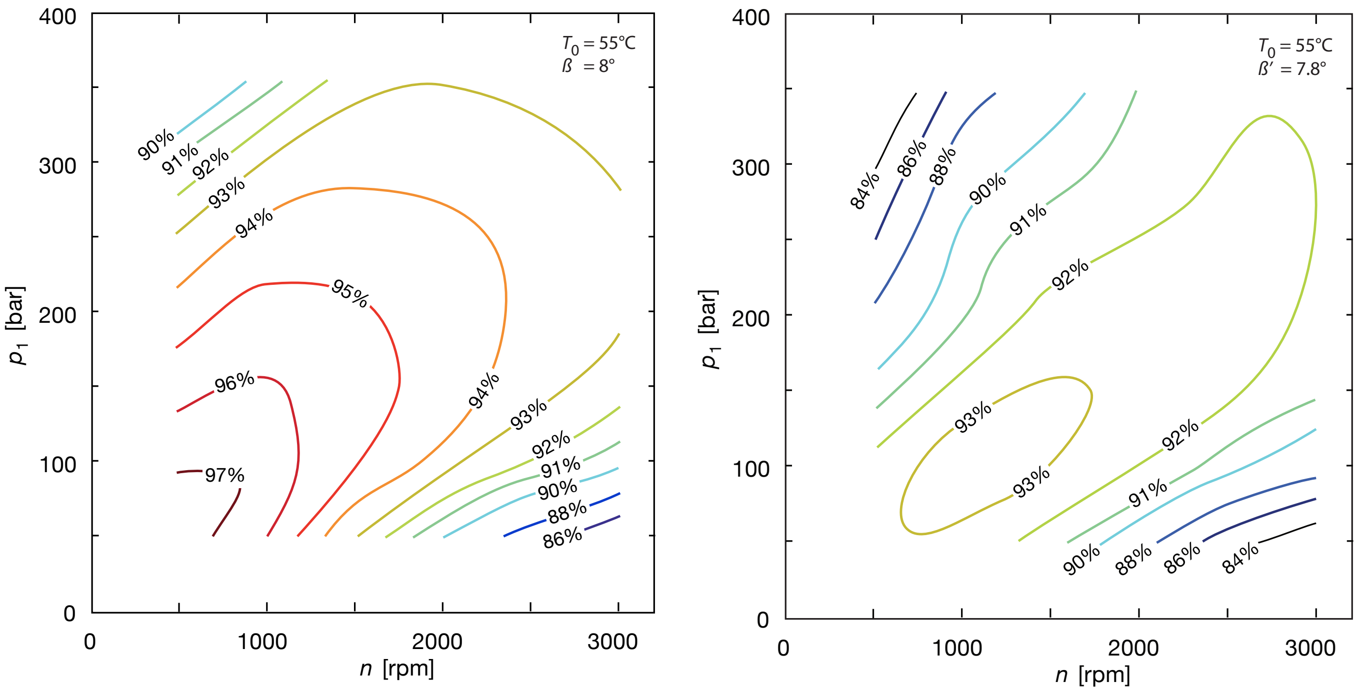


a) pump speed n = 500 rpm



b) pump speed n = 3000 rpm

Fig. 19:Calculated cup pressure and volume in case of a locked swash angle β and in case of a free oscillating swash angle β. The grey areas in the middle of each diagram indicates the position of the silencing grooves (n = 3000 rpm, p_1 = 300 bar, p_0 = 10 bar, β' = 7.8°)



a) Locked swash plate with constant ß (ß = 8°)

b) Oscillating swash plate (ß' = 7.8°)

Fig. 20: Measured total efficiency of the variable displacement floating cup pump (Measurements performed at Eindhoven University of Technology)

7 Effects on the efficiency

The efficiency and performance of the pump have been tested in a wide range of operating conditions and swash angles at Eindhoven University of Technology. The results for the total efficiency of the pump at full displacement are presented in Figure 20. Two situations have been examined:

a. the pump is operated as conventional against a mechanical end stop,
b. the swash plates were free to oscillate and the swash plate position was controlled by means of the ß-valves.

The oscillation of the swash plate and the opening position of the ß-valve at *ß'* = 7.8° resulted in an average swash angle of 7.7°, which is about 4% smaller than the situation in which the pump was operated at a constant non-oscillating swash angle of *ß* = 8°. Nevertheless, the two measurements will be compared further on.

Figure 21 presents the difference between the total efficiency of the two measurements. According to the test results, the oscillation of the swash plates results in a reduction of the efficiency of up to 6%.

About half of the efficiency reduction is due to increased leakage, possibly of the actuator pistons (which are pressurised in situation b.), but also because of leakage of the ß-valves. The other half is for the largest part due to the flow restriction of the orifices in between both actuator systems. Both power losses are approximately linearly dependent of the pump pressure but nearly independent of the rotational speed.

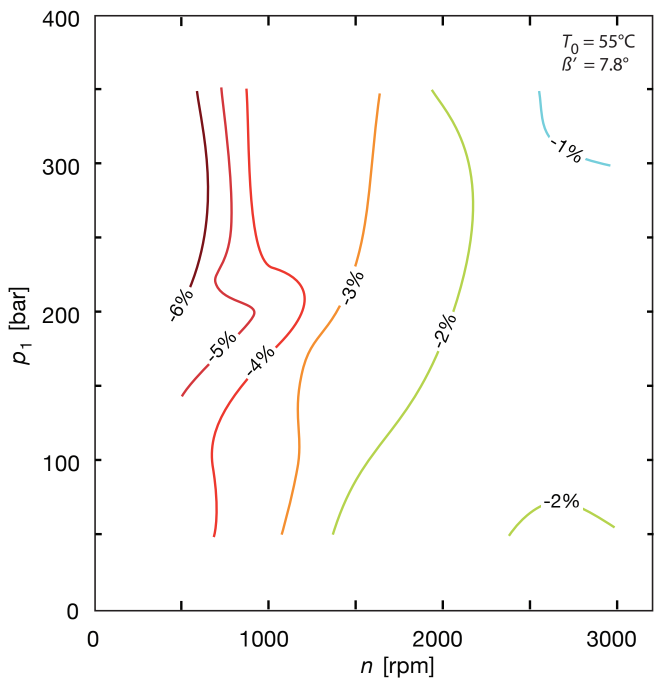


Fig. 21: Effect of the swash plate oscillation on the total efficiency of the pump, measured at full pump displacement (ß ≈ 8°)

This corresponds with the measured reduction of the overall efficiency (Figure 21): a reduced pump speed results in a reduced pump power, whereas the additional power losses due to the pump oscillation are nearly independent of the pump speed.

The simulation model also allows a calculation of the flow losses in the restriction of the silencing grooves of the port plates. However, the simulations show no significant influence of the swash plate oscillation on the dissipation in the silencing grooves.

The data of Figures 20 and 21 were acquired for a pump running at full displacement. The measurements and simulations show that the extra leakage and flow losses from the actuator system are not influenced by the pump displacement i.e. the swash angle of the pump. However, the pump power is linearly dependent on the pump displacement. Therefore, the relative effect of the swash plate oscillation (i.e. the effect on the overall efficiency) is much stronger at small displacements. This can be seen in Figure 22, which is the same diagram as in Figure 21, but now based on a pump operation at 25% displacement, i.e. at $\beta' = 2°$. Now, the efficiency reduction, which is caused by the operation of the actuator system, is about four times as high as at maximum pump displacement.

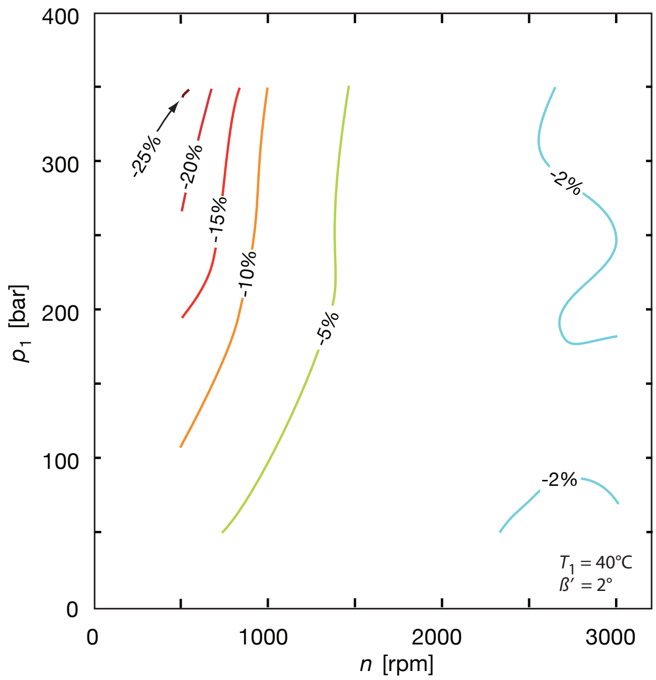


Fig. 22: Effect of the swash plate oscillation on the total efficiency of the pump, measured at 25% pump displacement ($\beta' = 2°$)

It will certainly be possible to reduce the extra leakage of the actuator system and the ß-valves. It is estimated that the additional losses can be halved. But even then, the displacement control and oscillation of the swash plate will have a significant effect on the efficiency, especially at low operating speeds and reduced pump displacements.

8 Conclusions

In variable displacement pumps and motors, the swash plate oscillates during normal operation. Yet, it is common to treat the variable displacement machine as a constant displacement machine when analysing the efficiency, noise or pulsations. The swash plate is then rotated until a mechanical end stop is reached. As a consequence, the normal dynamic oscillation of the swash plate is suppressed and the losses and effects caused by the oscillation and the swash plate control are disregarded.

In this study, the performance of a variable displacement floating cup pump is measured and analysed, while allowing the swash plate to respond to the dynamic torque loads. A simulation model has been created to get a better understanding of the dynamic behaviour of the swash plate oscillation and its effects on commutation and pump efficiency.

From the study it can be concluded:

- The amplitude of the oscillation of the swash plate increases proportionally with the delivery pressure of the pump. The amplitude is largest at low pump speeds and high pump pressures. The pump displacement (i.e. the average swash angle) does not influence the swash plate oscillation.
- The oscillation of the swash plate causes an additional displacement of the individual cylinders of the rotating group. While passing the silencing grooves this additional movement is larger than the base sinusoidal movement. In the design of port plates and silencing grooves the effect of the swash plate oscillation should therefore be taken into account.
- The oscillations of the swash plate and the control system of the swash angle have a significant effect on the overall efficiency of the pump, especially at low pump speeds and reduced displacements.

Although not reported in this paper, the noise and pulsation levels of the pump are also strongly influenced by the new test procedure.

There are many differences between the floating cup pump –which is studied in this paper– and conventional state-of-the-art slipper type and bent axis pumps and motors. However, previous research [1-6] has already proven that also in conventional pumps and motors a similar dynamic behaviour occurs. It is strongly recommended to investigate the effects of the swash plate oscillation and the swash plate control system in these conventional pumps and motors.

If, as expected, the swash plate oscillation influences the commutation, the efficiency, the noise level and the pulsation level, then the common procedure to lock the swash plate must be abandoned, also for the analysis of conventional piston pumps and motors. It is also certain that the power losses of the control valves should be included in the determination of the efficiency. The standards, which describe the general procedure for determine the performance of hydrostatic machines, should be changed accordingly.

Nomenclature

Designation	Denotation	Unit
D_{hydr}	hydraulic diameter of the opening area between the barrel port and the ports and silencing grooves of the port plate	[mm]
F_b	axial barrel force	[N]
F_{bias}	force bias cup	[N]
F_{act}	force actuator cup	[N]
n	rotational speed of the pump shaft	[1/min]
p_0	supply pressure of the pump	[bar]
p_1	delivery pressure of the pump	[bar]
p_{act}	pressure actuator system	[bar]
p_{cup}	pressure in a cup of the rotating group	[bar]
M_{B}	torque load of the barrel on the swash plate	[Nm]
t	time	[s]
T_0	oil temperature at the supply side	[°C]
T	duration of 1/12 of a revolution	[s]
V_{cup}	oil volume in the cup of the rotating group	[cm^3]
x	displacement	[mm]
z	Number of pistons of the pump	[-]
$ß'$	swash angle were the ß-valve opens	[°]
$ß$	swash angle	[°]
φ	rotational position of the barrel	[°]

References

[1] K. Inoue, and M. Nakasato, Study of the operating moment of a swash plate type axial piston pump, First report: Effects of dynamic characteristics of a swash plate angle supporting element on the operating moment, *The Journal of Fluid Control,* Vol. 22, Issue 1, p 7-27, 1994

[2] K. Inoue, and M. Nakasato, Study of the operating moment of a swash plate type axial piston pump, Second report: Effects of dynamic characteristics of a swash plate angle supporting element on the cylinder pressure, *The Journal of Fluid Control,* Vol. 22, Issue 1, p 30-45, 1994

[3] J.W. Dobchuk. Sub-component Level Modeling of a Variable Displacement Axial Piston Pump, Thesis University of Saskatchewan, Saskatoon, Canada, December 2000

[4] J.W. Dobchuk, P.N. Nikiforuk, P.R. Ukrainetz, R.T. Burton. "Effect of internal pump dynamics on control piston pressure ripple, Proc. of *the 6th Triennial International Symposium on Fluid Control, Measurement and Visualization*, Sherbrooke, Canada, August 13-17, 2000

[5] S.A. Kassem, M.K. Bahr. On the dynamics of swash plate axial piston pumps with conical cylinder blocks", Proc. of the *6th Triennial International Symposium on Fluid Control, Measurement and Visualization*, Sherbrooke, Canada, August 13-17, 2000

[6] M.K. Bahr, J. Svodoba, R.B. Bhat. Vibration analysis of constant power regulated swash plate axial piston pumps", *Journal of sound and vibration*, 2003, Vol 259(5), p. 1225-1236, Elsevier Science Ltd.

[7] P.A.J. Achten, T.L Van den Brink, T. Paardenkooper, T. Platzer, H.W. Potma, M.P.A. Schellekens and G.E.M. Vael. "Design and testing of an axial piston pump based on the floating cup principle", *Proc. 8th Scandinavian Int. Conf. on Fluid Power SICFP'03*, Vol. 2, Tampere University of Technology, pp. 805-820, 2003

[8] P. Achten. Dynamic high frequency behaviour of the swash plate in a variable displacement, axial piston pump, *Proc. of the IMechE, Part I: Journal of Systems and Control Engineering*, Volume 227 Issue 6 July 2013 pp. 529 – 540, doi: 10.1177/0959651813483419

[9] P.A.J. Achten, "Designing the impossible pump", *Proc. Hydraulikdagar* 2003, Linköping University, Sweden 2003

[10] P.A.J. Achten, T. van den Brink, M. Schellekens, (2005) Design of a variable displacement floating cup pump, *Proc. 9th Scandinavian Int. Conf. on Fluid Power, SICFP'05*, Linköping, Sweden, 2005