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'SHUTTLE' TECHNOLOGY FOR NOISE REDUCTION AND EFFICIENCY IMPROVEMENT OF HYDROSTATIC MACHINES - PART 2

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ABSTRACT

In 2001, INNAS introduced the 'Shuttle' technology for noise reduction and efficiency improvement of hydrostatic machines. The current study revisits this technology for application in hydrostatic pumps and motors.

In many hydrostatic pumps and motors, commutation is imposed by a fixed component like a valve plate. Designing a valve plate (or comparable component) that ensures good commutation at one specific operating condition, is fairly simple. However, an inherent problem of such a component is that it should ensure good commutation at all of the operating conditions.

In an attempt to minimise losses and reduce noise emission caused by improper commutation, so-called shuttles were introduced by INNAS in 2001. These shuttles act as small pistons between two working chambers, essentially providing a connection to the ports while the valve plate is still closed. In theory, this will result in a check-valve like commutation.

In the original paper, shuttles were implemented in a hydraulic transformer. This paper discusses and analyses the use of shuttles in pumps and motors. Simulation results show that the introduction of shuttles can reduce commutation losses to negligible levels. Furthermore, the results suggest that the use of shuttles could also reduce noise emissions.

1 INTRODUCTION

By nature, all hydrostatic pumps and motors have some form of commutation: the transition from the supply port to the discharge port of the machine (and vice versa). In a recent study, the process of commutation was shown to be a significant source of power loss in many hydrostatic pumps and motors [1].

While this definition of the commutation loss was made only recently, there have been several attempts at improving the commutation behaviour of pumps and motors. These studies mainly focused on minimizing the output flow and pressure ripples, and noise emission, that result from the use of a stationary commutation unit, like a valve plate. Examples include the introduction of pre-compression volumes [2], the use of check-valves inside of the valve-plate [3], or placing the valve-plate at a cross-angle [4].

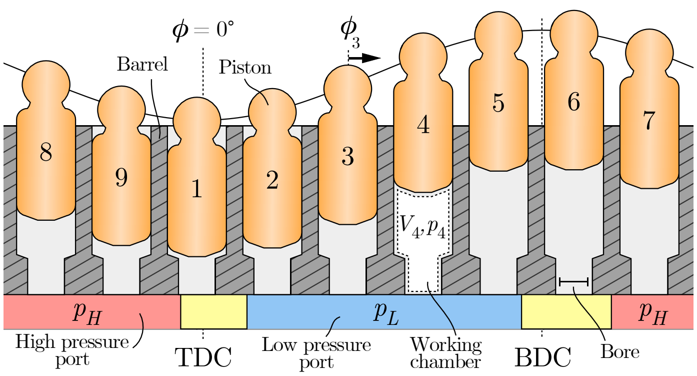
Apart from hydrostatic pumps and motors, the INNAS Hydraulic Transformer (IHT) [5] also uses a valve plate to ensure the commutation between its three pressure ports. To reduce noise and improve the efficiency of the IHT, 'shuttle' technology was introduced [6].

The current research focuses on the implementation of shuttle technology in a commonly used, slipper-type axial piston unit. This machine is simulated both as a pump and as a motor, with and without the use of shuttles.

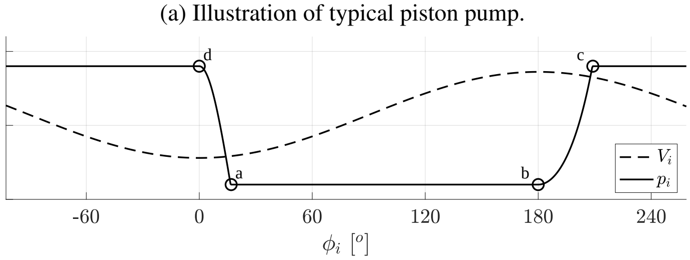
2 COMMUTATION

Figure 1a shows a two dimensional cross-sectional view of a typical piston pump. This particular pump has nine pistons, shown in orange, that reciprocate in a barrel, shown in gray, following the shown sine-wave. Each piston i has a corresponding working chamber in the barrel with volume V_i and pressure p_i, and an angular position ϕ_i. The valve plate, shown at the bottom of fig. 1a contains two openings: One to the low pressure port (or supply port, pressure p_L), shown in blue, and one to the high pressure port (or discharge port, pressure p_H), shown in red (the right side is of the figure is connected to the left side). The valving lands, shown in yellow, are located around the two outer positions of the pistons, top dead centre (TDC, ϕ=0°) and bottom dead centre (BDC, ϕ=180°). The bores in the bottom of the barrel (at the contact between valve plate and barrel) are the connection between the chambers in the barrel and the two pressure ports.

When the pump shaft rotates, the barrel and the pistons move to the right (as shown by ϕ_3), while the valve plate remains stationary. Since the valving lands are located around TDC and BDC, the chamber is filled with oil from the low pressure port when the piston is moving up. When the piston movement reverses, the oil from the chamber is pushed out, into the high pressure port.



(a) Illustration of typical piston pump.



(b) V_i and ideal p_i as a function of ϕ_i (exaggerated).

Fig. 1: Cross-sectional view of a simple pump with nine pistons. Bottom graph shows the volume and pressure of the working chambers in an ideal pump (i.e. without energy loss).

2.1 Ideal pump cycle

Figure 1b shows the ideal pressure level of the oil in a single working chamber. From point **a** to point **b**, the piston is moving away from the valve plate (up in the shown orientation). The chamber volume increases, while oil flows into the chamber from the low pressure port. At point **b**, the chamber volume is at its maximum, and the connection to the low pressure closes. Looking at fig. 1a, this means that the bore is closed by the BDC valving land. Starting at point **b**, the working chamber is not connected to either of pressure ports, and the piston is commuting from the low pressure port to the high pressure port. Since the piston continues to move, the chamber volume decreases, which increases the pressure of the oil due to compression. At a certain point in the rotation, the valving land ends, and the connection to the high pressure port is opened. Ideally, this happens when the pressure level of the oil in the working chamber is equal to that of the high pressure port, which is labelled point **c** in fig. 1b.

From point **c** to point **d**, the working chamber is connected to the high pressure port. The movement of the piston forces the oil out of the working chamber into the high pressure port. At point **d**, the chamber volume is minimal, and the connection is closed by the TDC valving land. Starting at point **d**, the piston movement increases the chamber volume, which decreases the oil pressure. Ideally, the valving land ends when the pressure level of the oil in the working chamber is equal to the pressure of the low pressure port, which is labelled point **a** in fig. 1b.

2.2 Design compromise

When designing a valve plate, the required size of the valving lands can be calculated using the ideal pressure line shown in fig. 1b. However, this required size is different when the pump is operated at a different discharge pressure. For example, suppose that pressure p_H were twice as high for the pump shown in fig. 1. When the same valve plate is used, the connection between the chamber and the high pressure port will open too soon. This results in oil suddenly flowing from the high pressure port into the chamber instead of the other way around.

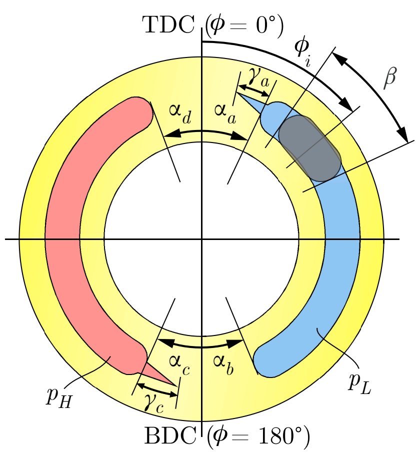
On the other hand, when pressure p_H is only half of the pressure shown in fig. 1b, the connection between the chamber and the high pressure port will open too late. In this case, the pressure in the working chamber will be much larger than the discharge pressure, resulting in higher throttling losses on opening. There are similar issues when the piston is commuting in the TDC.

The situation described above indicates that there exists an optimal valve plate for each possible pressure difference of the pump. However, since a pump needs to be able to be operated at all possible pressures, the design of a valve plate is always a compromise. This is an inherent problem for hydrostatic machines with valve plates or other types of stationary commutation components.

2.3 Simulation example

To illustrate the impact of such a compromised valve plate design, a simulation model of a commonly used slipper-type axial piston pump was created. The dimensions of the simulated pump are shown in the appendix. Figure 2 shows a sketch of a typical valve plate for pump operation. In this figure, we see the two pressure ports in blue and red. In the shown orientation, the barrel rotates over the valve plate in a clockwise direction. The angle ϕ_i indicates the center of the barrel bore (shown in gray, with angular width β). The angles of the edges of the pressure ports with respect to the TDC and BDC, are given by α_a to α_d. The length of the silencing grooves are given by γ_a and γ_c.

The angular values shown in fig. 2 are measured values from the actual pump. It is important to notice that while the low pressure port ends 22.0° before BDC (α_b), the bore is connected to the low pressure port until ϕ_i=173.1° (180°-α_b+$\beta/2$).



Angle	Value
α_a	25.3°
α_b	22.0°
α_c	24.2°
α_d	21.5°
β	30.2°
γ_a	19.5°
γ_c	19.9°

Fig. 2: Valve plate as found in the simulated slipper-type pump. Angles α are edges of the ports, β is width of the bore, γ are the length of the silencing grooves.

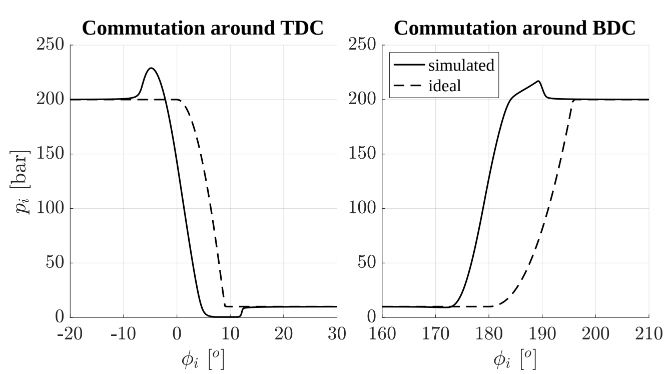


Fig. 3: Pressure in a working chamber in a simulation model of a commonly used slipper-type axial piston pump, with a stationary valve plate, at 1500 rpm.

Another thing to notice about the valve plate shown in fig. 2 is that the valving lands are shorter than the width of the bore, due to the silencing grooves (or pressure relief grooves). For example, in TDC, the valving land is 27.3° wide (α_d+α_a-γ_a). This means that there is a brief moment at which the working chamber will have a connection to both pressure ports. If the valve plate would not have any silencing grooves, the pressure in the working chamber would suddenly change at the moment the connection to the next pressure port is opened. This means the pressure transient (the rate at which the pressure changes) becomes very large. These high pressure transients are seen as one of the main sources of noise emission in hydrostatic pumps and motors [7]. Silencing grooves are commonly used for these kind of valve plates, as they smoothen the transition between the two ports resulting in lower noise levels and less pulsations in the output flow.

Figure 3 shows the simulated pressure in a single working chamber around the two commutation areas, at a speed of 1500 rpm. The left side of the figure shows the commutation around TDC. Following the solid line, we find that the connection to the high pressure port closes before the actual TDC is reached (at ϕ=0°). This results in an increasing chamber pressure, since the chamber volume is still decreasing at this point. There is a peak at roughly -5°, after which the chamber pressure decreases. This means that the connection to the low pressure port is already opened. In the case of the simulated pump, this connection is made via the silencing groove. Around 5°, the pressure in the chamber is decreased to 0 bar, which could indicate that this pump might have some cavitation issues. From 13° the pressure in the working chamber is on the same level as the supply pressure, which means that the commutation is complete.

The right side of fig. 3 shows the commutation around BDC. Following the solid line again, it can be found that the connection to the high pressure port opens before the BDC is reached (at ϕ=180°). Similar to the commutation in TDC, this connection is made by a silencing groove.

Whenever the commutation does not follow the ideal commutation, there is some power loss. The amount of power loss as a result of non-ideal commutation, the commutation loss, can be calculated using the simulation model [1]. For the results shown in fig. 3, the total commutation loss is calculated to be 306 W. This is roughly 2.5% of the total mechanical power required to operate the pump.

3 SHUTTLES

In the original paper, a solution to the inherit problem of valve plate pumps is presented in the form of so-called 'shuttle' technology [6]. These shuttles are small (spherical) pistons between two consecutive working chambers, which is shown in fig. 4. The shuttles, which are labelled A to I, are able to move between two end positions (the left and right seats), based on the difference in pressure between the two connected bores.

3.1 Working principle

The working principle of shuttles will be discussed using some example situations. Figure 5 shows the commutation of piston 1 around TDC. When the piston is in the TDC (fig. 5a), the pressure in chamber 1 is larger than the pressure in chamber 2. As a result, shuttle *A* is positioned on the right seat. Due to movement of piston 1, the chamber pressure decreases. At some point after TDC, the pressure in chamber 1 becomes lower than the pressure in the low pressure port, and thus the pressure in chamber 2. As a result, shuttle *A* will start moving towards its other seat (fig. 5b). Due to this movement, the chamber volume is kept the same, and the pressure in chamber 1 does not decrease further.

On the other side of the pump cycle, the commutation around BDC occurs in a similar way. Figure 6a shows piston 5 in the BDC position. The pressure in chamber 5 is lower than the pressure in chamber 6, so shuttle *E* is positioned on the left seat. The movement of piston 5 increases the chamber pressure, due to compression. When the pressure in chamber 5 becomes larger than the pressure in chamber 6, shuttle *E* will start moving towards its other seat (fig. 6b).

3.2 Valve plate design

As is described above, an important part of the working principle of the shuttles is that the connection to the pressure ports is closed long enough for the chamber pressure to compress. This means that a different valve plate design is needed. Figure 7 shows a valve plate designed for the use of shuttles. There are three main differences between the valve plates shown in figs. 2 and 7. Firstly, both ports end when the bore is exactly in the TDC and BDC position (α_b=α_d=$\beta/2$). Secondly, there are no pressure relief grooves, because the pressure peaks will be compensated for by the movement of the shuttles. Thirdly, the start of the two pressure ports is delayed significantly (α_a and α_c larger than before). The delayed port opening is needed to ensure that the connection to the pressure port is closed long enough for the pressure in the working chamber to reach the correct pressure level before opening.

3.3 Importance of shuttle position

In the situation shown in fig. 6, the succeeding shuttle (shuttle *D*) is at its left seat when piston 5 starts commuting in BDC. Suppose that piston 5 is commuting in BDC, while shuttle *D* is at its right seat. This situation is shown in fig. 8. When the piston starts to move down, shuttle *D* can freely move away from its initial position to account for the change in volume. This means that compression of the oil in chamber 5 does not start until shuttle *D* reaches its left seat. As a result, the connection to the high pressure port could open too soon when the same valve plate is used. Something similar happens when the succeeding shuttle is at its left seat at the start of TDC (instead of on its right seat, as shuttle *I* is shown to be in fig. 5a).

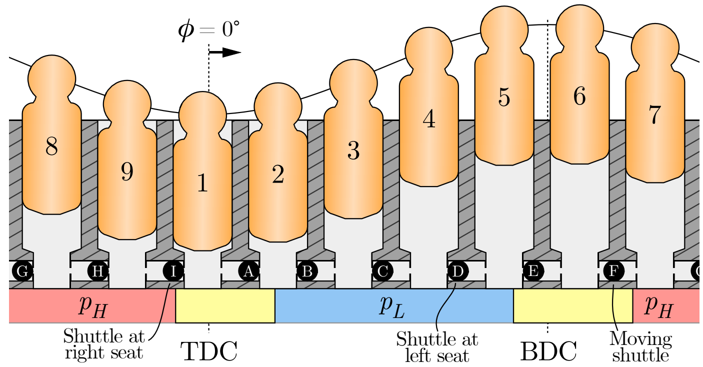
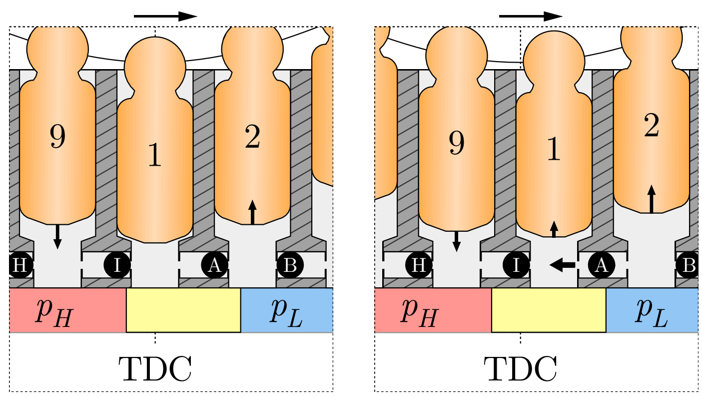


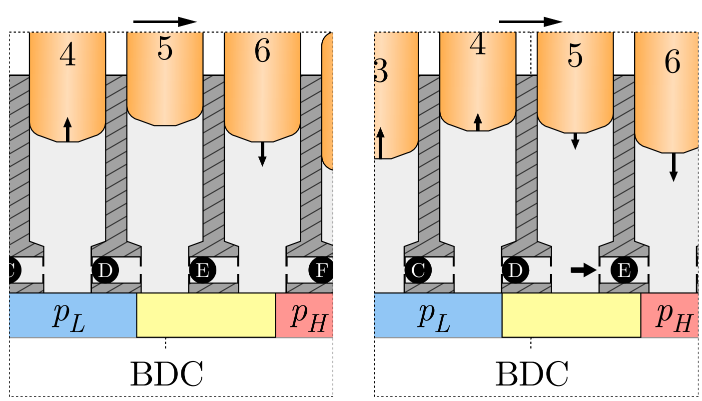
Fig. 4: Cross-sectional view of the implementation of shuttles (labelled *A*-*I*) in the piston pump.



(a) Piston 1 at TDC

(b) Piston 1 after TDC

Fig. 5: Chamber volume of piston 1 at TDC (a) and just after TDC (b). Shuttle *A* starts moving when $p_1 < p_2$.



(a) Piston 5 at BDC

(b) Piston 5 after BDC

Fig. 6: Chamber volume of piston 5 at BDC (a) and just after BDC (b). Shuttle *E* starts moving when $p_5 > p_6$.

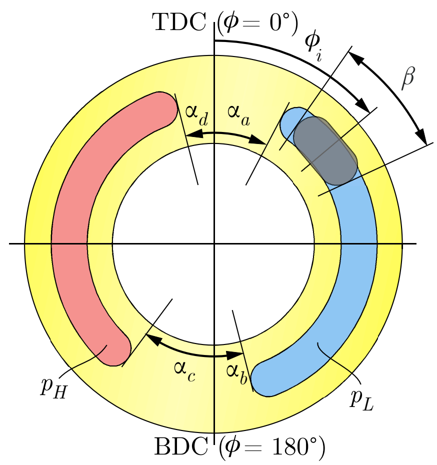
As shown above, the position of the shuttles at the start of the commutation has a large effect on the working principle. To account for the problem illustrated in fig. 8, the opening of the high pressure port could be delayed even further (increase α_c). This only works when the shuttle position at the start of each commutation is the same under each possible operating condition. However, the shuttles are free moving bodies, and there is no mechanism that forces them into a particular position. When two consecutive pistons are connected to the same pressure port, the chamber pressures will theoretically be equal. Without a pressure difference, the position of the shuttle between the two chambers is undefined; it can be either in its left seat, in its right seat, or anywhere in between.

4 SHUTTLE POSITIONING

To define the position of the shuttle at the moment commutation starts, an additional force should act on the shuttle. A small ceramic shuttle with a diameter of 5 mm weighs roughly 0.2 g, which means that a net force of 1 N results in an acceleration of 5.000 m/s^2. As a result, small forces can already result in a movement of the shuttle. An example of small forces acting on the shuttle are centrifugal forces. Unfortunately, introducing a centrifugal force will only define the position of the shuttles at high operating speeds, and only in one direction. From simulation models, it has been found that this results in different shuttle positions (and thus a different valve plate design) at low operating speeds.

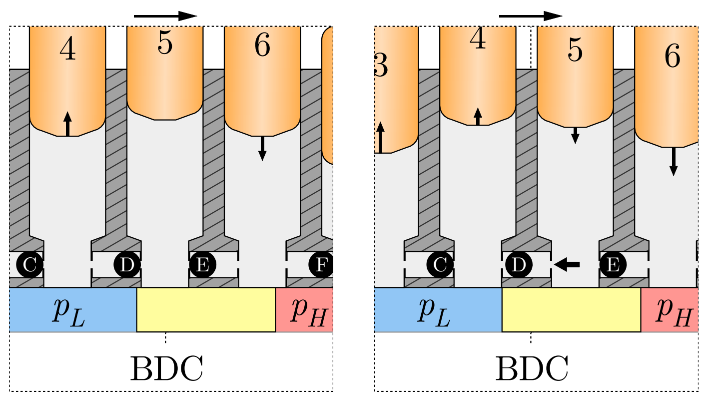
Since a small force will already be able to move the shuttle, a small pressure difference between both sides of the shuttle could also be able to define the position of the shuttles before commutation. Typically, the bores that connect the cylinders to the pressure ports are narrower than the cylinders themselves. This means that when a piston pushes oil out of the working chamber, there is some throttling with a pressure difference Δp. The pressure in the cylinder is thus slightly higher than the pressure at the end of the bore. When oil is flowing from the low pressure port into the cylinder, the throttling works the other way around. The pressure in the cylinder is now slightly lower than the pressure at the end of the bore.

Figure 9 shows that the shuttles can be positioned in a way that uses this throttling effect to define the position of the shuttles. The preceding shuttle (so shuttle B for piston 2) is connected to the end of the bore that is closer to the piston. The succeeding shuttle (so shuttle A for piston 2) is connected to the end of the bore that is near the valve plate. The Δp over the bores ensures that the shuttles reach the correct seat before commutation starts.



Angle	Value
α_a	28.1°
α_b	15.1°
α_c	37.1°
α_d	15.1°
β	30.2°

Fig. 7: Valve plate designed for the pump with shuttles.



(a) Shuttle D at right seat in BDC (b) Shuttle D moving

Fig. 8: Chamber volume of piston 5 at BDC (a) and just after BDC (b), with shuttle D starting at its right seat. Movement of piston 5 results in movement of shuttle D instead of compression.

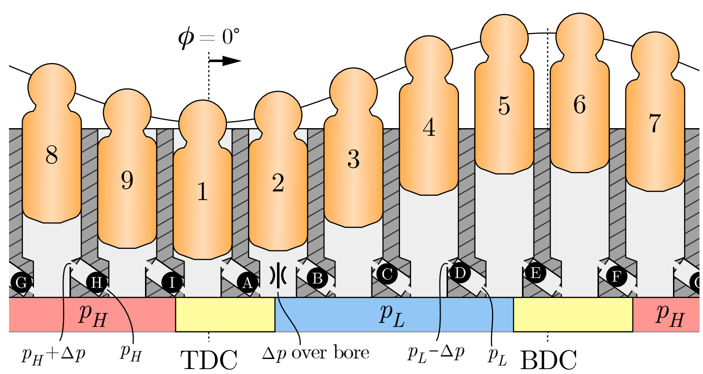


Fig. 9: Proposed solution to define the shuttle positions. Pressure difference over the bores ensures that the shuttles are in the correct position when commutation starts.

5 SIMULATION MODEL

5.1 Model description

The proposed shuttle orientation was implemented in the simulation model that was used for the standard pump. Figure 10 shows a comparison between the model of a standard pump, and a pump with shuttles. In a typical model of a pump, as shown in fig. 10a, each working chamber is modelled as a separate volume V_i. The size of this volume changes with the piston position x_i, which is a function of the pistons angular position ϕ_i. The connection between the chamber and the two pressure ports is modelled as orifices with a flow area that is also a function of ϕ_i.

To implement the shuttles as proposed in fig. 9, the volume V_i is split into four volumes, as shown in fig. 10b. These four volumes are: the cylinder volume, $V_{i,c}$, the bore volume $V_{i,b}$, the preceding shuttle volume, $V_{i,ps}$, and the succeeding shuttle volume, $V_{i,ss}$. Furthermore, the figure shows a shuttle position, s_i, which is the position of the preceding shuttle. Since the shuttles introduce a form of communication between two consecutive working chambers, the chambers of this simulation model are no longer separated from one another.

5.2 Simulation results

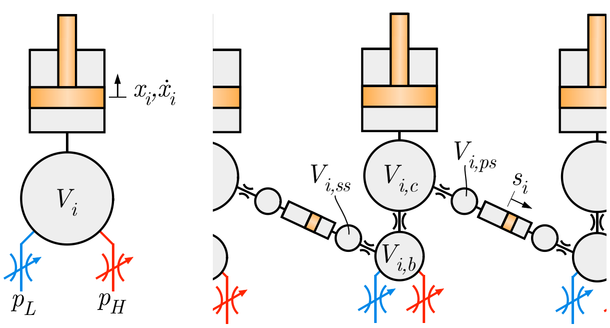
The result of the simulation with shuttles at 1500 rpm is shown in fig. 11. Figure 11a shows the chamber pressure and the position of the preceding shuttle (so shuttle B for piston 2) during a full pump cycle of one piston. For illustrative reasons, the values of the two lines are scaled. The figure shows that the shuttle is in the correct seat at the start of the commutation. The shuttle starts moving exactly when the chamber pressure equals the port pressure ($\phi_i = 10°$ and $\phi_i = 195°$). Between the two commutation areas, the shuttle moves towards the other seat.

Figure 11b shows the resulting chamber pressure in more detail. The ideal chamber pressure is also shown in this figure. Unlike the results shown in fig. 3, the chamber pressure closely follows the ideal pressure line. The moment at which the port pressure is reached, the chamber pressure shows some fluctuations. These fluctuations indicate rapid changes in the volume of the shuttle chamber, caused by the moving shuttles.

The commutation loss of the simulated shuttle pump, is calculated to be reduced to 3.7 W, roughly 0.03% of the required mechanical power for this unit. This means that the commutation loss has become negligible compared to other losses. Furthermore, the smooth pressure transient of the results shown in fig. 11 could indicate that this pump has low noise levels. Experimental measurements will be needed to confirm this.

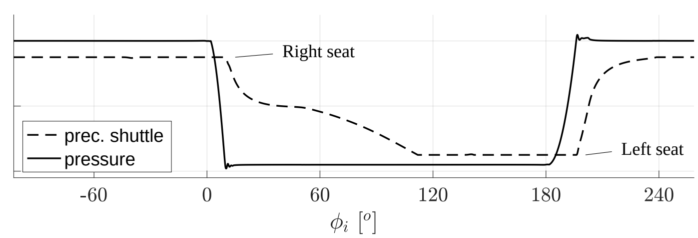
The reason why the commutation loss is so small, is that the shuttles effectively create a variable valve plate. When the working chamber is commuting in BDC, oil compresses up to the pressure level of the high pressure port. While there is no connection to the high pressure port at this point, the excess oil results in a movement of the preceding shuttle. Since the other side of this shuttle is connected to a chamber that is connected to the high pressure port, the excess oil is pushed out at the preceding working chamber.

Figure 12 shows the described flow rate from a single working chamber to the high pressure port. The ideal flow rate can be calculated using the movement of the piston, and the moment at which the pressure level in the chamber equals the pressure of

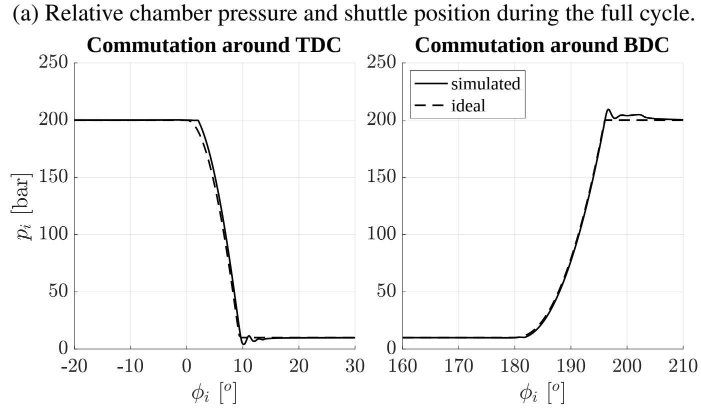


(a) Standard (b) With shuttles

Fig. 10: Simulation model of a standard pump, and of a pump with shuttles. Circles are volumes with a uniform pressure, orange pistons add variable volume to the connected circle.



(a) Relative chamber pressure and shuttle position during the full cycle.



(b) Chamber pressure around commutation areas.

Fig. 11: Simulation results of the pump with shuttles, at 1500 rpm.

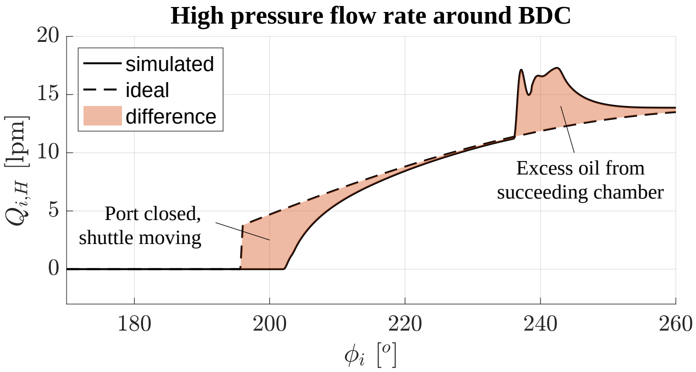
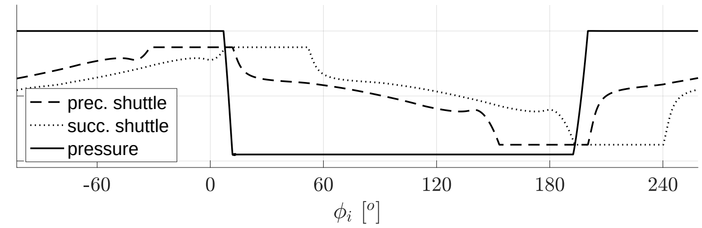
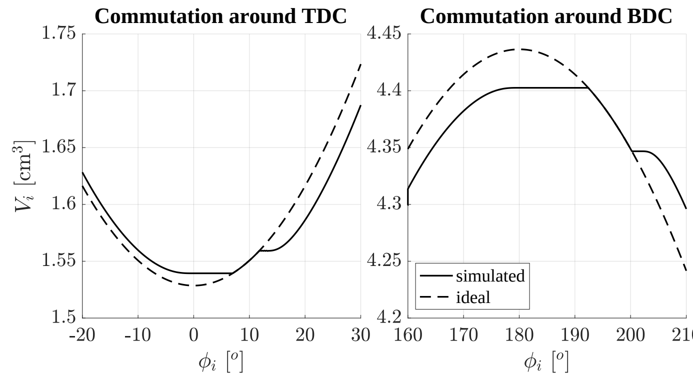


Fig. 12: Simulated flow rate from a working chamber to the high pressure port, for the shuttle pump.



(a) Relative chamber pressure and shuttle positions.



(b) Chamber volume around commutation areas.

Fig. 13: Simulation results of the pump with shuttles, at 100 rpm.

the high pressure port. The figure shows that the flow rate of the simulated pump actually starts too late (ϕ_i=202° instead of ϕ_i=195° for the ideal line), and the flow rate is less than the ideal flow rate. However, at a later point, the flow rate is higher than the ideal flow rate, since the succeeding chamber is now commuting. The two areas that indicate the difference between the ideal and simulated flow rate are of equal size, which means that the total output volume is equal to the ideal output volume.

6 LOW SPEED BEHAVIOUR

The simulation results have shown that the pressure difference over the bore can be used as an effective solution to define the position of the shuttles. This pressure difference is a function of the speed at which the piston moves. When the rotational pump speed is very large, the piston moves fast, resulting in a relatively large pressure difference. However, at low rotational speeds, the piston will move much slower, which leads to a smaller pressure difference.

Figure 13 shows simulation results at 100 rpm. At this speed, there is not enough of a pressure difference to completely move the succeeding shuttle into the correct seat, before the commutation starts. Looking at the BDC shown in fig. 13a, we find that after ϕ_i=180°, the succeeding shuttle is first pushed into the left seat. When the seat is reached, compression starts (ϕ_i=192°). The preceding shuttle then starts moving when the chamber pressure reaches the pressure of the high pressure port.

Figure 13b shows the total volume in the chamber during commutation ($V_{i,c} + V_{i,b} + V_{i,ss} + V_{i,ps}$). The volume does not follow the same line as the ideal pump cycle. When the connection to the high pressure port closes (at ϕ_i=0°), the chamber volume is slightly higher than in the ideal cycle. On the other hand, when the connection to the low pressure port closes (at ϕ_i=180°), the chamber volume is slightly lower than in the ideal cycle.

The difference in chamber volume at the start of commutation has an effect on the total output flow of the pump (decreases by roughly 1.5% in this case). However, the process of compressing and expanding the oil in the working chamber still happens almost without energy loss. In other words, at these low operating speeds, the commutation loss is still negligible, but the effective displacement of the pump slightly decreases.

7 MOTORING

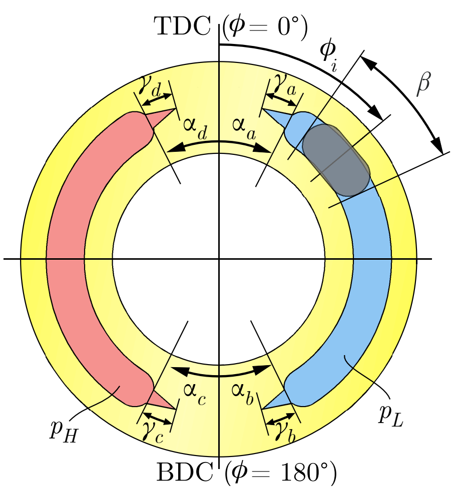
Up until now, the focus has been on a hydrostatic pump. When motoring, the barrel and the pistons rotate in the opposite direction. This means that oil flows from the high pressure port into the working chambers during the suction stroke, and is pushed out of the chamber into the low pressure port during the discharge stroke.

7.1 Standard slipper-type motor

While hydrostatic pumps can often be operated as a motor as well, the valve plates are designed specifically for pump operation. For the commonly used slipper-type pump that was simulated before, a nearly identical machine exists for motor operation, which has a different valve plate geometry. The motor valve plate is shown in fig. 14. The valve plate of the pump (fig. 2) had silencing grooves on the inlets of the two pressure ports. The valve plate of the motor has silencing grooves on both the inlets and the outlets of the two pressure ports, and is symmetrical about the TDC-BDC axis. This kind of symmetric valve plate is commonly used for hydrostatic motors.

Figure 15 shows the simulation results of the standard slipper-type motor. At this operating pressure, the connection to the low pressure port should ideally close at roughly ϕ_i=9°. Since the piston is not yet in TDC, the movement from 9° to 0° will compress the oil in the chamber to reach the pressure level of the high pressure port. Due to the geometry of the valve plate of this motor, the working chamber stays connected to the low pressure port for a very long time. In this case, there is barely any compression, and the increase in pressure is mainly caused by the opening of the high pressure port. The commutation in BDC shows a similar pressure profile.

The commutation loss of the results shown in fig. 15 is calculated to be roughly 205 W, which is a loss of about 1.7% of the theoretical power.



Angle	Value
α_a	26.2°
α_b	26.2°
α_c	26.2°
α_d	26.2°
β	30.2°
γ_a	10.5°
γ_b	10.5°
γ_c	10.5°
γ_d	10.5°

Fig. 14: Symmetric valve plate found in the slipper-type motor.

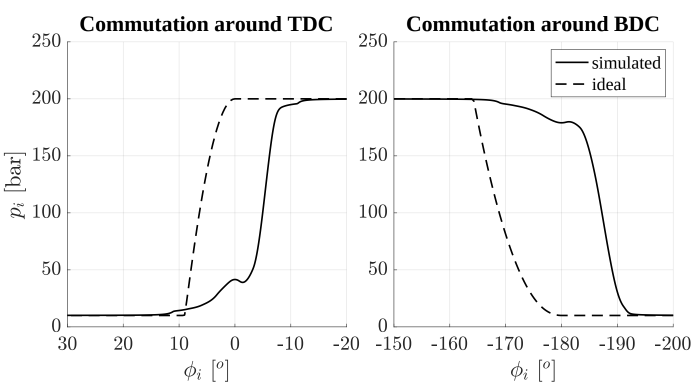


Fig. 15: Pressure of a working chamber in a simulation model of a commonly used slipper-type axial piston motor, at 1500 rpm.

7.2 Motor with shuttles

When the slipper-type machine with shuttles is operated as a motor, the flow through the bores changes direction. This means that the pressure difference caused by throttling works the other way around, as shown in fig. 16. Due to this pressure difference during motoring, the shuttles are pushed into the opposite seats as they would during pump operation.

As was seen in fig. 15, the commutation for ideal motor operation does not start at, but before TDC and BDC. When using the same valve plate as used for the pump with shuttles (fig. 7), this means that the commutation will start too early in motor operation.

Figure 17 shows simulation results of the machine with shuttles, operated as a motor. The chamber pressure and shuttle position of the preceding shuttle are shown in fig. 17a. Note that the succeeding shuttle during pump operation, is the preceding shuttle during motor operation (so shuttle *A* for piston 2).

For the shown results, the compression in TDC starts at ϕ_i=13°, since this is when the connection to the low pressure port closes for this valve plate. Figure 17b shows that the chamber volume follows the ideal line at this angle. At 10°, we find that the compression is completed, and the chamber pressure equals the pressure of the high pressure port. At this angle, the shuttle starts to move away from its right seat. The result is that the chamber volume in fig. 17b stays constant after this angle. From ϕ_i=0°, the connection to the low pressure port opens, and the volume follows a sinusoidal pattern again at an offset with respect to the ideal line.

Similar to the results for the low rotational speed simulation shown in fig. 13, this machine has a larger chamber volume in TDC and a smaller chamber volume in BDC. In other words, the displacement volume of this motor will be smaller than it would be ideally. However, also similar to the low speed behaviour, the compression and expansion of the oil in the working chamber occurs with little to no power loss.

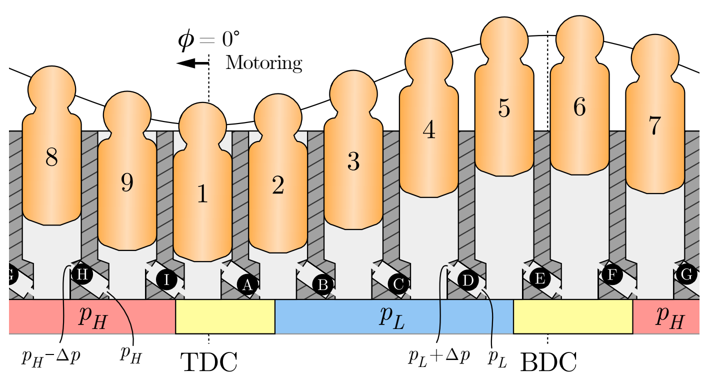


Fig. 16: Machine with shuttles, operated as a motor. Pressure difference over the bores is reversed.

8 CONCLUSION

Many hydrostatic pumps and motors suffer from the limitation of a stationary valve plate design. The fact that this component should work in a broad range of operating conditions inherently leads to compromises that result in power loss during the commutation periods of the working chambers.

To solve this issue, a novel application of so-called 'shuttle' technology in a commonly used slipper-type pump and motor is investigated using a simulation model. The results indicate that the implementation of shuttles has the potential to reduce the commutation loss of these machines to negligible levels. Furthermore, the results suggest that the use of shuttles could also reduce noise emissions. An experimental study should confirm these theoretical findings, and will be the topic of future research.

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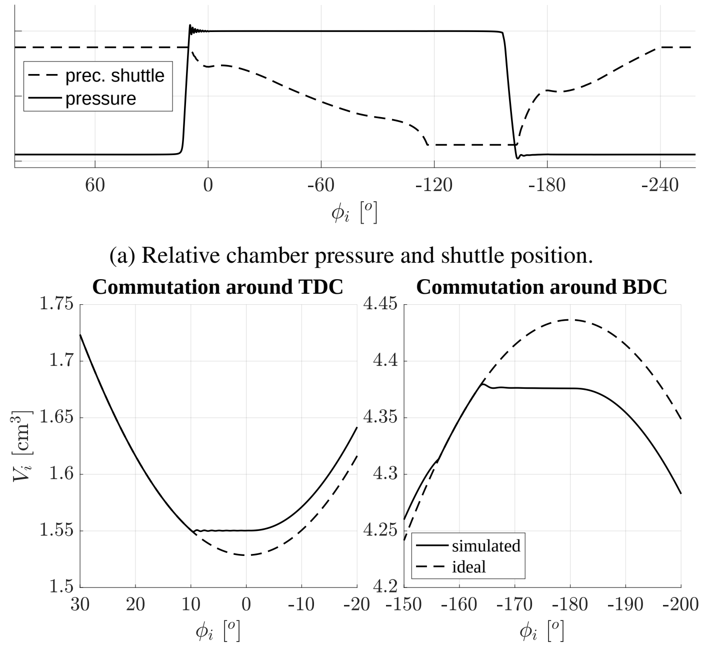

(b) Chamber volume around commutation areas.

Fig. 17: Simulation results of the machine with shuttles, operated as a motor at 1500 rpm.

APPENDIX

The following parameters are used in the simulation models. They are based on measurements of a commonly used, constant displacement, slipper-type unit.

Pump parameters		
Piston diameter	13.5	mm
Pitch radius	29.5	mm
Swash angle	19.0	°
Number of pistons	9	-
Volume per piston:		
Minimum V_i	1.40	cm^3
Minimum $V_{i,c}$	1.13	cm^3
Constant $V_{i,b}$	0.27	cm^3
Minimum $V_{i,ss}$ and $V_{i,ps}$	0.01	cm^3
Flow area bore	60.60	mm^2

Shuttle parameters		
Diameter	4.8	mm
Stroke (left to right seat)	6.0	mm
Mass density (ceramic)	3200	kg m^{-3}
Flow area to shuttle chambers	5.41	mm^2