15th IFK
Proc. of the 15th International Fluid Power Conference < Syı

# Review of Single-Sided and Double-Sided Axial Piston Machine Characteristics

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## Abstract.

Most conventional axial piston machines have a single rotating group. However, double-sided machines with another set of pistons can have the same displacement with different properties. Double-sided machines can counteract forces and moments between each side and therefore reduce structure-borne noise and bearing loads. Further, there are different setups of double-sided machines, with a main difference being if there is a phase shift between the sides. This paper qualitatively compares the properties of different machine setups.

**Keywords.** Axial piston pump, fluid power, pump design, double-sided machine.

## 1. Introduction

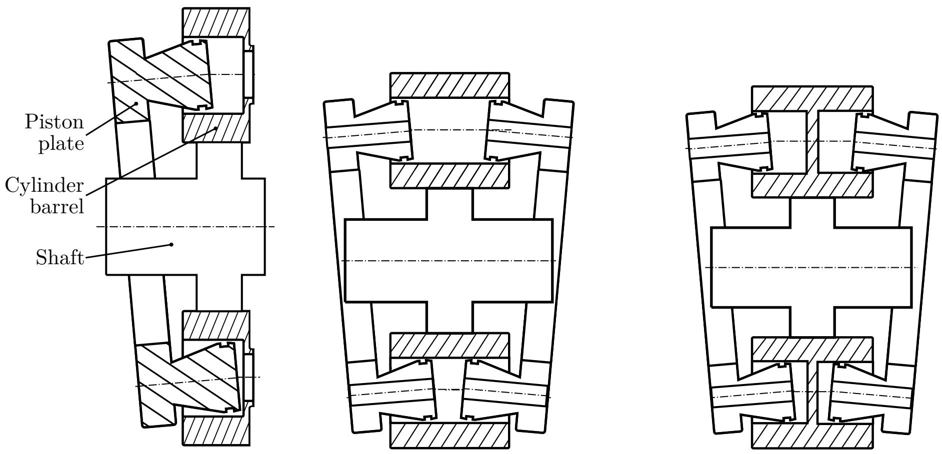
There are various types of positive displacement machines. One machine type which is very popular in mobile machinery is the axial piston machine type. For this machine type, there are different options available on the market. One significant difference can be how the rotating group is set up. Many machines on the market have one set of pistons (e.g., [1, 2]), which will be referred to as single-sided machines in this paper. However, several sets of rotating pistons can be driven on the same shaft and are available on the market, e.g., double row machines with two sets of pistons on different diameters (e.g., [3]), machines using modular back to back designs (e.g., [4]) where each set of pistons has its own housing, or tightly integrated machines where a mirrored design of two sets of pistons shares the same housing and the same inlet and outlet [5].

Some advantages of using two sets of pistons have been presented in the literature. For example, Innas BV presented their floating cup machine with two sets of pistons (e.g., [6]), and mentions the benefit of increasing the number of pistons, and the compensation of axial forces due to the mirrored design. This results in reduced bearing loads and is expected to reduce structure-borne noise. The high number of pistons also reduces both kinematic and compressible flow pulsations and thus reduces fluid-borne noise. Similarly, the Vickers MPEV3-040-2 aerospace motorpump included two inline machines driven in opposite directions by individual electric motors, which cancels both radial and axial piston forces. This results in reduced yoke and bearing loads. Further, the two inline machines shared one displacement controller [7]. Mehta [8] used a tandem pump with an angular between the mirrored sets of pistons to reduce the torque ripple by 75%. Danes and Vacca [9] aimed for destructive interference of the flow of a tandem pump in order to reduce fluid-borne noise. For low harmonics, measurements showed significant reductions of sound pressure levels, whereas, contrary to the expectation from simulation, this was not the case for higher harmonics. Ericson and Heeger discussed using mirrored pump designs to achieve discretely variable by idling one pump half [10] or continuously variable displacement by rotating the valve plates [11].

## 1.1. *Machine Configurations*

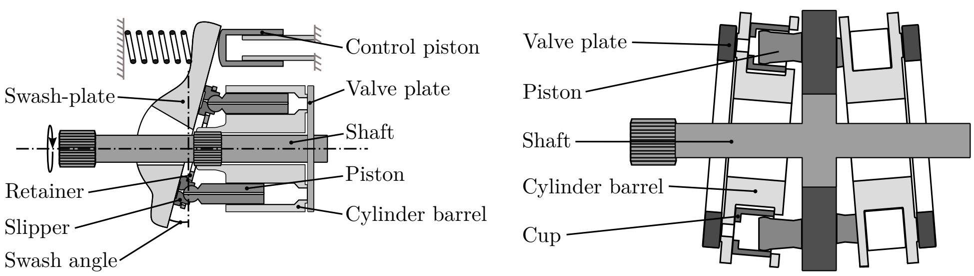
Figure 1.1 sketches different axial piston machine configurations on the example of floating piston machines. Figure 1.1a shows a single-sided set up, whereas Figures 1.1b and 1.1c show double-sided machines with mirrored rotating groups. In Figure 1.1b, pistons from each side share a displacement chamber, whereas in Figure 1.1c, the cylinder barrel provides a separate displacement chamber for each piston. Note that further configurations are feasible, e.g., by using multiple cylinder barrels, but are not in scope of this paper.

Figure 1.1 shows floating piston machines, but the same machine configurations are applicable also for floating cup machines or conventional slipper type machines as sketched in Figure 1.2. As bent-axis machines do not provide a through-drive possibility [12], double-sided machines are not applicable for them.




(a) Single-sided machine. (b) Double-sided machine with connected sides. (c) Double-sided machine with separated sides.

Figure 1.1: Schematic of different floating piston machine versions [10, 13].




(a) Sketch of main components of a slipper-type machine. (b) Sketch of main components of a floating cup machine of mirrored design. Note the angular offset between the machine halves.

Figure 1.2: Schematic of axial piston machines [13].

### 1.2. *Research Gap*

Each of the aforementioned references was focusing on some specific design aspects. This paper aims to provide a wider comparison of pumps with one set of pistons with the two types of mirrored pump designs. The discussed aspects include flow, force, moment, and torque ripples; bearing loads; barrel tipping; aspect ratios; piston count.

### 1.3. *Limitations*

The simulations in Section 9 assume a inlet pressure of 3 bar and therefore cannot indicate differences in self-suction speed of the different machines. The pressure relief grooves in the study were manually tuned and therefore a comparison of commutation behaviour is not fully objective.

### 1.4. *Paper Outline*

Section 2 summarises background on flows, forces, and moments in axial piston machines. Based on this, Sections 3 to 8 qualitatively discuss the properties of the machine setups shown in Figure 1.1. Section 9 presents a lumped-parameter simulation study for specific designs. Finally, Section 10 summarises the properties for the different setups and shows conclusions.

## 2. Background on Flows, Forces, and Moments in Axial Piston Machines

The flow, force, and moment contributors in axial piston machines are crucial to understand their noise behaviour and bearing loads. Manring [14] and Ivantysyn and Ivantysynova [15] provided detailed information on this topic and Heeger et al. [16] summarised and visualised the noise contributors. The following section repeats this background information. Figure 2.1 sketches an exemplary sketch of nine pistons and valve plate kidneys.

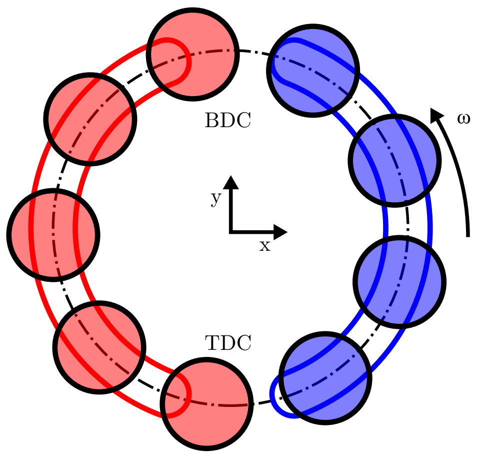



Figure 2.1: Sketch of nine pistons and valve plate kidneys. The sketch includes the location of bottom dead centre (BDC), top dead centre (TDC), as well as the coordinate system and direction of rotation.

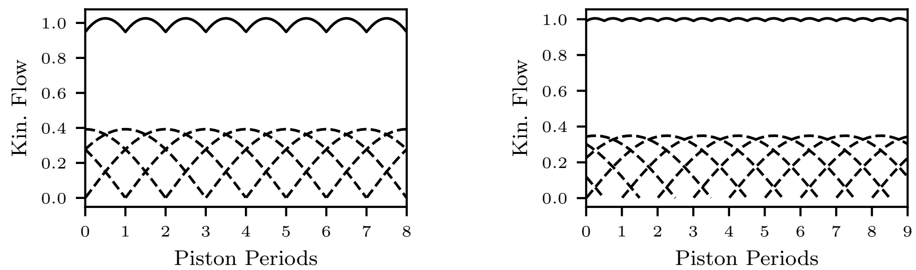
In stationary operation, the same events are in principle repeated for each piston. In a simulation environment, neglecting e.g., manufacturing tolerances, the events are even exactly repeated for each piston and therefore piston periods are used to visualise the machine's behaviour. A piston period represents the time it takes between two pistons passing the same location, and the figures show the results for two consecutive piston periods. Note that the time of one piston period depends both on the machine's speed and piston count.

## 2.1. *Flow Ripples*

The overall flow ripple is the sum of the kinematic and the compressible flow ripples.

### 2.1.1. *Kinematic Flow Ripples*

In axial piston machines, the piston velocity and therefore the flow of one displacement chamber follow a sine shape. As the number of displacement chambers is finite, the total delivered flow is not constant and kinematic flow ripples as shown in Figure 2.2 occur.




(a) Machine with 8 pistons.

(b) Machine with 9 pistons.

Figure 2.2: Exemplary kinematic flow ripples. Dashed: contributions of the individual pistons, solid: total kinematic flow. The figures are normalised with the average total kinematic flow.

The kinematic flow ripple behaves differently for machines with odd and even piston counts $z$. For even piston counts, the chambers connect and disconnect from the ports in pairs, whereas for odd piston counts, there is half a piston period phase shift between the events of connecting and disconnecting. Consequently, odd piston counts yield lower peak-to-peak kinematic flow ripples but higher frequencies [15]. Conventionally, 7 or 9 pistons are used [14].

### 2.1.2. *Compressible Flow Ripples*

Pressure equalisation (and the fluid's compressibility) cause compressible flow during commutation. In mobile working machines, the compressible flow ripples are usually larger than the kinematic ones [17]. Without any commutation features, using so-called zero-lapped valve plates, very large compressible flow ripples as shown in Figure 2.3 occur. Various commutation features can reduce the compressible flow ripple amplitude, and reviews are provided e.g., by Johansson [17] and Heeger et al. [18]. The frequency of the compressible flow is not affected by whether the piston count is odd or even.

Increasing piston counts can decrease the compressible flow ripple, as the stroke and dead volume of each displacement chamber are reduced, thus reducing the compressible flow required for pressure equalisation.

## 2.2. *Force, Moment and Torque Ripples*

### 2.2.1. *Axial Force Ripples*

The axial force of a piston depends on its pressure level. A varying count of pressurised pistons therefore causes axial force ripples. When neglecting commutation effects, even piston counts do not have an axial force ripple, as the chambers connect and disconnect in pairs. Odd piston counts however do have force ripples in the magnitude of the force of one piston. Figure 2.4

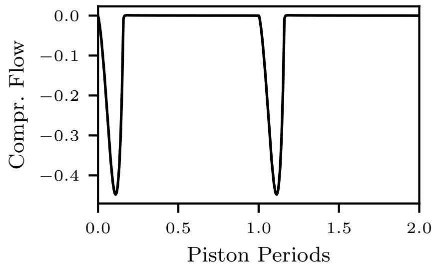
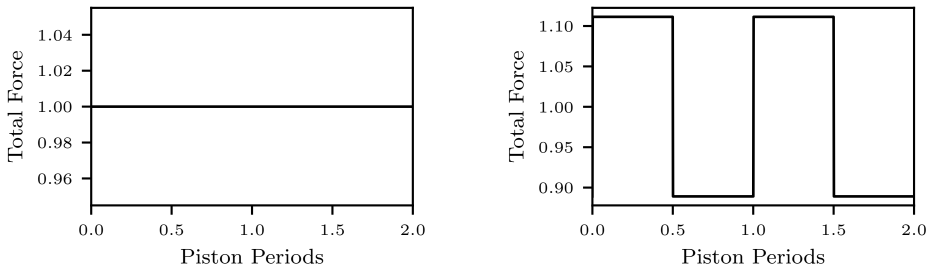



Figure 2.3: Compressible flow ripple for a zero-lapped valve plate, normalised with the average total kinematic flow.

shows the piston force ripple, neglecting commutation effects. Johansson [17] considers the axial force as the main source of structure-borne noise.



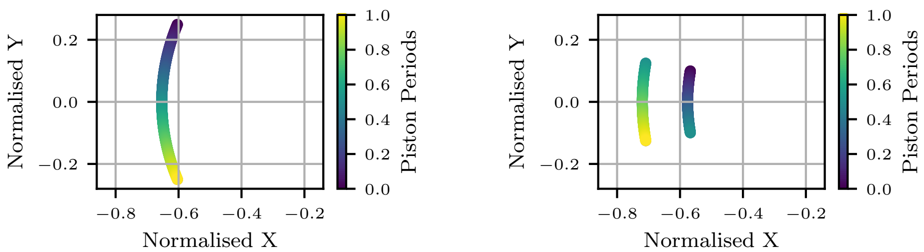

(a) Machine with 8 pistons.

(b) Machine with 9 pistons.

Figure 2.4: Exemplary normalised piston force ripples.

### 2.2.2. *Bending Moment and Drive Torque Ripples*

A varying count of pressurised pistons and the varying piston positions cause bending moment ripples, which also contribute to structure-borne noise. Figure 2.5 shows the position of the force resultants for one piston period 2.5.

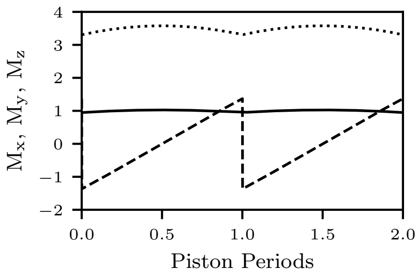



(a) Machine with 8 pistons.

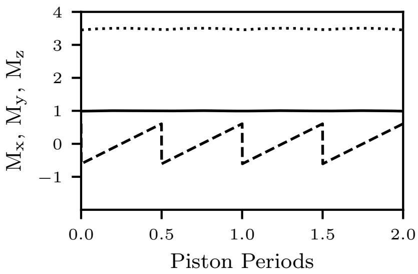
(b) Machine with 9 pistons.

Figure 2.5: Exemplary location of resultant of piston forces, neglecting commutation effect. Normalised with the radius of cylinder barrel bore location.

Bending moments occur around the x- and the y-axis (refer to Figure 2.5 for the axis definition) and cause loads on the bearings. The moment around the z-axis is the drive torque. Figure 2.6 shows bending moment ripples and drive torque ripples.




(a) Machine with 8 pistons.




(b) Machine with 9 pistons.

Figure 2.6: Exemplary bending moment ripples ($M_x$ - dashed, $M_y$ - dotted) and drive torque ripples ($M_z$ - solid line), normalised in relation to the average drive torque.

The bending moment ripple is increased for even piston counts, as the location of the resultant piston forces varies more. The ripple around the x-axis is larger than around the y-axis, as the normalised x-position is larger than the normalised y-position when the pistons enter / leave a port. Further, the x-moment has a varying sign and can therefore release geometric plays.

## 3. Bearing Loads and Barrel Tipping

In single-sided machines, the axial piston forces load a bearing. For double-sided machines, the axial forces from each side counteract each other. In case of the double-sided machine with connected sides, the forces and moments of each side are identical, leading to low bearing loads. For double-sided machines with a phase shift between the sides, the axial piston force is of alternating sign and of the magnitude of the force of one piston. Figure 3.1 visualises the resulting forces and moments for the three considered machine setups and a machine with 11 pistons. The origin for the moment calculation is in the intersection of the shaft axis and the mirroring plane to create the mirrored pumps. Moments are normalised by the average drive torque required to drive a single-sided pump [10].

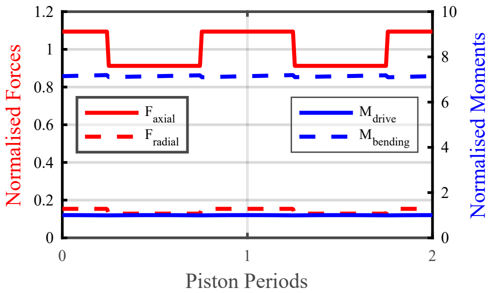
Mirroring a rotating group obviously doubles the displacement, the radial forces, and the drive torque, but reduces axial forces and bending moments. Note that for comparing single-sided and double-sided machines of the same displacement, the dimensions can be scaled according to the scaling laws presented by Achten [19].

### 3.1. *Bearing Loads*

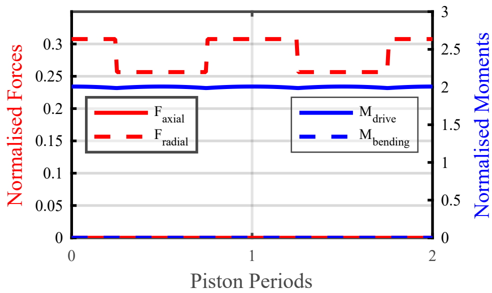
In case of a single-sided machine, the bearings are loaded with axial and radial forces and bending moments as shown in Figure 3.1a. As Figure 3.1b shows, the axial forces and bending moments from each machine side counteract each other in double-sided machines with connected sides. Therefore, the bearing are in principle only loaded with radial forces. If a double-sided machine with an angular offset between the two sides is used, the bearings are loaded with alternating forces with the magnitude equivalent to one pressurised piston as shown in Figure 3.1c.

### 3.2. *Barrel Tipping*

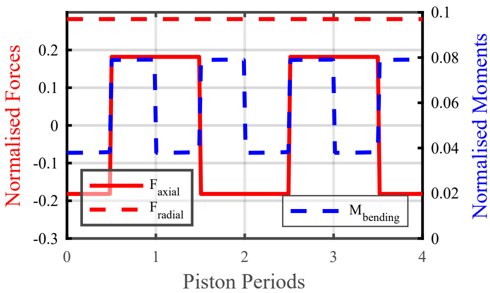
Achten et al. [20] investigate barrel tipping in single-sided floating cup and slipper-type machines, and show that hydrostatic, centrifugal, friction, spring, and reaction forces affect the cylinder barrel and therefore its tipping moment. In double-sided machines without an angular offset between the machine sides, the tipping moments from both machine sides counteract




(a) Normalised forces and moments for a single-sided machine.




(b) Normalised forces and moments for a mirrored machine with connected machine sides.




(c) Normalised forces and moments for a mirrored machine with independent machine sides and an offset of half a period between both sides. Due to the offset between the machine sides, there are twice as many piston periods per revolution.

Figure 3.1: Forces and bending loads on the bearings for different machine configurations. The double-sided machine in this figure have twice the displacement of the single-sided machine [10].

each other, thus providing higher robustness against barrel tipping. If there is an angular offset between the machine sides, the moments from each side are not fully compensated, but the resulting oscillating moments on the barrel are of lower magnitude than in single-sided machines.

# 4. Flow Ripple

## 4.1. *Interference of Flow from Double-sided Machines*

Danes and Vacca [9] aimed for destructive interference of the flow of a tandem pump in order to reduce fluid-borne noise. The considered tandem pump consisted of two pumps with nine pistons each. Simulation results showed that a phase shift of half a period eliminates the odd pressure ripple harmonics, whereas a phase shift of a quarter period eliminates harmonics of the order $2 + 4n$. Further, the pressure ripples for the tandem machine were lower than for a simulated single-sided machine of the same displacement, independent of the phase shift applied to the tandem machine halves. For the first and third harmonic, measurements showed significant reductions of sound pressure levels when using a phase shift of half a period between the pump halves, whereas, contrary to the expectation from simulation, this was not the case for higher harmonics. Danes and Vacca give insufficient symmetry in their experiment as a potential cause for this.

### 4.2. *Dead Volume*

Larger dead volumes cause increased compressible flow pulsations. Double-sided machines that require flow through the pistons (see, e.g., Figure 1.1) possess increased dead volume ratios.

## 5. Torque Ripple

Mehta [8] considered the torque ripple to be the main source of noise in hydraulic machines. He presented a slipper type machine with a central manifold and mirrored rotating groups on each side of the manifold. Mehta used this design to optimise the angular offset between the rotating groups to minimise the torque ripple. For even piston counts on each side, the ideal offset is half a period, whereas for odd piston counts, the ideal offset is a quarter of a period. When neglecting commutation and compressibility effects, the best offset angle reduces the peak-to-peak torque pulsations by 75% in comparison to the worst offset angle.

## 6. Aspect Ratios

Heeger et al. [21] expanded Achten's [19] method for the scaling of rotating groups of axial piston machines for floating piston machines. Figure 6.1 shows the rotating group's diameter, axial length, and volume for single-sided and double-sided floating piston machines of the same displacement. Obviously, the double-sided machine has smaller outer diameters and larger axial lengths. Also, its volume of the rotating parts is slightly lower and the total piston count is larger for the most compact machine.

In the context of the integration with electric machines, Heeger et al. [22] stated that single-sided machines are more advantageous for a compact integration via axial stacking, whereas double-sided machines are more advantageous for a compact radial integration within the core of an electric machine.

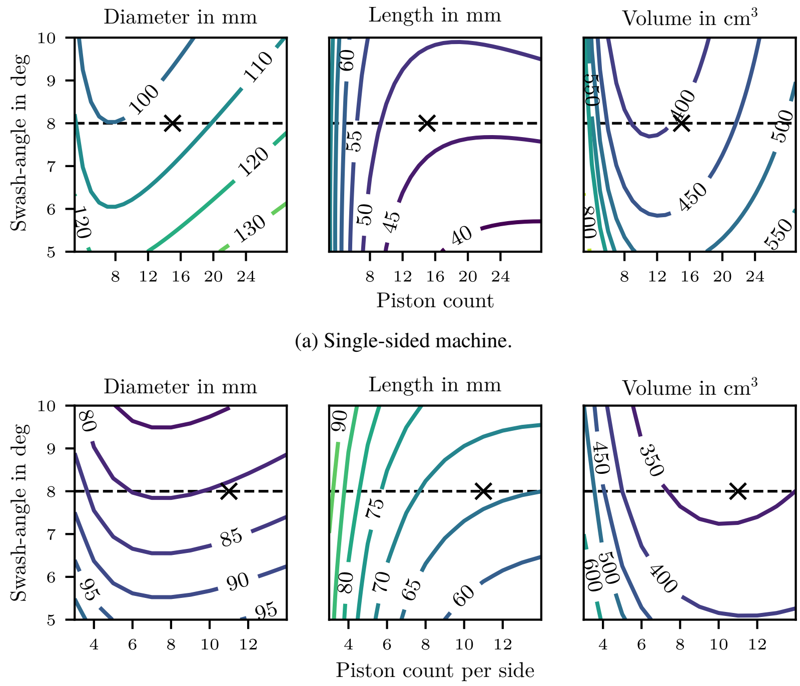
## 7. Variable Displacement

Variable single-sided machines vary their displacement by varying their swash-angle. Using double-sided machines opens opportunities for unconventional methods to vary displacement.

Valve plate rotation has been considered to alter the effective pre-compression for small rotation angles [23], and to provide variable displacement for large valve plate rotation angles [24]. Variable displacement via valve plate rotation however requires commutation to take place away from the dead centres and at significant axial piston speeds. This results in challenges such as cavitation, high-pressure peaks, and pulsations, and has therefore hindered a widespread application of valve plate rotation [25]. Note however that the Innas Hydraulic Transformer relies on valve plate rotation to alter the flow through each of its three ports [25].

In double-sided machines with connected sides as shown in Figure 1.1b each valve plate affects the effective pre- and de-compression angles. Therefore, the displacement and commutation behaviour can be altered at the same time. However, the high piston velocities at low setting ratios provide a significant throttling effect, and Heeger et al. [11] concluded that this method is mainly applicable to vary displacement at relatively large setting ratios but unsolved challenges remain for low setting ratios.

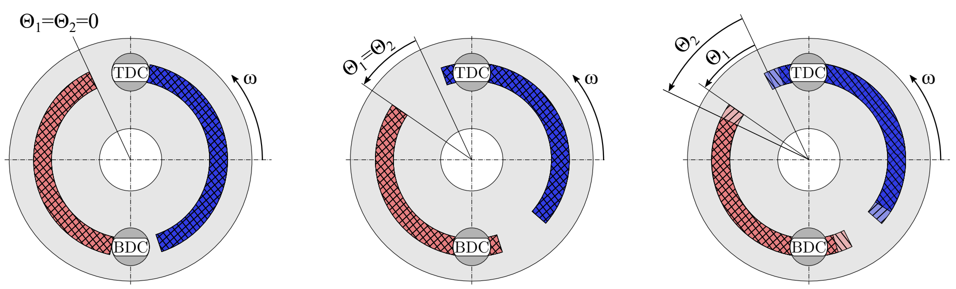
Alternatively, double-sided machines with separated sides can provide discretely variable displacement by circulating the flow of one machine side, e.g., using the setup sketched in Figure 7.2. This functionality is in principle available on the market where dual-flow machines can be equipped with shut-off valves (e.g., [27, 28]). This method can also be applied on double-sided machines with separated sides. However, when circulating oil at tank pressure level, the axial force is no longer balanced between the machine sides. To provide a better




(a) Single-sided machine.

(b) Double-sided machine with mirrored rotating group.

Figure 6.1: Rotating group outer diameter, axial length, and volume for floating piston machines with a displacement of 35 cc/rev. The crosses mark the designs of reference machines [21].




(a) Both valve plate rotation angles ($\Theta_1$, $\Theta_2$) are zero, and the valve plates are in the same position at maximum displacement.

(b) Joint valve plate rotation: both valve plates have been rotated by the same angle.

(c) Joint and relative valve plate rotation: valve plate 2 has been rotated by an additional angle.

Figure 7.1: Concept of joined and relative valve plate rotation [26].

balancing of the axial forces, an elevated low-pressure level can be achieved by a setup as sketched in Figure 7.2 [10].

## 8. Channel Design

The channel design depends on the machine type and on whether one or two cylinder barrels are used. When using just one cylinder barrel and axial ports, double-sided designs require flow through the pistons. This is a significant restriction, increasing the risk for cavitation and reducing self-suction speeds [21]. Mehta [8] avoided this for a slipper-type machine by using a

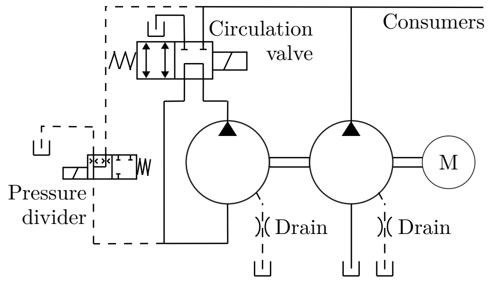



Figure 7.2: A possible setup for circulating flow from one pump half at an elevated pressure level [10].

central manifold and two cylinder barrels. A central manifold is however inconvenient for a radial integration of the hydraulic machine inside an electric machine.

The floating cup machines also uses two cylinder barrels and the flow goes through the cups, which can provide larger flow channels than pistons. The floating cup machine does not have a central manifold, instead it has two inlets and outlets, which can be combined through channels in the housing.

# 9. Exemplary Rotating Group Simulation Study

## 9.1. *Setup*

An exemplary simulation study aims to compare machines that are similar to traditional machines setups and to alternative machine setups. The study uses a lumped parameter model as sketched in Figure 9.1 using the software Hopsan [29]. Each machine has a displacement of 35 cc/rev. Table 9.1 shows the model inputs. The models inputs represent typical values for single-sided machines of bent-axis (A), inline (B), and floating piston (C) type, as well as for double-sided machines of floating cup (D) and floating piston (E) type. The inlet pressure is set to 3 bar. As commutation feature, pressure relief grooves as sketched in Figure 9.2 are applied. The pressure relief groove shape is set to triangular. The pressure relief grooves are manually tuned to reduce flow pulsations for operating with up to 3000 rpm and 350 bar. Only one quadrant is considered.

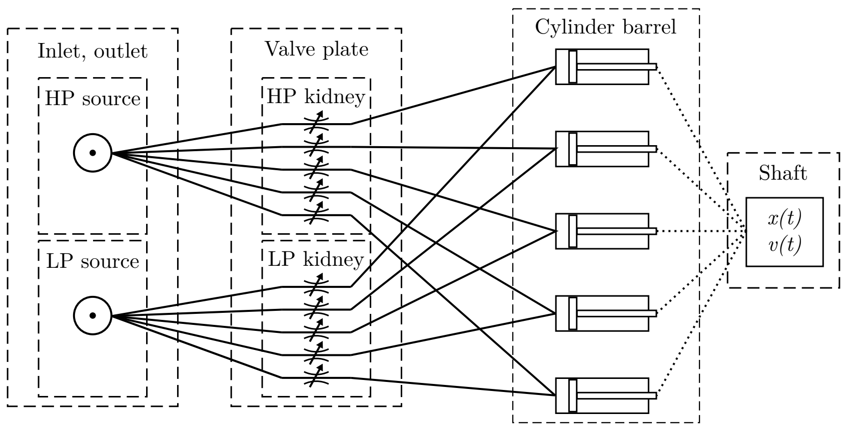



Figure 9.1: Simulation model for axial piston machines, modified from [11]. $t$ - time, $v$ - axial piston velocity, $x$ - axial piston position.

Table 9.1: Simulated setups.

| Setup | Machine type | Piston count | Swash angle in degrees | Dead volume ratio | Piston pitch parameter |
|---|---|---|---|---|---|
| A | Single-sided | 7 | 40 | 0.2 | 0.13 |
| B | Single-sided | 9 | 20 | 0.8 | 0.33 |
| C | Single-sided | 15 | 8 | 0.2 | 0.13 |
| D | Double-sided (separated) | 2x12 | 8 | 0.2 | 0.27 |
| E | Double-sided (connected) | 2x11 | 8 | 0.7 | 0.13 |

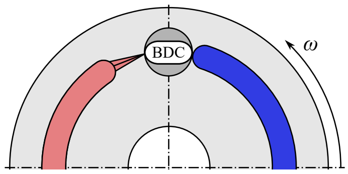
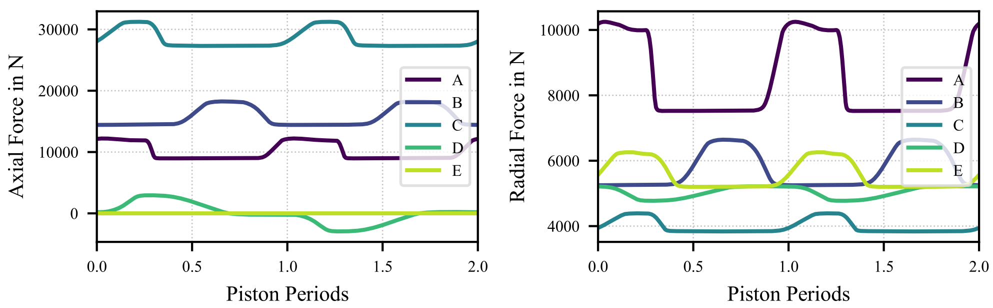



Figure 9.2: Pressure relief groove [18].

## 9.2. *Results*

Figures 9.3 to 9.6 show simulation results for the setups given in Table 9.1.

Figure 9.3 shows the axial and radial forces for the different setups. As expected, single-sided machines possess larger axial forces. The low displacement angle of the single-sided machine setup C causes high axial forces, and the bent-axis type machine setup A possesses large radial forces. The double-sided setups D and E provide low axial forces, but do not reduce the radial forces.




(a) Axial forces on bearings.

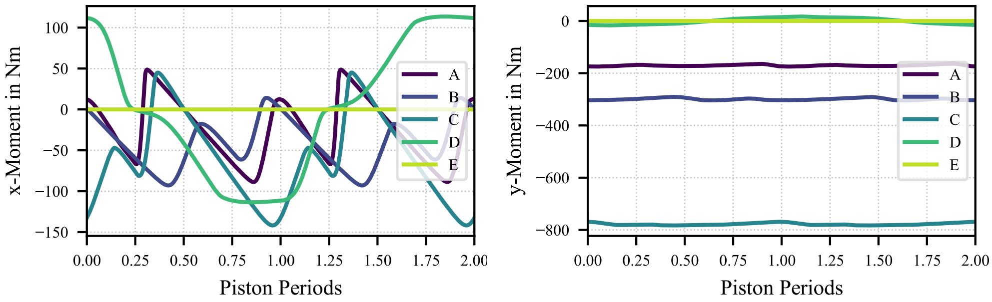
(b) Radial forces on bearings.

Figure 9.3: Comparison of bearing forces. Machine types see Table 9.1.

Figure 9.4 shows the bending moments for the different setups. For single-sided designs, increased displacement angles reduce the bending moment magnitudes. The double-sided setup E fully cancels the bending moments. The double-sided setup D which includes a phase shift between the sides has low moments around the y-axis but high moments around the x-axis.

Figure 9.5 shows the drive torque ripple. A higher piston count reduces the torque ripple. The double-sided design E without phase shift has significantly higher torque ripple than the double-sided design with phase shift D.

Figure 9.6 shows the flow ripples for the different setups. Similar to the drive torque ripple, a higher piston count reduces the flow ripple and a phase shift is advantageous for double-sided designs.




(a) Bending moments around the x axis.

(b) Bending moments around the y axis.

Figure 9.4: Comparison of bending moments on the cylinder barrel. Machine types see Table 9.1.

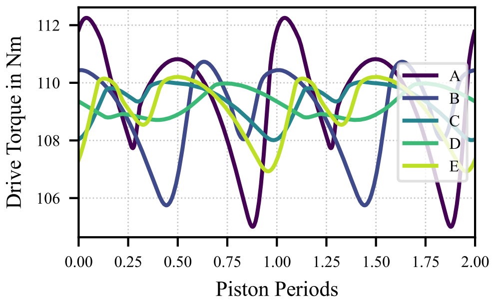
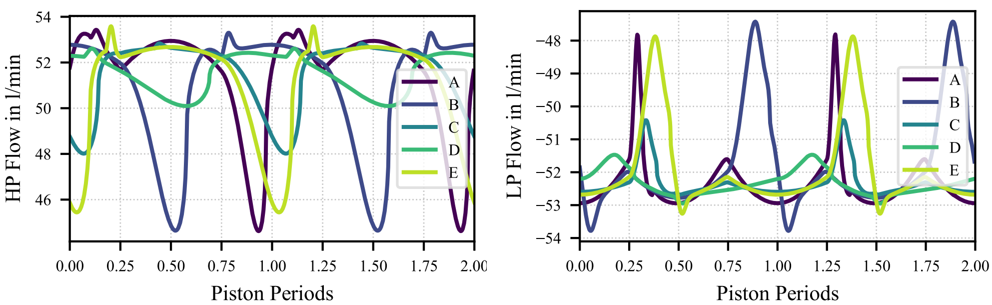



Figure 9.5: Comparison of drive torques. Machine types see Table 9.1.




(a) Delivered flow.

(b) Suction flow.

Figure 9.6: Comparison of flows. Machine types see Table 9.1.

## 10. Summary and Conclusion

Single-sided and double-sided machine can have the same displacement with different properties. Table 10.1 gives a simplified, qualitative summary of the properties of the different setups.

Table 10.1: Qualitative summary of properties, with "-" being the worst relative rating and "++" being the best relative rating.

| Property | Single-sided | Double-sided (connected) | Double-sided (separated w phase shift) |
|---|---|---|---|
| Axial force ripple | o | ++ | o |
| Side force ripple | o | o | + |
| Moment ripple (x-axis) | o | ++ | - |
| Moment ripple (y-axis) | - | ++ | + |
| Drive torque ripple | o | o | + |
| Bearing load (average) | - | ++ | + |
| Flow ripple | o | o | + |
| Aspect ratio | baseline | longer and thinner | longer and thinner |
| Variable displacement | well-proven for slipper-type | option for valve plate rotation (not well proven) | option to circulate flow (proven) |
| Channel design | + | depends on machine type | depends on machine type |

Double-sided machines can reduce noise contributors and bearing loads by cancelling of forces and moments between the machine sides.

Double-sided machines without a phase shift can in principle fully cancel axial forces and bending moments.

Double-sided machines with a phase shift increase the effective piston count, thereby reducing the amplitudes of most noise contributors. Axial forces and bending moments around the y-axis are largely compensated.

## 11. References

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