Proceedings of the ASME/BATH 2023
Symposium on Fluid Power and Motion Control
FPMC2023
October 16-18, 2023, Sarasota, Florida, USA

FPMC2023-111127

COUNTERACTING CENTRIFUGAL FORCES ON THE CUPS IN A FLOATING CUP PUMP

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ABSTRACT

Within hydraulic axial piston machines, critical components experience centrifugal forces, which can limit the performance of the actual machine. Often, this results in tipping of the cylinder block or barrel, causing friction in the barrel-port plate interface and affecting the efficiency and durability of the machine. In a hydraulic machine based on the floating cup technology, such as pumps, motors, and hydraulic transformers, the centrifugal force experienced by the cups is counteracted by the fixed pistons. The combined reaction forces on the cups cause a tipping torque on the barrel plate. Although a hydrostatic trust bearing can be used to compensate for such tipping, the effectiveness of the bearing depends on the pressure within the system and is limited in the amount of torque it can compensate for. This solution is sufficient for machines up to a certain size and a maximal rotational speed. However, if the machine requirements exceed these limits, a different solution is needed.

Ideally, the centrifugal forces on the cups are counteracted by the barrel plate, since it rotates around the same axis as the cups. This can be realized by introducing an opposite force using counterweights connected to the barrel plate. The centrifugal force on the counterweight is redirected via a lever to counteract the centrifugal force on the cup, minimizing the interaction between the cup and piston. As a result, the barrel plate tipping torque is diminished, decreasing the friction between the barrel plate and the port plate, which in turn results in an increase in efficiency and durability of the machine. The new solution also creates the opportunity to further increase the maximum rotational speed. This is especially important for electro-hydraulic actuators and hydraulic transformers.

Keywords: floating cup pump, barrel tipping, centrifugal force, counterweight

NOMENCLATURE

Symbol	Description	Unit
F_{centr}	Centrifugal force	N
F_{cont}	Contact force	N
F_{cw}	Counterweight forces	N
F_{fr}	Friction forces	N
F_p	Oil pressure force	N
F_r	Reaction forces	N
F_{ret}	Retainer plate force	N
L	Length	m
m_c	Mass cup	kg
m_{cw}	Mass counterweight	kg
R_{cw}	Pitch radius counterweight	m
R_p	Pitch radius piston	m
η_{cw}	Efficiency with counterweights	-
η_{wo}	Efficiency without counterweights	-

1 INTRODUCTION

In recent years, the hydraulic industry has become increasingly interested in electro-hydraulic actuators (EHAs). Such systems are typically used in the aerospace industry because of their high power density. Alongside this industry, EHAs are now also applied to other applications, such as off-road machinery, in an attempt to electrify these systems. For these applications, the possible advantages of EHAs over centralized hydraulic systems are its energy efficiency and, with the right electrical equipment, the possibility to recover energy. In particular, the requirement of high power density is of importance, since it affects the energy efficiency of the complete system. Whether an EHA is used in an airplane wing or mounted on the boom of an excavator, less mass results in smaller forces needed for acceleration. The reduction in required force translates to less energy usage by these systems.

One of the components of an EHA, which greatly influences its power density, is the hydraulic pump, not only because of the power-to-weight ratio of the pump itself, but also because of the effect on the sizing of the other components of an EHA. For example, the overall efficiency of the pump greatly influences the size of the cooling needed within the system, especially for applications within the mobile machinery market. When looking purely at the power density of the pump itself, it can be improved in two ways. One option is to raise the maximal allowable pressure of the pump, which is limited by several factors, such as material properties. The other option is to increase the maximum rotational speed of the pump, which is restricted by various effects such as cavitation, heat and tilting motion of the cylinder block, often called tipping [1]. If these restrictions can be overcome, other benefits of increased rational speed arise on a system level, especially for the electric motor [2].

The sizing of an electric motor in an EHA is often governed by the rated torque or the starting torque. When a certain maximum flow is prescribed for the application, the required displacement of the pump can be lowered if the maximum rotational speed is increased. Since the required rated torque is a product of the maximum pressure and the displacement of the pump, the required size of the electric motor decreases as well, as long as it meets the requirement on the starting torque. A smaller displacement volume also minimizes that the region in which the starting torque is prevalent [3].

To take advantage of these benefits, the floating cup principle tries to overcome the aforementioned restrictions on the rotational speed. In earlier research, it has been shown that barrel tipping has been a predominant factor in limiting the rotational speed of floating cup machines [4]. Unlike other axial piston machines, the pistons in a floating cup machine are rigidly mounted to the axle. The corresponding cylinders are no longer connected to the barrel by a form-closed construction. Instead, isolated cylinders or cups are floating by a hydrostatic balance on a barrel plate due to its cuplike shape. The centrifugal forces on these cups and their content causes them to tip around the spherical piston head. The sum of these centrifugal forces generates a tipping-torque on the barrel plate, destabilizing it at higher rotational speeds.

In previous research, the maximum rotational speed of the floating cup has been raised by reducing the wall thickness, lowering its mass and thus the effect of the centrifugal force [5]. It has been shown that this works well for a 24cc floating cup unit. However, when increasing the size of the pump, the centrifugal force acting on a cup becomes bigger because of multiple factors, causing barrel tipping at high rotational speeds. This article presents a new method to counteract the centrifugal forces acting on the cups by the use of counterweights, thus expanding the range of rational speed in which floating cup machines can be operated.

2 BARREL TIPPING IN THE FLOATING CUP

In hydrostatic machines based on the floating cup principle, the cylinders or cups are isolated and separated from the cylinder block. Instead, the remaining barrel plate provides a surface on which each separate cup can float. This is needed since the pistons are rigidly mounted on a flange of the axis, thus following a circular trajectory. The barrel plates are positioned at an angle with respect to the shaft, such that the cups slide along the pistons. Projected on the plane perpendicular to the shaft the cups make an elliptical shape, which is why they must be physically disconnected from the barrel. The trajectory of the cups on the barrel plate differs from this elliptical shape depending on the connection between the barrel plate and the shaft [6].

The assembly of the rotary group is shown in Figure 1. It shows that the spherical piston crown determines the x and y-position of the cup on the plane of the barrel plate, while the cup can move along the z-axis according to the barrel plate movement. To minimize leakage, the cup must be pressed to the barrel plate such that the gap between them is only a few micrometers high. This is realized in two ways. Firstly, the dimensions of the cup bottom can be altered such that the pressure in the cup presses it down. However, this downward facing force is pressure dependent and is thus very small during the low-pressure suction stroke. Secondly, the cup is held in place by the retainer plate, which has multiple functions. The retainer plate makes sure that the cups are not lifted from the barrel plate during the suction stroke and holds the cups in place during assembly. It can also provide an additional force, pressing the cup onto the barrel plate. Together with the barrel plate, the retainer plate counteracts tipping of the cups.

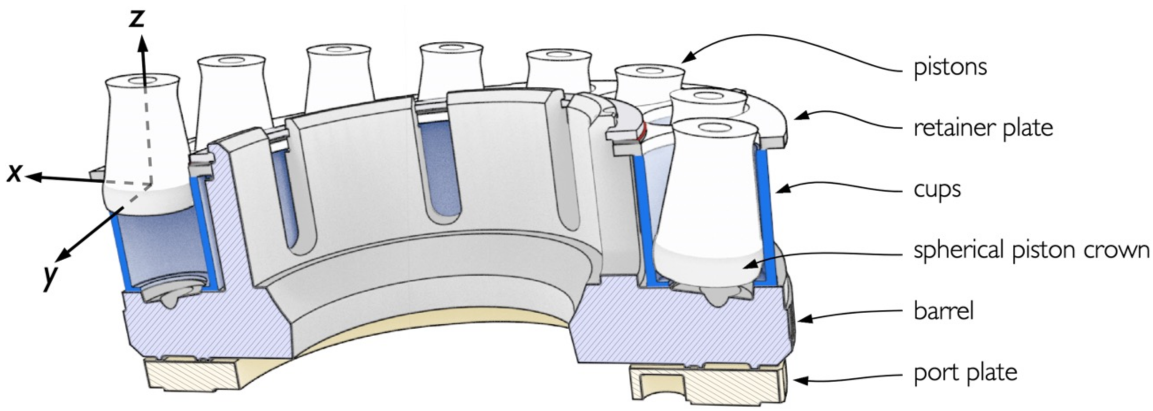

Fig. 1: Cross section of the barrel plate and port plate of a floating cup pump along the TDC-BDC line, including some of the pistons.

The tipping torque of the cup is a result of several forces, which are depicted in Figure 2. Due to the rotational movement, the cup experiences a centrifugal force F_{centr} from the mass of the cup and the oil inside of it. At high rotational speeds, this force becomes the predominant factor and introduces a torque around the center of the piston crown, denoted by *point A*. The arm of the torque depends on the position of the piston in the cup and the position of the center of gravity (CoG), denoted by *point B*. The centrifugal force is balanced at the piston-cup interface by the contact force F_{cont}, with the addition of other radial forces. Since the piston is moving upwards, there is a friction force $F_{fr,1}$ on the cup in the same direction. There are two forces pressing down on the cup, the pressure dependent force as a result of the dimension of the cup bottom F_p and the force from the retainer plate F_{ret}. The axial forces are balanced by a reaction force F_r. Since the cup is moving sideways with respect to both the barrel and retainer plate, the retainer force and reaction force also generate friction forces $F_{fr,2}$ and $F_{fr,3}$ respectively. The combined forces generate a torque around the center of the piston crown, which determines the radial position of the reaction force F_r, balancing the torques.

It is important to notice that Figure 2 is a simplified two-dimensional depiction of the real-world situation. A couple of remarks can be made. Except for the centrifugal force, the exact magnitudes of the forces are unknown, since they rely on many assumptions. Due to the design of the retainer plate, the exact position of the retainer force is unknown. However, it can be concluded that the retainer force should be large enough to counteract the tipping, but not too large to induce more tipping torque due to increased friction forces. More importantly, it can be concluded that the centrifugal forces on the cups are counteracted by both the pistons and the barrel plate. Therefore, not only the cups experience a tipping torque, but so does the barrel plate as a result of the combined tipping torques of the cups as shown in Figure 3.

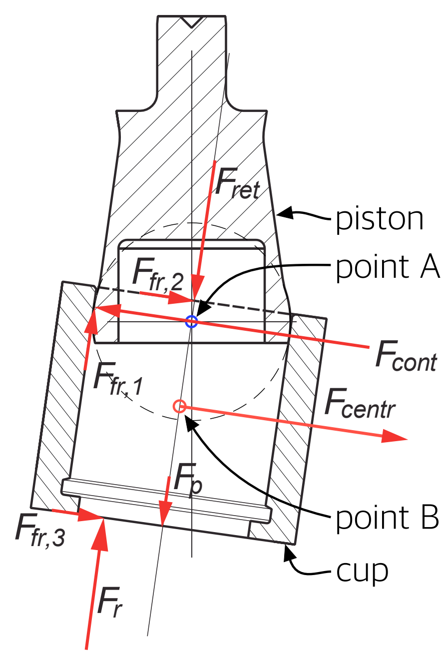

Fig. 2: Radial and axial forces on the cup, in a cylindrical coordinate system defined by the rotational axis of the barrel plate. In this case the piston has a relative movement upwards.

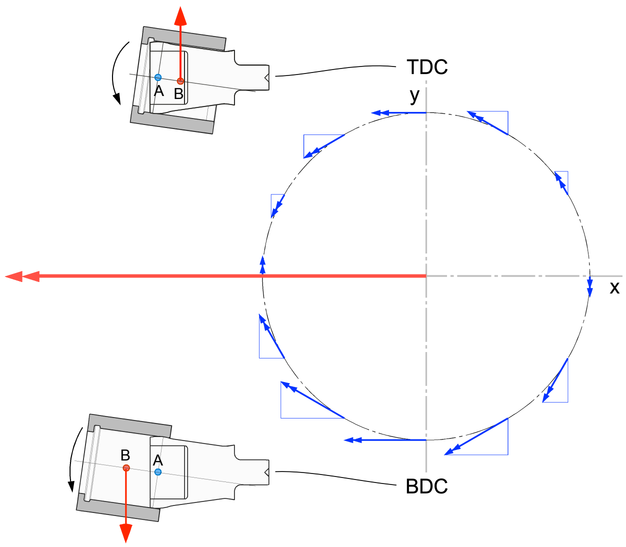

Fig. 3: The tipping torque of each individual cup (blue) adds to the total tipping torque on the barrel plate (red), depicted for a pump with 12 pistons per barrel.

In many ways, the floating cup design already tries to minimize the centrifugal forces and the subsequent tipping torques. The arm of the cup tipping torque is kept small by the relatively short stroke of the pistons. The wall thickness of the cups has been reduced, sequentially reducing its weight and thus minimizing the centrifugal force. Since the cylinders are decoupled from the barrel, it does not have to deal with the manufacturing tolerances of conventional axial piston machines. Neither does the design include a spline to transfer power and the barrel does not translate any torque to the shaft. Instead, the current design of floating cup machines uses so-called pockets to prevent the barrel from tipping [7]. These pockets utilize oil pressure to create a hydrostatic bearing. At low discharge pressures and high rotational speeds, this bearing becomes less effective at counteracting the centrifugal force induced barrel tipping. This is a problem when increasing the maximum rotational speed of floating cup machines, considering that the centrifugal force increases quadratically for a linear increase in rotational speed. Additionally, the centrifugal force on the cups becomes problematic when increasing the displacement volume of floating cup machines. Both the mass of the cup and its content increase, as well as the piston pitch radius. Also, the effective arm of the centrifugal force, which is the distance between center of gravity of the cup and the center of the piston crown, is increased. Thus, the centrifugal force increases significantly for floating cup machines with a larger displacement volume, as well as for machines for which an increase in rotational speed is needed. To counteract this centrifugal force, INNAS built in counterweights into a 45cc floating cup pump.

3 COUNTERWEIGHTS

The current configuration of these counterweights is shown in Figures 4, 5, 6 and 7, and uses 3D-printing technology. In this configuration, the hinge of the counterweight is placed in parallel to the axis of rotation of the barrel, such that the counterweight can be placed in the space between the cups. The hinge is a contact line between the counterweight and a support ring on the outer diameter of the barrel, which contains holes for the positioning pins on the counterweights. The contact between the counterweight and the cup is not a single point but a line to spread the force, since the contact force can become quite large. To prevent wear due to this high contact force, the counterweights have been tempered after the 3D-printing process. The triangular shape helps to have more weight on the side of the hinge which is not in contact with the cup, as well as making sure that it does not rotate around its positioning pen due to its entrapment by the barrel plate and the retainer plate. The counterweight has been designed such that its movement does not interfere with the neighboring cup and support ring.

Figure 6 shows how the centrifugal forces balance. The centrifugal force F_{centr} is always radially pointing outwards. Although the axial position of the CoG changes due to the variable oil volume, the mass of the cup is the dominant factor and the CoG deviates less than a millimeter from the center over the full rotation. The radial position of the cup and its CoG also change slightly over the length of the rotation. These small movements allow for a counterweight lever with a single degree of freedom, to account for this radial movement of the cup. The part of the lever with length L_1 is made heavier and experiences an outward centrifugal force $F_{cw,o}$. Subsequently, the ratio L_1/L_2 determines the counteracting inward force $F_{cw,i}$. The position of the contact line influences the required weight, such that the ratio for the effective weight of the counterweight m_{cw} and the weight of the cup m_c is given by

$$\frac{m_{cw}}{m_c} = \frac{R_P L_2}{R_{cw} L_1} \tag{1}$$

where R_P and R_{cw} denote the pitch radii of the piston and the counterweight respectively. Constructing the lever such that this force is in line with the CoG almost perfectly cancels F_{centr}.

The remaining forces on a cup are shown in Figure 8. The torque on the cup has been greatly reduced, removing the need for a downward facing force from the retainer plate. Since the centrifugal force has been canceled out by the counterweight force, the contact force, and thus the friction between the piston and the cup has been removed. Consequently, the friction between the cup and the barrel plate is solely influenced by the reaction force F_r from the pressure dependent force F_p. Since the centrifugal force is no longer compensated by the piston, the tipping torque it causes on the barrel plate is also nullified. This reduces the wear between the barrel plate and the valve plate, as well as allowing for an increase in rotational speed.

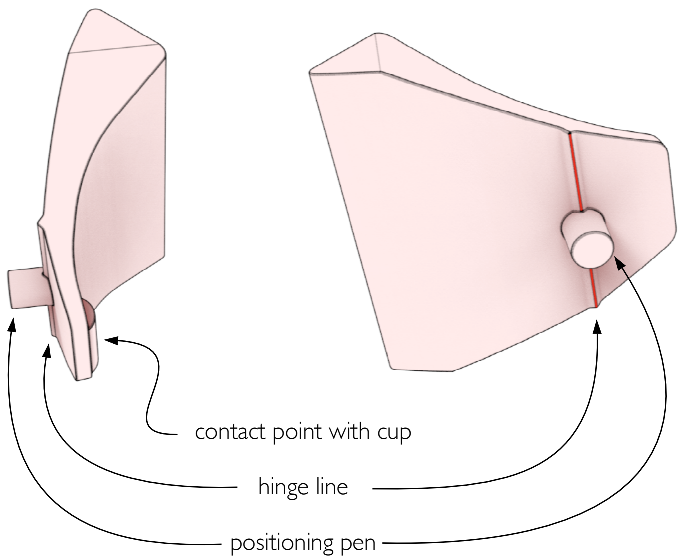

Fig. 4: 3D views of a counterweight.

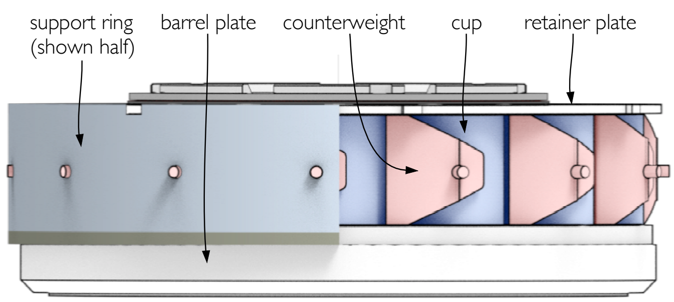

Fig. 5: Positioning of the counterweights in a barrel assembly.

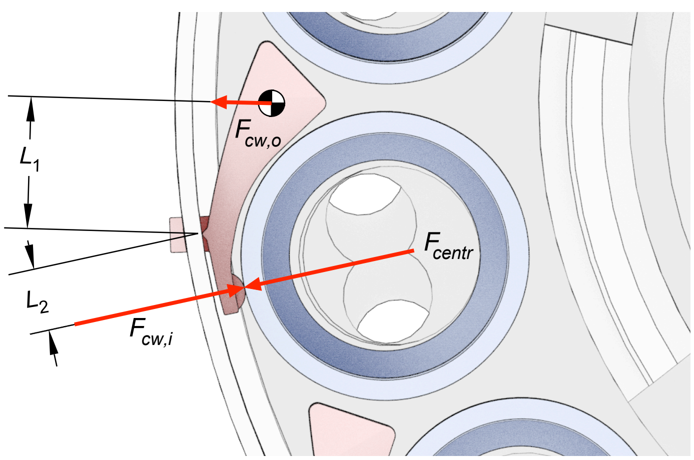

Fig. 6: A counterweight counteracting the centrifugal force on a cup.

Fig. 7: A floating cup with a counterweight. The band on the cup is a result of the contact with the counterweight.

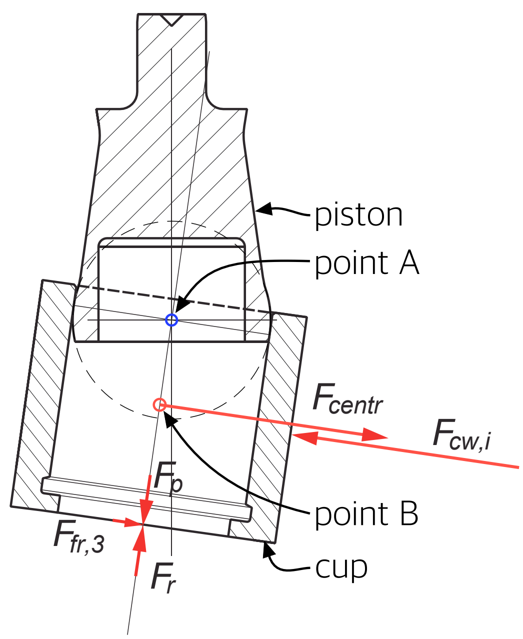

Fig. 8: The remaining forces on a cup when a counterweight is introduced.

4 PERFORMANCE MEASUREMENTS

For this research project, a 45cc fixed displacement floating cup pump, as developed by INNAS, is used. The barrel plates contain the aforementioned pockets, as well as shuttle technology [8]. The pump has been tested twice on the test bench of INNAS [9] as shown in Figure 9, once without the counterweights and once with counterweights. The supply pressure is set to 4 bar, such that cavitation is avoided at the highest rotational speeds and the inlet temperature of the oil is kept at 50°C.

The objective of this test is to investigate the maximum rotational speed of the pump for each configuration at different pressure levels. While increasing the rotational speed, the leakage flow and output pressure have been closely monitored. Figure 10 shows a part of the measurement data for each pump configuration at 50 bar at different rotational speeds. It can be seen that with increasing the rotational speed to 4500 rpm, the leakage flow not only increases, but also varies more over time. This behavior is, together with the rapid variation in output pressure, used as indicator for barrel tipping. Pushing the limit any further would have catastrophic consequences for the pump. Therefore, the pump is not tested beyond these operation points.

Figure 11 shows the measurement data for both configurations at 50 bar. The leakage flow of the pump without counterweights steeply increases after 3500 rpm. When the centrifugal force is counteracted by counterweights, the increase in leakage is more gradual, as well as being lower overall. Additionally, the pump with counterweights is able to go up to 5000 rpm, which is equal to the maximum rotational speed of the test setup.

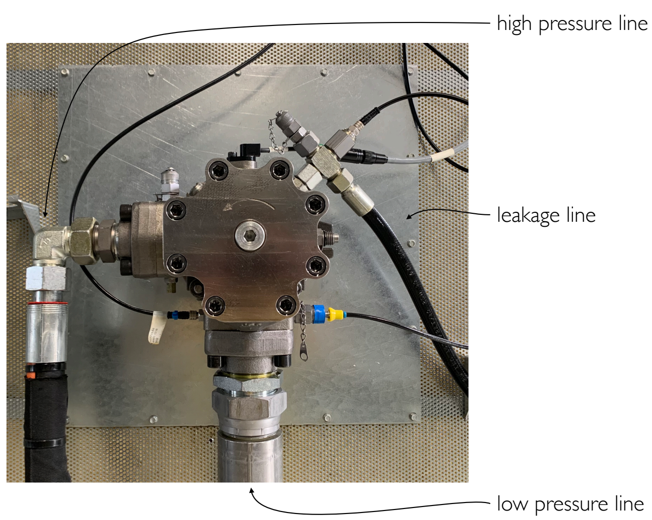

Fig. 9: The 45cc fixed displacement floating cup pump on the INNAS test bench, of which the detailed information is given in [9].

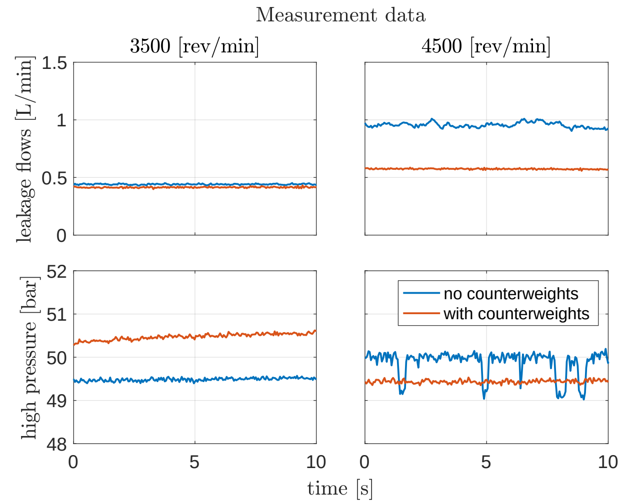

Fig. 10: Measurement data of the FC45 tested at 50 bar for different rotational speeds. At an rotational speed of 4500 $[rev/min]$, leakage flow and the high pressure of the pump without counterweights show indications of barrel tipping.

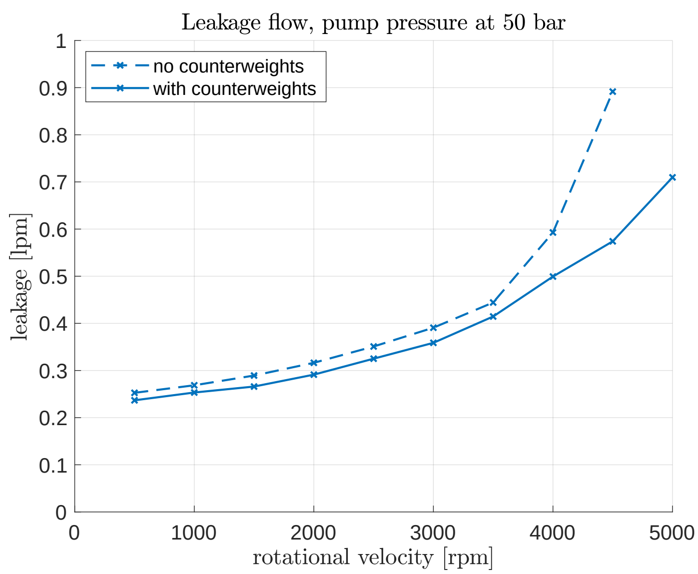

Fig. 11: The leakage flow from the pump at 50 bar, measured with (solid lines) and without (dashed lines) counterweights.

This procedure has been repeated at different pressure levels, as shown by Table 1. The counterweight configuration clearly has a larger operating range. Due to the restrictions of the test setup, it is unclear whether higher rotational speeds can be reached. Figure 12 shows the leakage flow for every measurement point, with similar behavior at high rotational speeds. Below 2500 rpm, the configuration with counterweights shows an increase in leakage for higher pressures in comparison to the other configuration. A possible explanation could be found in the difference in barrel plate balance, especially since the hydrostatic bearing load is changed. This hypothesis requires more research.

Figure 7 shows a cup and its corresponding counterweight after the tests. The only notable difference can be seen on the outer wall of the cup, where the lighter band around the cup shows where the counterweight has contact with the cup. The markings are very shallow and should not decrease the lifespan of the components. To test this hypothesis endurance tests are recommended for future research. The markings of the contact point form a band as a result of tangential movement of the cup relative to the piston, which caused by the movement of the cup on the barrel plate [6].

For both configurations the efficiency has been calculated according to [10] and is shown in Figure 13. The difference in efficiency is shown in Figure 14. From these figures, it can be concluded that the counterweights have almost no influence on the efficiency of the pump for these rotational speeds.

Tab. 1: Operating points for the both performance test. The cells marked in green show the operating points which could only reached with counterweights.

	Pump pressure [bar]				
n [rpm]	50	100	200	300	400
500	✓✓	✓✓	✓✓	✓✓	✓✓
1000	✓✓	✓✓	✓✓	✓✓	✓✓
1500	✓✓	✓✓	✓✓	✓✓	✓✓
2000	✓✓	✓✓	✓✓	✓✓	✓✓
2500	✓✓	✓✓	✓✓	✓✓	✓✓
3000	✓✓	✓✓	✓✓	✓✓	✓✓
3500	✓✓	✓✓	✓✓	✓✓	✓✓
4000	✓✓	✓✓	✓✓	✓✓	✓✓
4500	✓✓	✓✓	✓✓	✓✓	✗✓
5000	✗✓	✗✓	✗✓	✗✓	✗✓

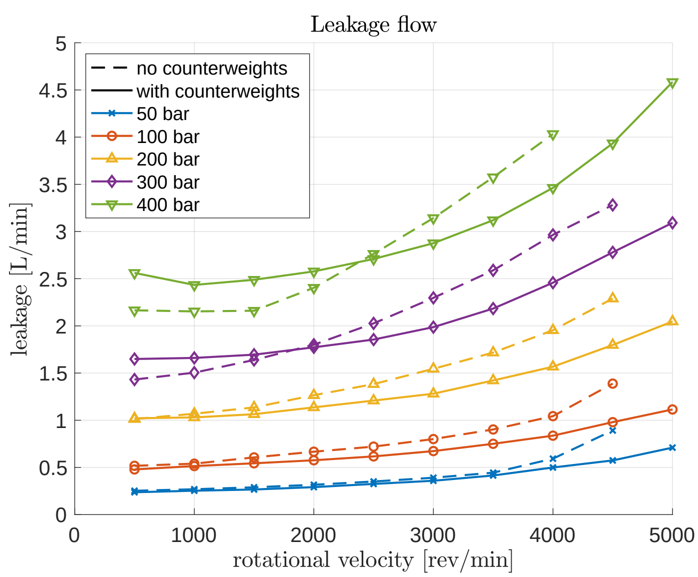

Fig. 12: The leakage flow from the pump for different pressure levels, measured with (solid lines) and without (dashed lines) counterweights.

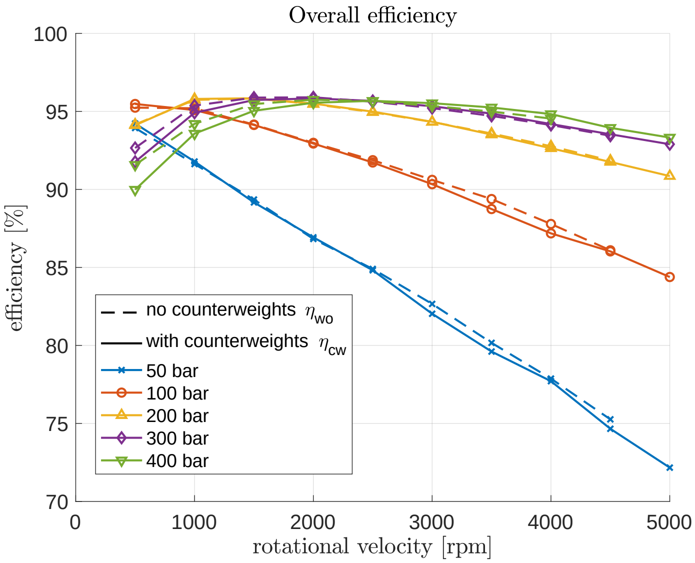

Fig. 13: The pump efficiency for different pressure levels, measured with (solid lines) and without (dashed lines) counterweights.

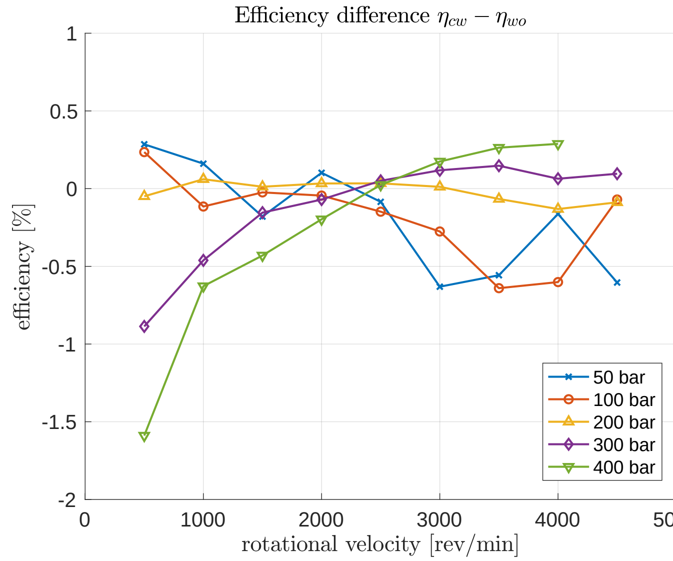

Fig. 14: The difference in pump efficiency for different pressure levels, where the efficiency of the pump without counterweights has been subtracted from the efficiency of the configuration with counterweights.

5 CONCLUSION

In this article a new way of counteracting the centrifugal forces on a floating cup has been introduced, as a means to extend the possible field of operation of a floating cup machine to higher rotational speeds. This counteracting has been done by applying counterweights to the barrel assembly, almost perfectly canceling out the centrifugal force on the cups and reducing the tipping torque on the barrel plate consequently. Putting this solution to the test has shown that applying counterweights decreases the overall leakage and allows for higher rotational speeds. The tests have been limited by the maximum rotational speed of the test bench, so more research is needed to test the full potential of this new technology.

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