Reducing the wall thickness of the cups and pistons in Floating Cup pumps and motors

Peter Achten, Sjoerd Eggenkamp, Jeroen Potma

INNAS BV, Nikkelstraat 15, 4823AE Breda, The Netherlands
E-Mail: pachten@innas.com

The rotational speed of slipper type, axial piston pumps and motors is limited. One of the most important reasons for this limitation is the barrel tipping torque, which is (amongst others) affected by the centrifugal forces of the pistons. The force of the barrel spring is needed to overcome the tipping of the barrel, and thus preventing the malfunction of the pump or motor. The hydrostatic pressure can create an additional hydrostatic force, pushing the barrel to the port plate, and thereby preventing the barrel to tip. But, at low operating pressures, the hydrostatic force is insufficient, and the tipping torque can only be counteracted by the central barrel spring. Due to the limited strength of this spring, the barrel will tip above a certain operating speed. At that point, the face seal of the barrel will no longer make a full contact with the port plate, and the pump or motor cannot any longer be operated, due to excessive leakage and wear.

Floating cup (FC) pumps and motors have the advantage that the pistons are press-fitted into the central rotor. Therefore, the pistons can't create anymore any tipping torque load on the barrel. But, unlike in conventional axial piston designs, in FC-machines, the cylinders are no longer integrated and machined inside the cylinder block or barrel. Instead, they are separated and have become cup-like cylinders which are floating on, and supported by the remaining barrel plate. Being isolated from the barrel itself, these 'cups' will create another tipping torque load on the barrel. The weight of the cups is however small, much smaller than the weight of the pistons in an equally sized slipper type machine. Furthermore, the centrifugal tipping torque is reduced by the short cup-stroke in the floating cup machine.

Nevertheless, it is desirable to further reduce the mass of the cups, not only to increase the maximum operating speed, but also to reduce the force of the central barrel spring, which would further increase the overall efficiency. The best way to reduce the mass of the cups, is by means of a reduction of the wall thickness of the cups. It needs to be considered that the cups expand when being pressurized. Therefore, in order to maintain a tight sealing between the cups and the pistons, the piston crown needs to expand as well, thereby following the radial expansion of the cup. Consequently, the wall thickness of the piston crown needs to be reduced as well.

The question is whether these conditions can be met when the wall thickness of the piston crown and cup wall is reduced. In this paper, the effects of a 50% reduction of the wall thickness are investigated and described. The deformation is calculated by means of FEM analysis of the piston and the cup.

Keywords: Floating cup, FEM-analysis, barrel tipping torque
Target audience: Hydrostatic pump and motor designers and manufacturers

1 Introduction

Hydrostatic piston pumps and motors are positive displacement machines. In Floating Cup (FC) pumps and motors, the displacement is created by cup-like cylinders. These 'cups' are translating on pistons, which have a ball shaped, spherical piston crown. Unlike in other axial piston designs, the pistons are press-fitted into the rotor. Compared to slipper type and bent axis machines, the FC-design is therefore more or less inverted: the pistons are locked, and the cylinders, the 'cups', are free to move on the barrel plate. One of the advantages of the new design is, that the centrifugal forces of the pistons no longer create a torque load on the barrel. This so-called 'tipping torque' is one of the most important reasons why axial piston machines cannot be driven above a certain operating speed [1-3].

However, in the FC-principle, the cylinders are isolated from the barrel. The 'cups' can slide and translate on the remaining barrel plate. But, due to their degree of freedom, they will also create a new tipping torque on the barrel. To make things even worse, the FC-principle is a multi-piston principle, typically having around 12 pistons and cups per barrel. This is more than the average 9 pistons, that cause a tipping torque load in slipper type machines, and the larger number of cups will increase the tipping torque load. The barrel tipping torque is also strongly influenced by the radius of the pitch circle of the pistons. However, the FC-principle does not increase this radius [4], and therefore there does not have any effect on the tipping torque in comparison to slipper type machines having the same geometrical displacement.

On the positive side, the FC-principle has a swash angle of around 8°, which is less than halve the swash angle of most slipper type machines. This is advantageous for reducing the tipping torque, since the small swash angle results in a small stroke of the cups. In other words: the center of mass of the cups is always close to the midpoint of the piston head, which results in a smaller arm for the centrifugal forces to create a torque load.

But the most important effect is the difference between the mass of the cups and the mass of the pistons of a comparable slipper type machine. Figure 2 shows the cup of a floating cup machine and the piston of a slipper type machine, both having a displacement of 28 cc. The mass of the cup is 13.1 gr, being only 23% of the weight of the piston of the slipper type pump, which has a mass of 56.2 gr.

Fig. 1.: Piston of a 28 cc slipper type pump (56.2 gr) and cup of a 28 cc FC-pump (13.1 gr)

The combined effect of the reduced stroke and the lower mass of the cups results in a strong reduction of the tipping torque [3], even including the detrimental effect of the large number of cups. Nevertheless, at some point, the centrifugal forces of the cups still limit the maximum operating speed of the FC-pumps and motors. In order to further reduce the tipping torque load, i.e. to increase the maximum rotational speed, this study investigates the opportunity to reduce the mass of the cups by means of a reduction of the wall thickness. In a 24 cc prototype, the wall thickness of the cups has been reduced from 2.25 mm to 1.1 mm, a reduction of about 50%. FEM-analysis have been performed to calculate the effect of the reduced wall thickness on the deformation of both the cup and the piston crown.

2 The effects of centrifugal forces on the tipping torque

The cups create several loads on the barrel, many of which result in a tipping torque load on the barrel. Figure 2 shows the forces acting on the cup, excluding the hydrostatic forces. The most dominant forces are the centrifugal force F_{centr} and the friction force $F_{fr,1}$ between the cup and the piston. The last-mentioned friction force is an indirect effect of the centrifugal force. The centrifugal force creates a lateral reaction force in the contact between the piston and the cup, thus causing a friction force. Since the centre of mass of the cup (point B in Figure 2) differs from the point A of rotation, there is a reaction force $F_{r,2}$ in the contact between the cup and the barrel plane. All these reaction forces of the cups together create a tipping torque on the barrel.

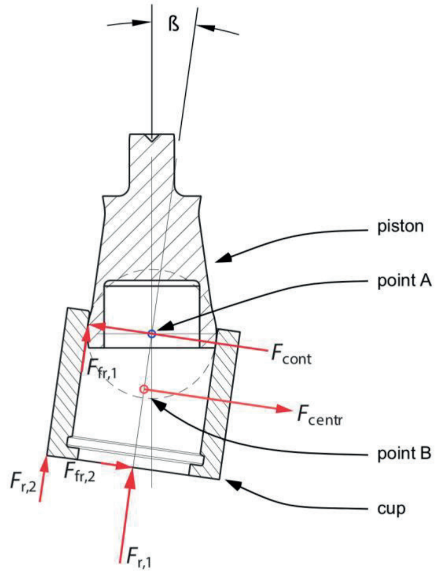

Fig. 2: Forces acting on the cup. Point A is the center of the sphere of the ball-shaped piston crown. Point B is the center of mass of the cup and ots oil contents.

The effects of these forces on the tipping torque can be calculated. First of all, there is the direct effect of the centrifugal forces of the cups on the tipping torque. These create a torque load on the barrel in the x-direction (see Figure 3). In the y-direction, the sum of all centrifugal torque loads amounts to zero: the torque loads are cancelled internally. Figure 3 also illustrates how the torque vector is influenced by the cup position on the piston crown. In the Top Dead Centre (TDC) the centre of mass of the cup, point B, is to the right of the point of rotation A. In the Bottom Dead Centre (BDC), point B has shifted to the left of the point of rotation A. The result is that all torque vectors are pointed towards the minus-x-direction. The resulting torque tends to tip the barrel towards the TDC. This is independent of the direction of rotation of the pump or motor.

The second cause for the tipping torque is the friction between the cups and the pistons. This force is directly related to the centrifugal force. In the case of solid or mixed friction, the centrifugal load of the cups is taken in the contact between the cup and the piston, thereby creating a friction force. Even when hydrodynamic lubrication is assumed, the centrifugal force will push the cup towards one side of the piston, thereby reducing the gap height of the oil film, and increasing locally the viscous friction.

It is important to realize that the FC-principle does not create any hydrostatic load between the piston and cylinder, as is the case in slipper type machines. The only remaining lateral load on the piston is created by the centrifugal force of the cup. Although the magnitude and direction of the centrifugal load can be calculated accurately, the magnitude of the friction force very much depends on the unknown tribological conditions. However, the direction of the friction force is well defined: when moving from TDC to BDC, the friction force pushes the cups onto the barrel plate. From BDC to TDC the friction force is in the opposite direction. The resultant tipping torque will therefore be on the y-axis. The direction of this torque vector is dependent on the direction of rotation of the main shaft. The magnitude of the friction tipping torque strongly depends on the friction coefficient. Assuming a value of around 0.1, the friction force itself is much smaller than the centrifugal force. But, since the friction force creates a tipping torque around an arm which is about 10 times larger than the centrifugal tipping torque, both tipping torques have about the same magnitude.

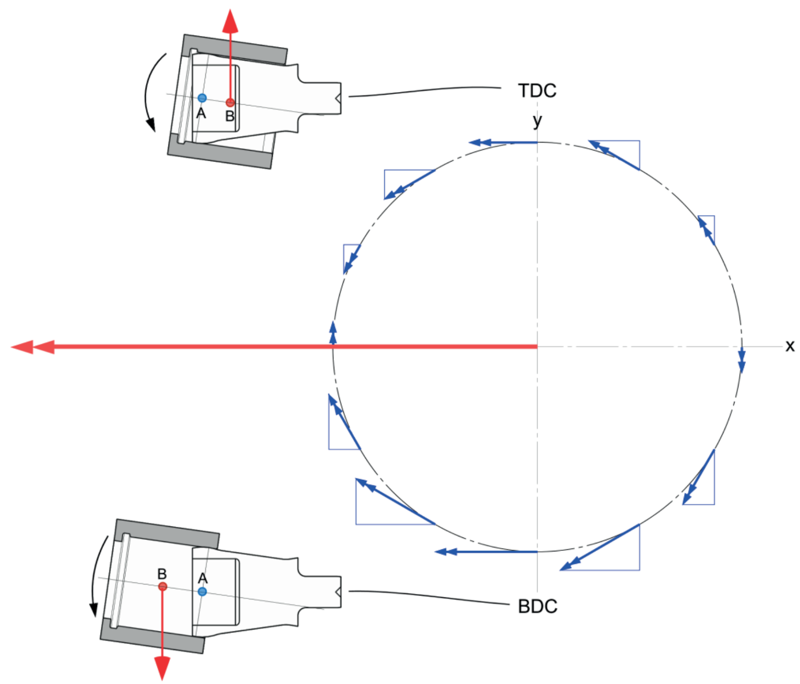

Fig. 3: Barrel tipping torque generated directly by the centrifugal forces of the cups (TDC = Top Dead Centre, BDC = Bottom Dead Centre)

The third tipping torque is caused by the friction between the rotating barrel and the stationary port plate. Due to the previous two torque loads, the barrel will already slightly incline towards a certain direction. At the point where the gap height is smallest, the friction between the barrel and the port plate will be higher, which will generate another tipping torque.

Figure 4 shows all three torque vectors, as is calculated in the simulation model for pump operation and projected on top of the port plate. The high-pressure side is on the right side of the port plate. The calculation has been performed assuming a near to zero pump pressure. Pressure dependent torque loads are therefore not included. The figure shows that the resultant torque load tends to tip the barrel towards the high-pressure side of the port plate, having the smallest gap height around 45° before the TDC. If a barrel would start tipping, this is where it would make contact with the port plate.

This has been confirmed with a number of experiments. Figure 6 shows a test of about 5 minutes, in which a 24 cc constant displacement pump has been operated at the lowest possible pump pressure, while gradually increasing the speed. At around 250 seconds, the pump reached a speed of 4920 rpm. At that point, the case drain flow suddenly increased and the pump pressure dropped to zero. The test was performed on a 4-quadrant hydrostatic machine which demanded a pre-charge pressure level at the supply side during pump operation.

After inspection of the pump, a clear mark of the barrel edge was found (see Figure 5). This mark can only be explained by the tipping of the barrel. The mark was visible on both port plates of the floating cup pump, indicating that the critical tipping speed was the same for both barrels. Since both barrels are identical, this was expected.

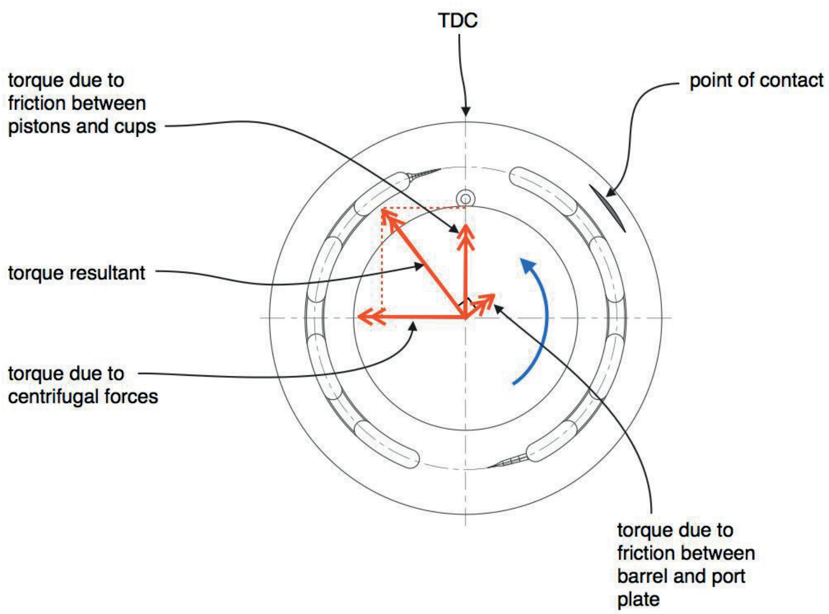

Fig. 4: Centrifugal and friction tipping torque acting on the barrel

Fig. 5: Port plate showing the tipping mark of the barrel

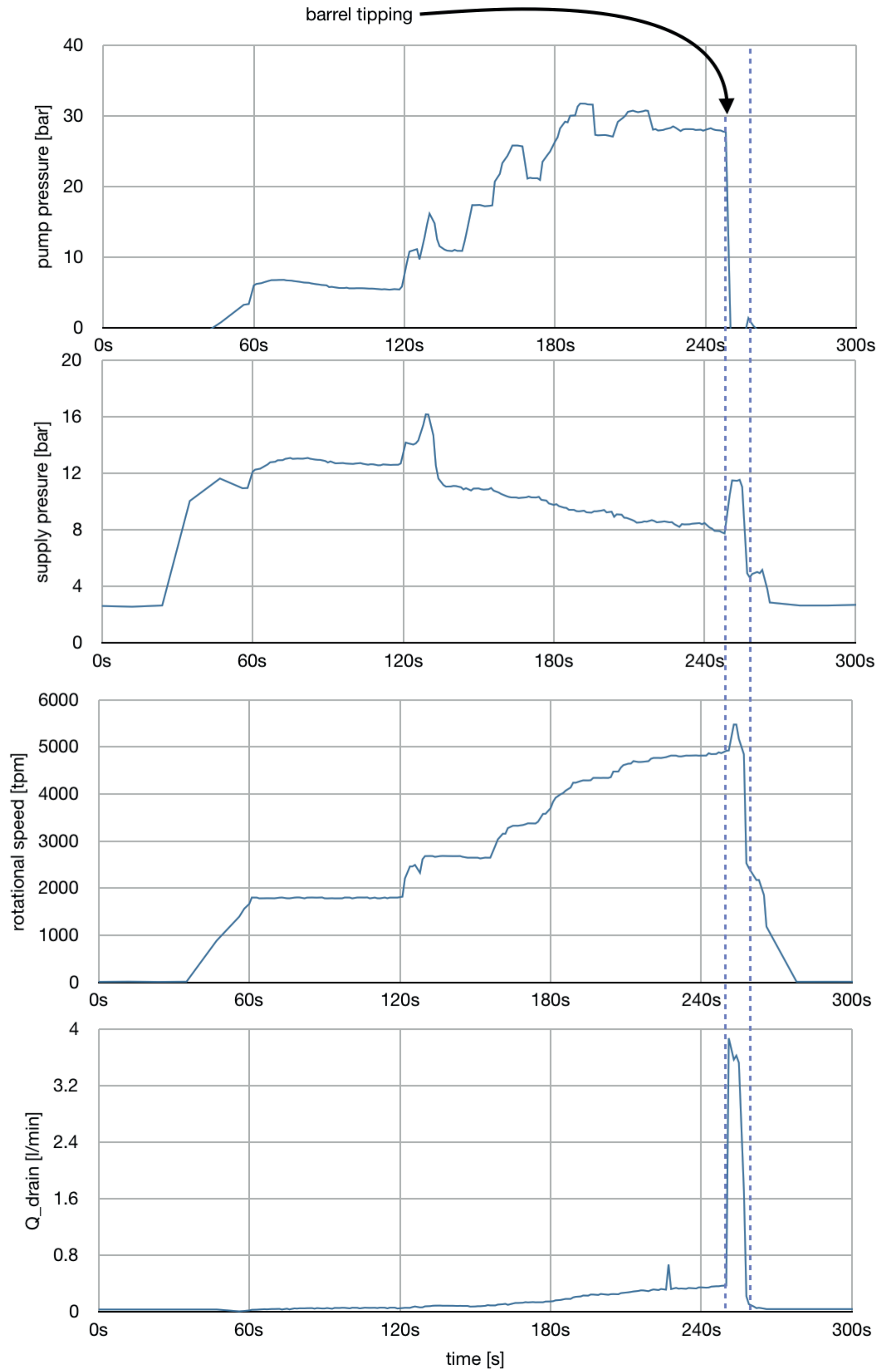

Fig. 6: Test of a 24 cc constant displacement FC-pump at low operating pressures (Oil temperature 40°C, HLP46)

3 Reducing the wall thickness of the cups

The centrifugal force of the cups is the root cause of the barrel tipping at low operating pressures. The barrel tipping could be reduced by increasing the strength of the central spring, which pushes the barrel towards the port plate. But this would create an additional bearing load in the contact between the barrel and the port plate, thereby increasing the friction losses. The most adequate way to reduce the centrifugal tipping torque is to reduce the mass of the cups.

The centrifugal tipping torque T_{ct} is linearly dependent on the mass of the cups:

$$T_{ct} \equiv m_{cup} \cdot n^2 \tag{1}$$

Without a hydrostatic load, the tipping torque can only be counteracted by the central barrel spring, which has a certain spring force F_{sp}. The barrel will tip whenever the centrifugal tipping torque is larger than the counteracting torque generated by the barrel spring. Consequently, there is a critical rotational speed n_{cr}, which is dependent on the spring force and the cup mass:

$$n_{cr} \equiv \sqrt{\frac{F_{sp}}{m_{cup}}} \tag{2}$$

In this study, the mass of the cups of a 24 cc FC-pump is reduced from 12.3 gr to 5.16 gr, a reduction of 61.3%. This would result in an increase of the maximum speed of 54.4%. With the original cups, a critical tipping speed of 4920 rpm has been measured. With the new, light cups, the critical speed would (in theory) be increased to 7600 rpm.

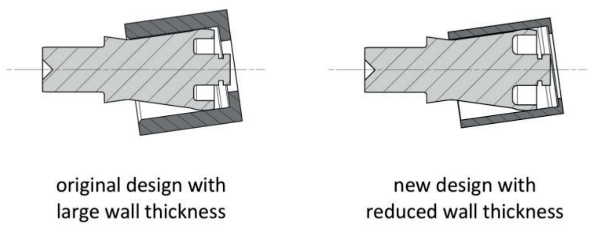

Fig. 7: Old and new designs of piston and cup

The weight reduction has been achieved by reducing the wall thickness of the cups. Figure 7 shows the original and the new design, side by side. The thickness of the wall has been reduced from 2.25 mm to 1.1 mm. The cups have an inner diameter of 12.5 mm. Also, the thickness of the cup base has been reduced from 1.5 mm to 1 mm.

Figure 7 also shows, that, aside from the cup design, also the piston design has been adapted. This is necessary to keep the gap between the piston crown and the cup small, even when the components are being pressurized [4]. The reduced wall thickness reduces the stiffness of the cup. Consequently, the thin-walled cups will expand more in the radial direction when being exposed to the internal hydrostatic pressure. In order to follow this expansion, the stiffness of the piston crown needs to be reduced as well.

4 FEM-analysis

FEM-analysis have been performed to investigate the deformation of the new cup and piston design. Several designs have been reviewed. Each design configuration requires a number of FEM-analysis at various positions of the cup on the piston crown. The analysis not only involves the radial expansion of the cup and the piston crown, but also calculates the axial deformation of the cup base. The cups are floating on the barrel plate (hence the name Floating Cup), and the cup base acts as a face sealing and bearing interface. The deformation of the sealing area strongly influences the sealing and bearing capacity between the cups and the barrel plate. Figure 8 shows the (enlarged) calculated deformation of two different cup designs. In the left design, the deformation of the cup base results in a convergent gap profile, whereas the right design creates a divergent gap profile. Both calculations are made for the same piston position and pressure load.

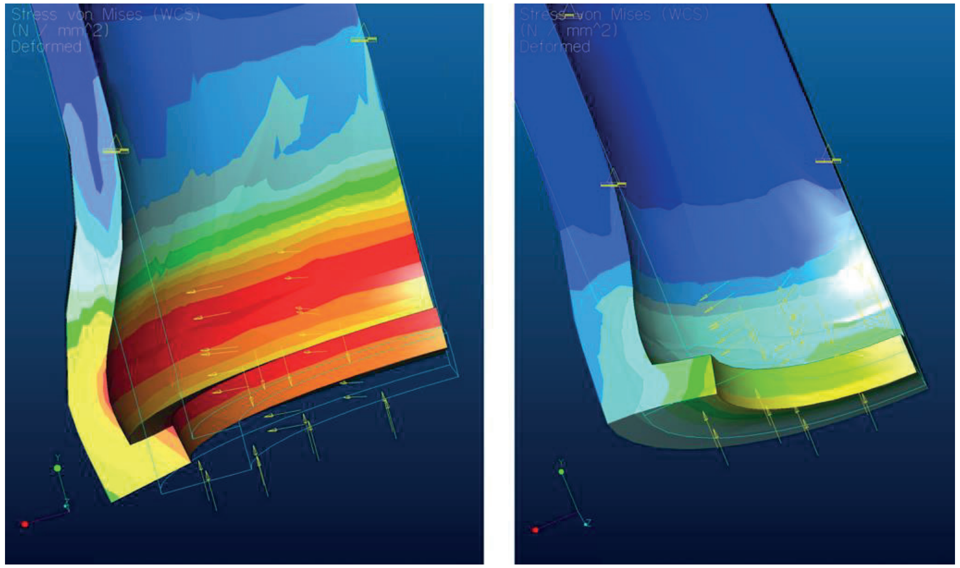

Fig. 8: Local deformation of two cup designs

The deformation of the cup changes with the position of the cup on the piston crown:

- The part of the cup cylinder which is pressurized depends on the position of the sealing line between the piston crown and the cup. The pressurized area of the cup wall is therefore variable;
- The inner shoulder of the cup base increases the stiffness of the cup. Therefore, the cup expansion is smaller close to the TDC-position.

By means of the FEM-analysis, a number of piston and cup designs have been evaluated for a 24 cc pump, being operated at a maximum pressure of 500 bar. Figures 9, 10 and 11 show the axial deformation of the inner diameter of the cup base shoulder, the radial expansion of the cup at the sealing line, and the maximum Von Mises stress. In each diagram, the final new design is compared to the original design having a wall thickness b of the cups of 2.25 mm, at various positions of the piston crown inside the cup.

The FEM-analysis clearly shows the larger expansion and the deformation of the thin walled cups. The mechanical stress is also increased. The axial deformation of the cup shoulder is somewhat less, especially between the BDC-position and half of the piston stroke. In this part of the stroke, the axial deformation has a positive value, meaning that the shoulder is bent away from the cup base, thus resulting in a divergent gap profile.

The most important result is the radial expansion of the cup at the location of the sealing line with the piston crown. Obviously, the thin walled cup expands considerably more than the heavier original design. The maximum value is increased from 2.8 µm to 4.6 µm, which corresponds to an increase of 66%.

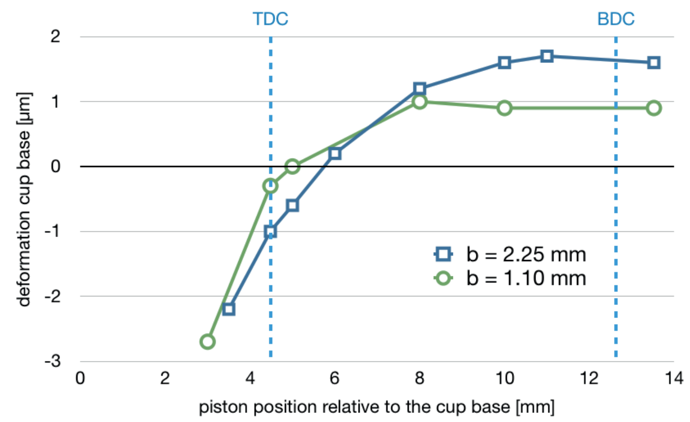

Fig. 9: Axial deformation of the cup base at p =500 bar (positive values result in a divergent gap profil

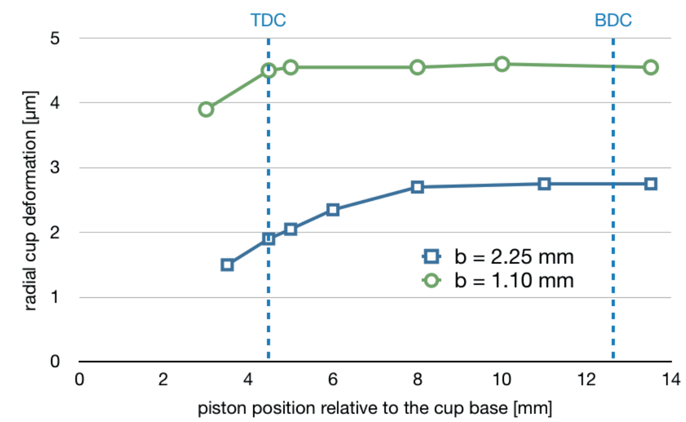

Fig. 10: Radial deformation of the cup cylinder at the sealing line with the piston (p = 500 bar)

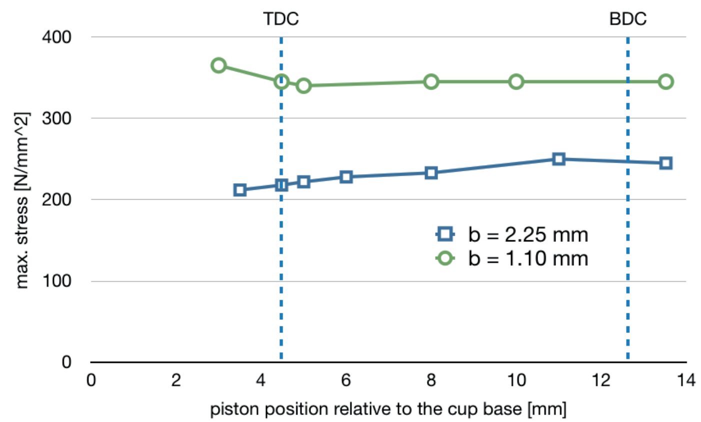

Fig. 11: Maximum value of the calculated Von Mises stress (p = 500 bar)

One of the advantages of the thin walled cups is that the radial expansion of the sealing line is almost constant during the entire stroke. In the original cup, the radial deformation varies between 1.9 and 2.75 µm. It is important to consider that the pressurized area of the piston crown is independent of the cup position. Furthermore, at any position of the cup, the piston expansion should be smaller than the cup expansion. Hence, the piston crown must be dimensioned as such that it matches the smallest radial expansion of the cup, i.e. the radial expansion at the TDC-position. Consequently, the gap between the expanded cup and the expanded piston crown widens when the cup moves to the BDC-position, since the cup expansion will become larger, whereas the piston deformation stays the same.

This effect cannot be seen with the new, thin walled cups, for which the radial expansion stays almost constant. This helps to reduce the gap height between the piston crown and the cup, but also creates the opportunity to shift the stroke closer to the cup base, thereby reducing the cup length. The shift will also improve the average position of the centre of mass of the cup, bringing it closer to the midpoint of the piston crown. This will further reduce the centrifugal tipping torque.

The reduction of the wall thickness will also create new opportunities for cost reduction, since the reduced wall thickness facilitates the use of mass production manufacturing technologies, such as deep drawing and stamping. The reduced tipping torque allows the pumps and motors to be operated at higher rotational speeds. This is important for many motor applications, but it also allows the application of high speed electric motors in electro-hydraulic actuators.

5 Test of the new cups and pistons

The original thick-walled cups and pistons have been replaced by the new cups and pistons having a smaller wall thickness. As before, the pump has been tested at minimum pump load in a speed range up till the point where the pump operation is no longer stable, or the maximum speed of the test bench has been reached. Unlike with the original cups, the pump operation remained stable. Due to the electric motor of the test bench, the pump could not be operated at higher rotational speeds than 5000 rpm. It is therefore not possible to verify whether, with the new cups and pistons, the pump can be operated up to the theoretical limit of 7600 rpm.

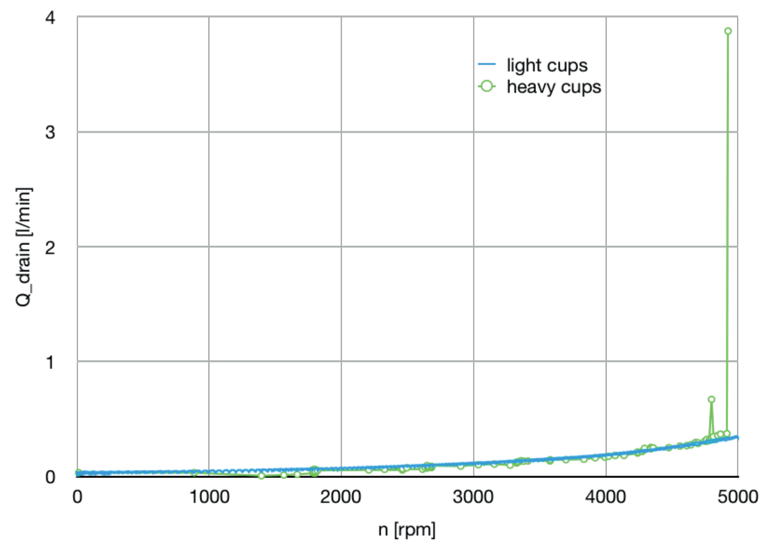

Fig. 12: Measured drain flow at minimum pump pressure

6 Conclusions and further outlook

This study investigates a subtle, but nevertheless important design change of the floating cup pumps and motors: the wall thickness of the cups has been reduced by about 50%, resulting in a weight reduction of 61%. Simultaneously, the wall thickness of the piston crown has been reduced as well. This was needed in order to maintain a tight sealing between the piston crown and the cup when being pressurized. The new cups and pistons have been successfully implemented in a 24 cc prototype. With the new cups and pistons, pump operation has been stable across the entire speed range up till 5000 rpm.

Aside from the improved high-speed stability, the new design offers a number of additional advantages:

- The new cup design has a more constant radial expansion, allowing a better match between the expansion of the piston crown and the expansion of the cup, even when getting close to the TDC;
- The new cup design reduces the outer diameter of the cups, thereby allowing a smaller pitch circle of the pistons i.e. a smaller design of the pump or motor;
- The reduced mass of the cups reduces the centrifugal tipping torque, thus allowing a higher rotational speed of the pump or motor;
- The new design allows a reduction of the wall thickness of the cup base, thereby reducing the length of the cup and thus the length of the entire machine;
- The reduced wall thickness reduces the width of the sealing land of the cup base, thereby allowing a larger diameter of the bore in the base of the cup, and thus a reduction of the flow losses;
- The reduced cup mass reduces the lateral centrifugal load of the cups. This reduces the friction between the cups and the pistons.

The reduced wall thickness also paves the way for further improvements of the cup and piston design. The aim is to produce these components by means of deep drawing, stamping and other low-cost, large series production and manufacturing technologies.

Nomenclature

Variable	*Description*	*Unit*
β	Swash angle	[°]
b	Wall thickness	[mm]
F_{centr}	Centrifugal force	[N]
F_{cont}	Contact force between cup and piston	[N]
$F_{fr,1}$	Friction force between cup and piston	[N]
$F_{fr,2}$	Friction force between cup and barrel plate	[N]
$F_{r,1}$	Reaction force between cup and barrel due to the hydrostatic balance	[N]
$F_{r,2}$	Reaction force between cup and barrel due to the cup tipping	[N]
F_{sp}	Force of the barrel spring	[N]
m_{cup}	Mass of the cup	[N]
n	Rotational speed	[1/minute]
n_{cr}	Critical rotational tipping speed	[1/minute]
p	Pressure	[bar]
T_{cr}	Critical rotational tipping torque	[Nm]

References

1. Manring, N.D., *Tipping the Cylinder Block of an axial-piston swash-plate type hydrostatic machine.* Transactions of the ASME, 2000. **122**: p. 216-221.
2. Manring, N.D., et al., *Scaling the Speed Limitations for Axial-Piston Swash-Plate Type Hydrostatic Machines.* Journal of Dynamic Systems, Measurement, and Control, 2014. **136**(3): p. 031004.
3. Achten, P. and S. Eggenkamp, *Barrel tipping in axial piston pumps and motors.* Proc. 15th Scandinavion International Conference on Fluid Power SICFP17, June 7-9, 2017, 2017.
4. Achten, P.A.J., *Power density of the floating cup axial piston principle.* Proc, of IMECE2004, November 13-19, 2004, Anaheim, California, USA, IMECE2004-59006, 2004.