MOVEMENT OF THE CUPS ON THE BARREL PLATE OF A FLOATING CUP, AXIAL PISTON MACHINE

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Abstract

In a floating cup axial piston machine, each piston has its own cuplike cylinder, floating on a barrel plate. On average the cups and the barrel rotate at the same rotational speed. A closer look at the kinematics of the floating cup principle however reveals that the cups make a small movement on the barrel plate. The size of this cup trajectory is strongly dependent on the tilt angle between the barrel and the rotor. Furthermore, the non-uniformity of the joint between the barrel and the rotor shaft can create an angular difference between the cup and the barrel position. This article will focus on the combined effect of the barrel tilt angle and the non-uniformity on the cup movement.

Keywords: axial piston machine, floating cup principle, construction, cup movement

1 Introduction

Axial piston pumps and motors are considered to be the most efficient hydrostatic machines available on the market. They can be applied in a wide range of pressures and flows. Most commonly used is the in-line, slipper type construction. Although the efficiency of in-line machines is somewhat lower than that of the bent axis design, the in-line machines are more cost effective. Furthermore, the in-line principle allows for a through drive, contrary to the bent axis design.

The most important disadvantages of axial piston machines (as well as of many other hydrostatic pumps and motors) are the high noise levels and the pressure pulsations. In addition there is a need for cost reduction. To address these issues a new axial piston principle called ”floating cup” has recently been introduced (Achten, 2002, 2003a, 2003b; Van den Oever, 2002). The new principle features a large number of pistons, about three times as much as in current pumps and motors. The pistons are arranged on two sides of a single rotor in a mirrored back-to-back configuration. The high number of pistons strongly reduces the pressure pulsations in the connecting lines, which has a positive effect on fluid borne noise as well as on the durability of fittings and lines. In addition the hydrostatically balanced construction lowers the bearing load thereby reducing the structure borne noise of the machine.

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In the floating cup concept, each piston has its own separate cylinder. These cuplike cylinders are hydrostatically balanced, floating on a barrel plate. One advantage of this construction is that the tolerance chain is limited to each piston-cup-combination. The design of these components is fit for high volume, low cost mass production techniques.

The kinematic principle of the floating cup machine results in a small shuffle movement of the cups on the barrel plate. This article will focus on the characteristics of this movement. The floating cup principle is suitable for application in pumps, motors, and hydraulic transformers. However, for reasons of simplicity, in this article only a pump will be described.

2 The Floating Cup Principle

The main functions of a hydrostatic pump are:

- displacement of oil volumes
- commutation from one pressure level to another.

In axial piston pumps the commutation is realized by means of a combination of a rotating barrel and a stationary port plate. The two parts act together as a spool valve, with the barrel as the moving part of the valve. The same principle is applied in the floating cup pump.

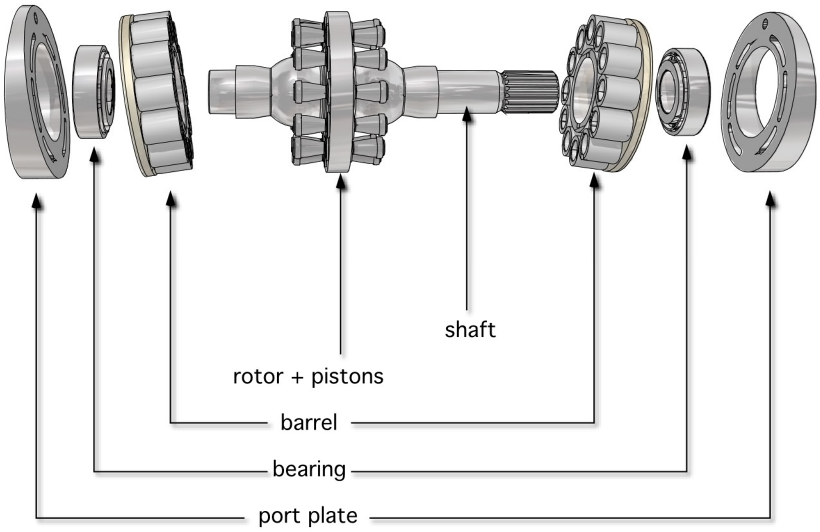

Fig. 1: *Exploded view of the main parts of the rotating group of the floating cup pump*

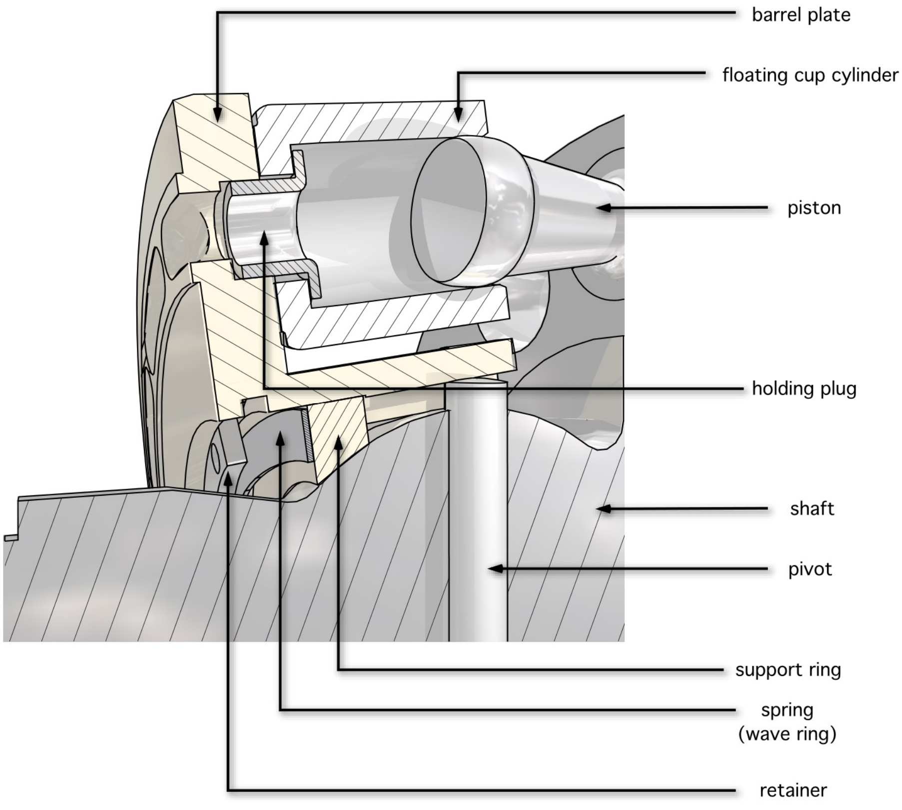

Fig. 2: *Detailed cross section of the barrel assembly*

The displacement in current axial piston machines is created by means of the translating movement of the pistons. The cylinders are an integral part of the barrel and cannot make a movement in the axial direction. This however is different in the floating cup principle. In a floating cup machine the pistons are locked onto a rotor and the pistons are only making a circular movement while the rotor is being rotated (see Fig. 1 and 2); there is no movement of the pistons in the axial direction.

Instead the cylinders move axially, thereby creating the necessary displacement of the machine. Each piston has a separate cuplike cylinder, with a connecting bore in the bottom of the cylinder (see Fig. 2). A barrel plate supports these ‘floating cups’. The configuration avoids the cups having to run on a stationary plane, as has been patented earlier by General Motors (Clarke, 1972).

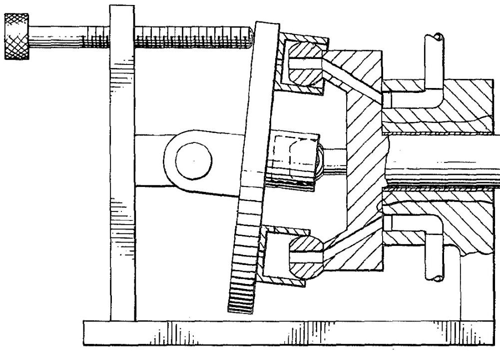

Fig. 3: *Pump design from William W. Clarke (US-patent US 3,648,567)*

The solution from Clarke has two major disadvantages:

- The friction between the moving cylinders and the stationary swash plate will tend to swing the cylinders away from the swash plate.
- The swash plate cannot act as a port plate due to the openings between the cylinders, and the commutation must be realized through the pistons.

In the floating cup principle these disadvantages are avoided. The cups and the barrel plate are moving at approximately the same speed, and therefore the (viscous) friction between the cup and the barrel plate is very low. The leakage gap between the cup and the barrel plate guarantees a good lubrication between the cup and the barrel plate. As a result the friction force acting on the sealing plane of the cup will be very low. The use of a barrel plate on the other hand also creates the possibility to seal off the leakage between the barrel ports. The commutation is realized in the same way as in conventional axial piston machines.

The barrel is also balanced in a similar manner as in slipper type and bent axis machines. The hydrostatic forces acting on the barrel are largely balanced by means of careful dimensioning of the seal lands between the barrel and the port plate. For operation at low pump pressures, a spring keeps the barrel pushed to the port plate. The construction, which is applied in the first prototype of the floating cup pump, can be seen in detail in Fig. 2.

The cups are hydrostatically balanced, in the axial direction as well as in the radial direction. Piston rings can be used to create a sealing between the pistons and the cup cylinders. It is also possible to seal directly on the piston head, thereby eliminating the piston rings (Achten, 2003a). In both cases a ball shaped object seals the gap between the piston and the cylinder. This will create a sealing line, which always stands perpendicular on the cylinder axis. Because of this the radial pressure load on the cup is equal in all directions and the cup is completely balanced.

In the axial direction the cup can be balanced in a similar way as the barrel. The dimensions of the sealing ring will determine the hydrostatic force that will push the cup to the barrel plate. As with the barrel there must be a retaining device to keep the cups to the surface of the barrel plate in case the internal oil pressure is not sufficient, for example during the suction stroke. Several design solutions can be conceived to fulfil this requirement. Figure 4 shows one solution, having a separate holding plug for each cup. The holding plug is attached to the barrel plate. The distance between the plug and the barrel plate leaves just enough room for the cup to move on the barrel plate.

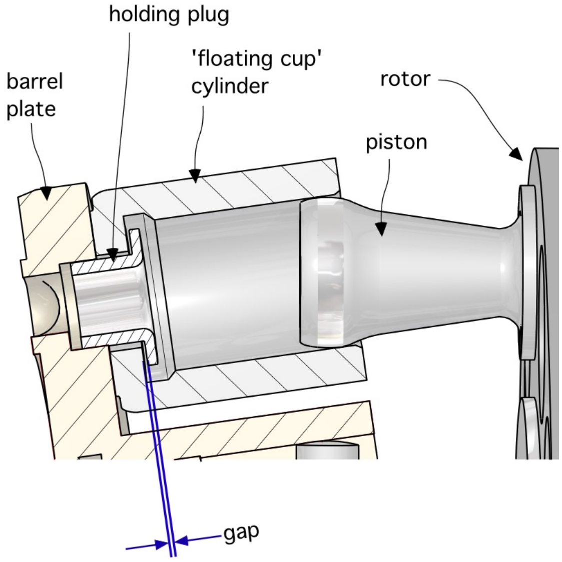

Fig. 4: *Holding plug*

The cups cannot be locked onto the barrel plate. The piston governs the movement of the cup and due to the tilt angle of the barrel, the cups will make an elliptic movement on the plane of the tilted barrel plate. Furthermore the barrel is synchronized with the rotor movement by means of a sliding joint. Figure 2 shows a cross section of one of the barrel slots together with the synchronizing pin or pivot.

Finally the free play of the cups on the barrel plane reduces for most components the need for close tolerances. The tolerance chain is limited to the piston-cup-combination. By means of selection, the cups and pistons can be matched, similar to the production of hydraulic tappets for valve trains in internal combustion engines (Achten, 2003a).

It is essential that the cups are almost completely hydrostatically balanced, and the forces acting on the cup should be as small as possible. Key to the design of the floating cup principle is that the conversion from hydraulic pressure to shaft torque (and vice versa) is realized directly in the contact between piston and oil (Achten, 2003b). Unlike for instance slipper type pumps, the pressure forces are not transmitted through the contact between piston and cylinder. The only remaining forces acting on the cup are:

- internal pressure forces
- friction forces
- inertia forces

The internal pressure forces are completely balanced in the radial direction (Achten, 2002; Achten, 2003b). Consequently the cup is not pushed in a radial direction to the piston, and the contact force between piston and cup is limited to friction and inertia forces. In the axial direction the pressure forces are slightly unbalanced to create a small force that keeps the cup pushed to the barrel plate. Since the relative velocity between the cup and the barrel plate is very low, the resulting viscous friction force between the cup and the barrel is also very small. Due to the small dimensions of the cup, the inertia forces are also relatively small compared to bent axis and slipper type machines.

The only possibility to create a large force on the cup is when the holding plug restrains the cup movement. In that case the movement of the piston-cup-combination will be transferred to the barrel via the holding plug. This will not only be in conflict with the barrel movement, which is governed by the pivots, but could also conflict with the movement of other piston-cup-combinations, which are similarly constrained.

For that reason it is essential for the design of the floating cup principle to have information about the amount of freedom the cups need to have in their movement on the barrel plate.

3 Position of the Cup on the Barrel Plate

The position of the cup is first of all determined by the position of the piston i.e. the ball shaped crown of the piston. The centre of the crown ball is taken as the zero-point for defining the cup coordinates. If the piston seals directly on the piston crown the piston position will determine the *x*- and *y*-position of the cup (see Fig. 5).

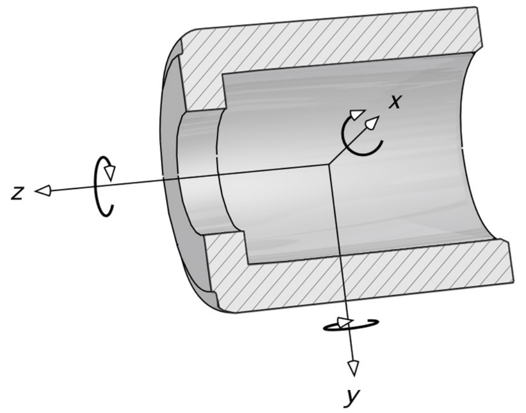

Fig. 5: *Cup coordinates*

If pressurized the floating cup is also pushed to the surface of the barrel plate. This will govern the *z*-position of the cup as well as the rotative position of the cup around the *x*- and *y*-axis. Of the original six degrees of freedom of the cup position, only one -the rotation position of the cup around its own axis (the *z*-axis)- is left undetermined. There is however no need to control this position in the machine and therefore the cup will be left free to rotate around its own axis.

3.1 Relative Cup Movement in case of a Constant Velocity Joint between Rotor and Barrel

This article addresses the determination of the *x*- and *y*-position of the cups on the barrel plane, assuming the cups are pushed to the barrel plate and the cup movement is not hindered or obstructed by the holding plug. Furthermore, it is of importance that the rotational position of the barrel does not need to be exactly equal to the rotational position of the rotor. Nevertheless in this paragraph a constant-velocity joint between the rotor and the barrel will first be assumed.

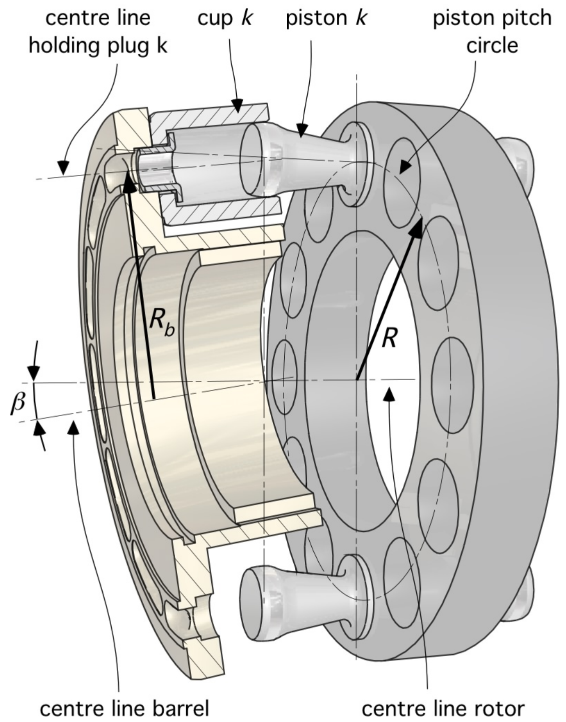

Fig. 6: *Nomenclature floating cup parts*

The pistons are locked onto the rotor, on a piston pitch circle with radius R. A vector $\mathbf{C}$ describes the piston position on the piston pitch circle for a rotational position of the piston (i.e. the rotor) of φ degrees:

$$\mathbf{C} = \left(R\cos(\varphi),\, R\sin(\varphi)\right) \tag{1}$$

If the barrel would be rotating around the same axis as the rotor, the position of the cup in the *x*-*y*-plane would be the same as the piston position. However, at a tilted position of the barrel of β degrees, the circular movement of the pistons will be projected on the tilted barrel plane. The position $\mathbf{C}'$ of the cup on this plane can be defined as:

$$\mathbf{C}' = \left(R\cos(\varphi),\, R\cos(\beta)\sin(\varphi)\right) \tag{2}$$

Figure 7 shows the cup positions and trajectories for a total number of 12 cups on each barrel and a tilt angle between the rotor and the barrel axis of $\beta = 40°$.

Assuming that a pure constant velocity joint drives the barrel, the relative movement of the cups on the barrel plate follows a circular trajectory. The centre point of each circular path is positioned on a pitch circle having a radius R_b:

$$R_b = \tfrac{1}{2} R\left(1+\cos[\beta]\right) \tag{3}$$

The circular trajectory itself has a radius ρ:

$$\rho = \tfrac{1}{2} R\left(1-\cos[\beta]\right) \tag{4}$$

Walzer (1984) has previously derived this equation for other axial piston machines and named the circular trajectory the "ρ-circle". The centre point $\mathbf{M}$ of each ρ-circle has the following coordinates:

$$\mathbf{M} = \left(R_b \cos(\varphi),\, R_b \sin(\varphi)\right) \tag{5}$$

In order to minimize the risk of a conflict between the movement of the cups on the one hand and the position and size of the holding plug (or other retaining device) on the other, the centre line of the holding plug must preferably go through **M**.

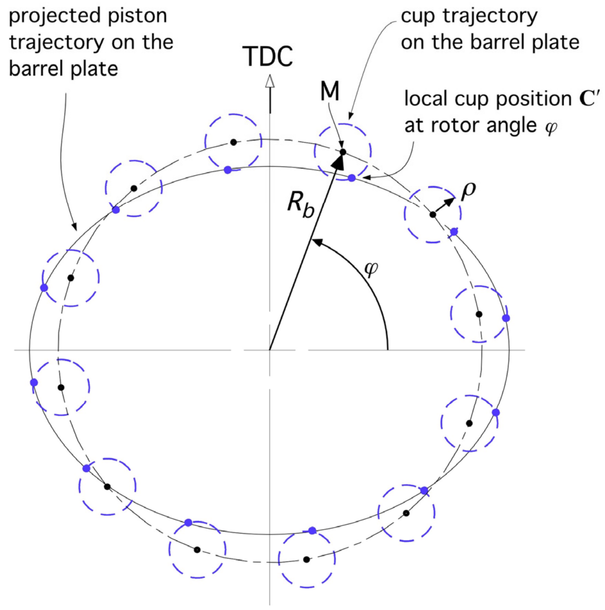

Fig. 7: *Relative movement of the cups on the barrel plate. In this example the tilt angle between the rotor and the barrel is set at a value of β=40°. TDC marks the point at which the pistons are in the top dead centre position i.e. where the cylinder volume is minimum*

The position **C′** of the cup on the barrel plane can be described in polar coordinates, having a radius R' and an angle φ':

$$R' = \sqrt{\left(R\cos[\varphi]\right)^2 + \left(R\cos[\beta]\sin[\varphi]\right)^2} \tag{6}$$

$$\varphi' = \arctan\left(\cos[\beta]\cdot\tan[\varphi]\right) \tag{7}$$

3.2 Pivot Joint

The only functions of the barrel are to support the cups and to allow for commutation in combination with the port plate. Since the cups are floating almost without any friction on the barrel plate (Achten, 2002) they cannot create a torque on the barrel. Consequently the hydraulic pressure does not create a torque load on the barrel. Instead the pressure forces directly act on the pistons where they create a torque on the rotor. As a result the torque load on the barrel is limited to some friction forces between the barrel and the port plate and the inertia load when accelerating and decelerating.

This creates the possibility to use a rather simple joint for driving the barrel. In the current design of the floating cup pump (Fig. 2) a kind of sliding joint is applied. Aside from the simplicity, the construction has the advantage that the barrels are free to move in the axial direction.

This pivot joint is not a pure constant velocity joint. It has the same non-uniformity as a Cardan or Hook´s joint (Schmelz, 1992). Given an angular position of the rotor φ and a tilt angle β between the rotor and the barrel axis, the rotational position ψ of the barrel is equal to:

$$\psi = \arctan\left(\cos[\beta]\cdot\tan[\varphi]\right) \tag{8}$$

The non-uniformity δ is equal to:

$$\delta = \psi - \varphi \tag{9}$$

Figure 8 shows δ as a function of the rotor angle φ for three different values of the barrel tilt angle β:

- β = 40°; this is a common value for constant displacement bent axis pumps and motors
- β = 20°; this is a common value for slipper type pumps and motors
- β = 10°; this is about the maximum value for the floating cup principle

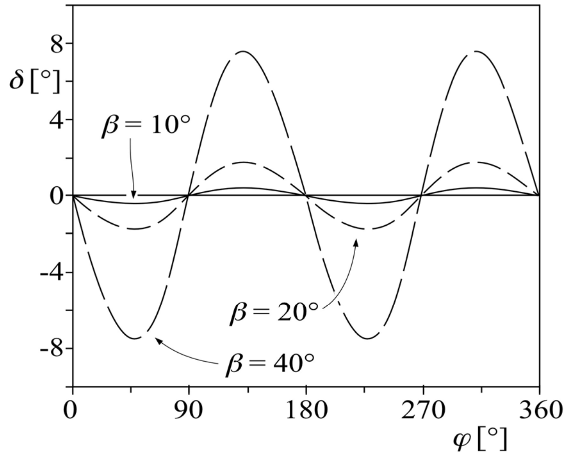

Fig. 8: *Angular difference between the rotor and barrel positions in case a sliding joint is applied to drive the barrel*

At a tilt angle of 40° (as in a bent axis machine) the angular difference with a sliding joint can be as large as 7.6°. In a floating cup machine having 2x12 pistons, the maximum tilt angle between the barrel and the rotor cannot be more than 10°. Consequently, the angular difference is much smaller and is limited to a maximum value of 0.4°.

3.3 Relative Cup Movement in case of a Pivot Joint

Although the non-uniformity of the pivot joint is rather small, it has a significant effect on the relative movement between the cups and the barrel. Since the pistons are locked onto the rotor, the x- and y-position of the cup is directly determined by the rotor position. An angular difference between the rotor and barrel position results therefore in an angular difference between the cup and the barrel position. As can be concluded from Eq. 7 and 8 the angular difference is of exactly the same magnitude:

$$\varphi' = \psi = \arctan\left(\cos[\beta]\cdot\tan[\varphi]\right) \tag{10}$$

The combined effect of the two angular differences is however dependent on the position of the cup-piston-combination on the rotor i.e. relative to the position of the pivot. Therefore the specific cup position has to be defined by introducing the angle φ_k of cup k at a rotor position of φ:

$$\varphi_k = \varphi + \frac{2\pi}{z_p} \cdot (k-1) \qquad (k = 1 \ldots z_p) \tag{11}$$

The counting of the cup numbers starts at a cup, which is positioned on the centre line of the pivot (Fig. 9).

Because the pivot joint only creates an angular difference between the barrel and the rotor it does not have an influence on the radial position of the cup on the barrel plate. The radial position of the cup is therefore still governed by Eq. 6, which can now be derived for cup k:

$$R'_k = \sqrt{\left(R\cos[\varphi_k]\right)^2 + \left(R\cos[\beta]\sin[\varphi_k]\right)^2} \tag{12}$$

Also the angular position of cup k with respect to the housing is not influenced by the non-uniformity of the pivot joint:

$$\varphi'_k = \arctan\left(\cos[\beta] \cdot \tan[\varphi_k]\right) \tag{13}$$

But with respect to the barrel position the non-uniformity does have an influence. This results in the following relative position of cup k on the barrel plate:

$$\mathbf{C'_k} = \left(R'_k \cos(\varphi'_k - \delta),\, R'_k \sin(\varphi'_k - \delta)\right) \tag{14}$$

The resulting cup trajectories are presented in Fig. 9, calculated for 12 pistons and a tilt angle $\beta = 40°$.

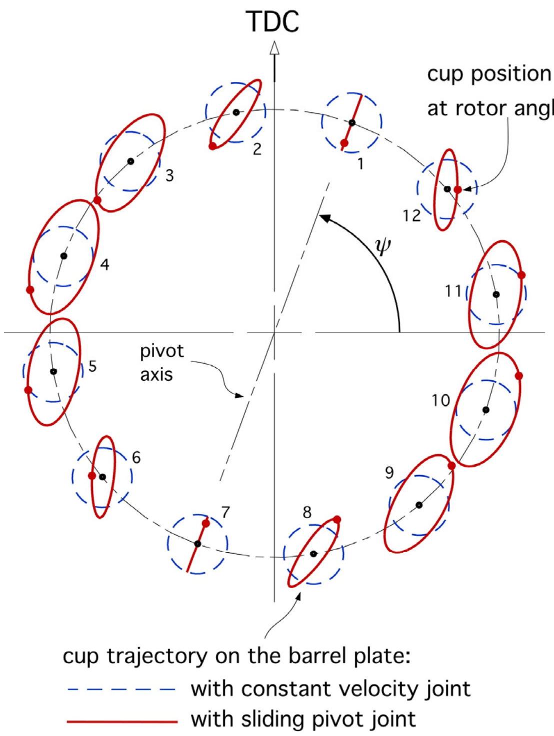

Fig. 9: *Relative movement of the cups on the barrel plate in case the barrel is driven by a pivot joint or some other kind of Cardan joint. In this example the number of pistons z_p = 12 and the tilt angle between the rotor and the barrel is set at a value of β =40°*

For most cups the original ρ-circle is now deformed to some kind of oval path. But for the cups, which are positioned precisely in line with the centre line of the pivots (i.e. cups 1 and 7), the relative cup movement on the barrel plate is only on a straight line in the radial direction. For these cups the angular deviation created by the tilt angle of the barrel is exactly counteracted by the angular deviation created by the pivot joint.

For cups 4 and 10 however the angular difference created by the tilted position of the barrel is added to the non-uniformity created by the pivot joint. These cups make the largest cup trajectory, having a width of 4ρ.

4 Practical Implications

The prototype of the floating cup pump (Achten, 2003a, 2003b) is a 28 cc/rev constant displacement pump having 12 pistons on each side of the rotor. The radius of the piston pitch circle R is 36 mm. If this pump would have a barrel tilt angle of 40° the cups would have a maximum relative movement on the barrel plate between 8.4 mm (cups 1 and 7 in Fig. 9) and 16.8 mm (cups 4 and 10). Figure 10 shows how this would affect the position of the cups on the barrel plate.

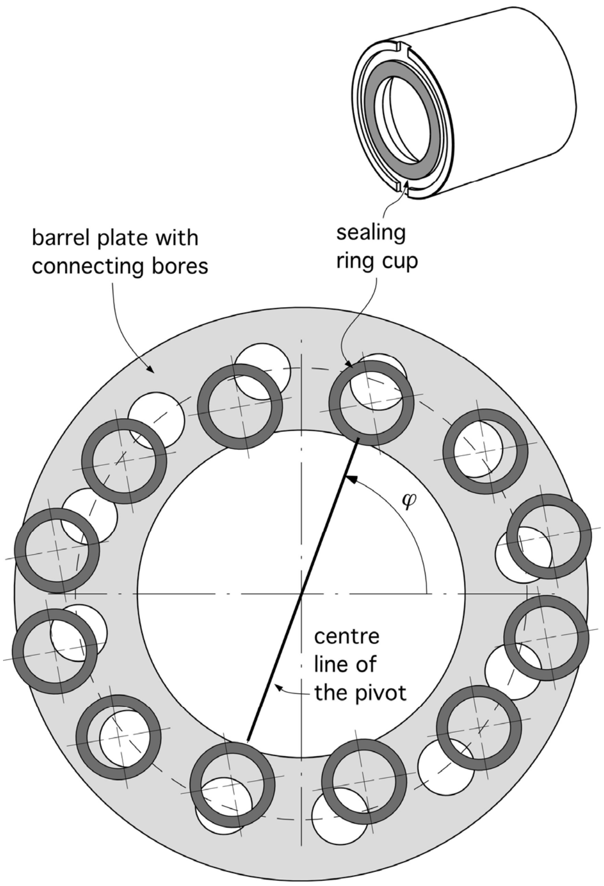

Fig. 10: *Position of the sealing ring of the cups in case of a pivot joint and a barrel tilt angle of β = 40°*

In this situation the cups would no longer seal the contact between the cups and the barrel plate and severe leakage would occur. It is also obvious that, due to the outsized movement of the cups, the construction with the holding plugs cannot be applied.

The relative cup movement on the barrel plate is however strongly dependent on the barrel tilt angle β. Figure 11 shows the relationship between the maximum cup displacement (equal to 4ρ) and the barrel tilt angle. In a floating cup pump with 2x12 pistons the tilt angle is limited to about 10° due to geometrical constraints. One of these constraints is the diameter of the piston neck. A large barrel tilt angle forces the pistons to be more tapered. In the end the piston neck will be too small to withstand the hydraulic load from the piston crown.

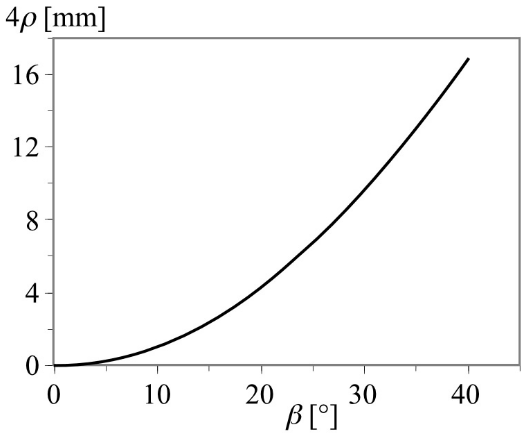

Fig. 11: *Maximum displacement of the cups on the barrel plate as a function of the barrel tilt angle β, for a floating cup pump having a radius of the piston pitch circle of 36 mm*

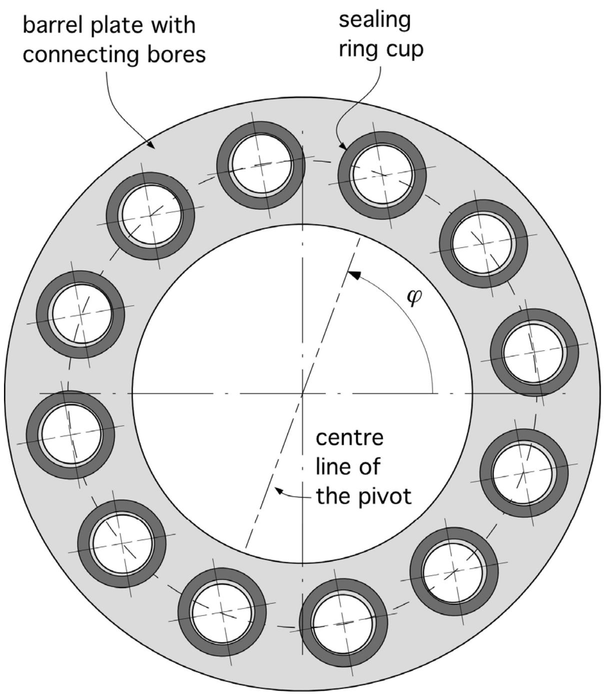

Fig. 12: *Position of the sealing ring of the cups in case of a pivot joint and a barrel tilt angle of β = 8°*

By reducing the barrel tilt angle from 40° to 10° the maximum cup displacement is reduced from 16.8 mm to 1.1 mm. In the prototype of the floating cup pump, having a barrel tilt angle of 8°, the maximum cup displacement is even reduced to 0.7 mm. Figure 12 shows the cup positions for the prototype configuration at a barrel tilt angle of 8°. Now the sealing rings of the various cups are in full contact with the barrel plate.

Nevertheless, the application of plugs holding the cups in the axial direction results in another design constraint. The holding plug restricts the *x*- and *y*-movement of the cup on the barrel plate. The combined dimensions of cup and holding plug must be chosen as such that the cup movement –imposed by the piston– is not hindered. Whenever the piston will move the cup beyond the radial gap between cup and holding plug the cup will be pushed in a tilted position.

Furthermore, it should be noticed that *z*-position of the cup on the barrel plate is only determined in combination with the axial pressure load, pushing the cups to the barrel plate. In case of a low pressure, the resultant axial pressure force could be too small to keep the cup pushed to the barrel plate. This situation for instance occurs if the cup is in contact with the low-pressure port or suction port. Due to the centrifugal forces the cup will move away from the barrel plate. Depending on the sizes of the cup and the holding plug, as well as on the gap between cup and holding plug, it could be possible that the cup slips beyond the edge of the holding cup flange (Fig. 13).

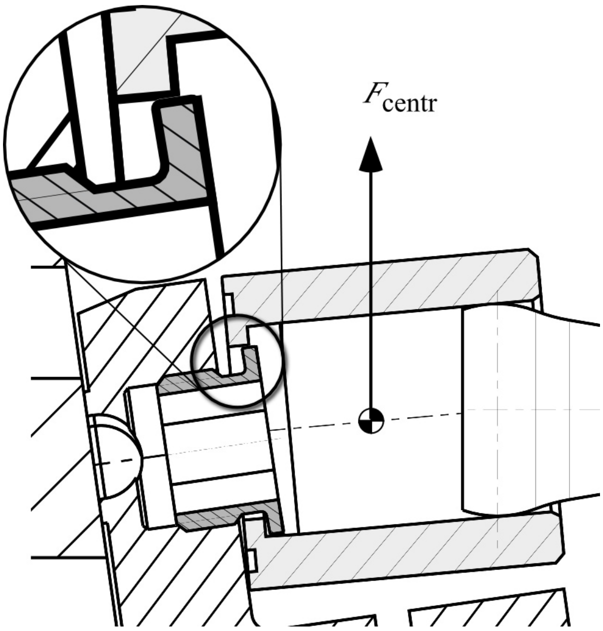

Fig. 13: *Tilted cup position in which the connecting bore of the cup has slipped beyond the flange of the holding plug*

In a situation like this the holding plug will still retain the cup (after all the diameter of the cup flange is larger than the diameter of the connecting bore in the bottom of the cup). But being in this position it is likely that the cup will become locked. If this condition occurs the cup is not able to move back to its sealed state on the barrel plate. The tilted cup position will increase the leakage through the gap between the cup and the barrel plate. But more important it will hinder the necessary movement of the cup on the barrel plate. Instead the combination of piston, cup and holding plug will create a new kinematic system, which will define a new barrel position.

But this is not the only kinematic system trying to define the angular position of the barrel. First of all there is the sliding pivot joint. And of course the locked cup position does not need to be limited to a single cup but could well occur with several other cups.

The conflicting demands on the barrel position will be solved by an elastic deformation of some of the components. This will create extra loads on all components involved. Aside from effects such as wear and efficiency loss, the extra load on the barrel will significantly influence the axial balance of the barrel. As a result the barrel could easily lift off the port plate. The conflict between the various kinematic systems will not be constant but will vary depending on the angular position of the barrel. Because of this the forces and torque loads acting on the barrel will vary as well and the barrel balance will become unstable. As a result the barrel could be pushed to the port plate in one position of the barrel and be lifted off in another. The variation in the gap height between the barrel and the port plate results in a variation of the leakage flow through this gap, which could well influence the flow pulsations and noise level of the machine.

Applying a homokinetic joint between the barrel and the rotor shaft can reduce the relative movement of the cups, similar to the idea proposed by Wagenseil (1986) for use in bent axis pumps. But these joints are rather complicated and will certainly not be the most cost effective solution. As a compromise it should be possible to apply a tripod sliding joint with three pivots in 3x120°-configuration. This would strongly reduce the angular difference between the barrel and the rotor while still having a cost effective and simple construction.

5 Conclusions

In floating cup machines the position of the cup on the barrel plate is defined by the piston. The tilted position of the barrel and the non-uniformity of the joint between barrel and rotor result in a movement of the cups on the plane of the barrel plate. A retaining device (or any other obstacle) on the barrel, or the barrel plate, must not hinder this movement. The magnitude of the movement is strongly dependent on the tilt angle between the centre line of the barrel and the centre line of the rotor shaft. Furthermore, the freedom for the cups to move on the barrel plate should be large enough to allow for deviations in production tolerances.

These conditions result in a number of clear dimensional demands for the retaining device that has to keep the cup attached to the barrel plate at low-pressure conditions. The magnitude of the cup movement on the barrel plate is not equal for all cups but is dependent on the angular position of the cup with respect to the angular position of the pivot of the sliding joint between the barrel and the rotor shaft.

The relative cup movement also sets a demand for the maximum tilt angle between the barrel and the rotor shaft. There is not a general rule for the maximum tilt angle since this will be influenced strongly by the number of pistons. For the current prototype having 2x12 pistons the tilt angle is limited to approximately 10°. If the tilt angle would be larger, the relative movement of the cup becomes so large that the sealing ring of the cups slides (for a part) over the connecting bore in the barrel plate. At a larger tilt angle the cup trajectory on the barrel plate will open up the connecting bore to the pump case and severe leakage will occur.

Nomenclature

β	tilt angle of the barrel	[°]
$\mathbf{C}$	piston position on the rotor	[mm]
$\mathbf{C'}$	cup position on the barrel	[mm]
$\mathbf{C'_k}$	Position of cup k on the barrel	[mm]
δ	angular difference between barrel and rotor	[°]
F_{centr}	centrifugal force	[N]
$\mathbf{M}$	position of the barrel port on the barrel	[mm]
R	Radius of the piston pitch circle	[mm]
R'	radial cup position on the barrel plate	[mm]
R'_k	radial position of cup k on the barrel	[mm]
ρ	Radius of the circular cup trajectory on the barrel	[mm]
φ	angular rotor position	[°]
φ'	angular cup position	[°]
φ'_k	angular position of cup k	[°]
ψ	angular barrel position	[°]
z_p	number of pistons per rotor side	[-]

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