DESIGN OF A VARIABLE DISPLACEMENT FLOATING CUP PUMP

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ABSTRACT

In floating cup pumps and motors, the pistons are arranged in a balanced, double ring design. Each piston has its own cuplike cylinder, floating on a barrel plate. The displacement is created by letting the barrels run on a tilted port plate, one on each site of the machine. Up till now, only constant displacement floating cup pumps and motors have been built and tested. This paper describes the design of a first (open circuit) variable displacement floating cup pump. The design is realized by applying the same pistons and cups as in the rotating group for the actuation and control of the tilt angles of the swash plates. The actuators are positioned as such that the deformation of the swash plates is reduced to a minimum. Furthermore the new design results in a very low contact force of the swash plates pushing against the cradle bearings, thereby resulting in a minimum swivel torque of the swash plates.

KEYWORDS: Axial piston pump, floating cup, open circuit, variable displacement.

1. INTRODUCTION

The floating cup is a new axial piston principle for hydrostatic pumps, motors and transformers. In the past three years, several prototypes have been built and tested [1…15]. The tests have revealed the high efficiency, the low pulsation levels and the low noise level of the new principle. The efficiency is even higher than of modern bent axis machines. This is primarily due to a strong reduction of the friction losses in the floating cup machine, which also results in a very high torque at startup and low speed conditions. Despite the relatively large number of leakage gaps, the volumetric efficiency of the floating cup pump is about equal to conventional slipper type and bent axis machines [15].

The floating cup principle is designed as such that many of its components can be produced by means of deep drawing, extrusion, sintering and other modern, non-machining production technologies. A cost study, which has been performed together with the industry, has proven the strong cost reduction potential of the floating cup principle. Further cost reductions are achievable in the hydraulic system, for instance because of reduced heat generation in the system which results in smaller cooler requirements.

All studies however have been limited to constant displacement machines. Although the market share of constant displacement pumps and motors is quite significant there is also a strong interest in variable displacement units, especially variable pumps. This paper describes a first design of a variable displacement floating cup pump for open circuits.

2. THE FLOATING CUP PRINCIPLE

The core of the floating cup principle is a combination of a simple piston and a small, cuplike cylinder (figure 1). The piston crown is a ball segment, which seals directly on the cylinder, without any piston rings. The sealing between the piston and the cylinder always stands perpendicular on the central axis of the cylinder. As a result the cylinder is balanced in all radial directions. Due to the internal oil pressure, the cylinder expands. To counteract this expansion, the piston head is made hollow which causes the piston head to expand as well. The diameter and the depth of this cavity are chosen as such that the gap height between the piston and the cylinder is more or less constant [15].

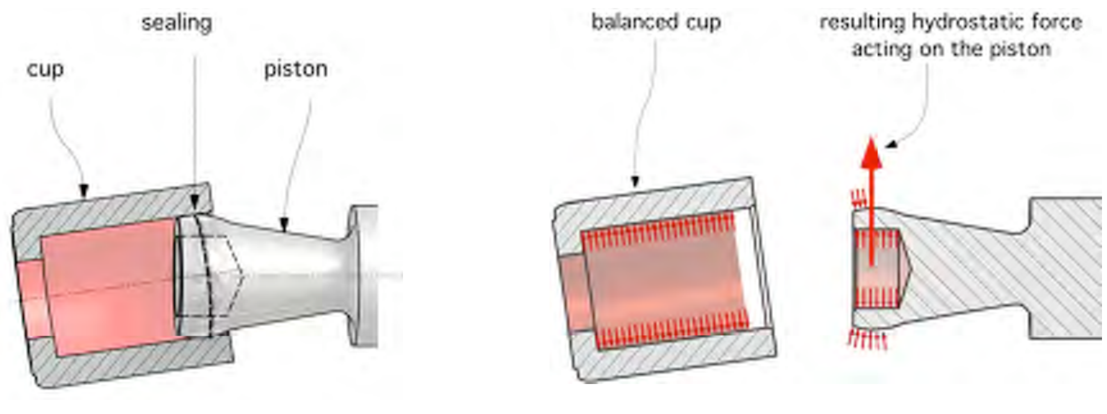

Figure 1: Piston and cup

Since the cup is balanced, it cannot transfer any hydrostatic force to the piston; its only function is to contain a pressurized oil column. Instead, the oil column itself presses directly against the piston, thereby creating a radial force on the piston. This radial component of the hydrostatic piston force, which is shown in the right picture of figure 1, creates the net resulting torque on the rotor, in which the piston is mounted. Consequently there is a direct conversion of the hydrostatic pressure forces acting on the piston to a net resulting rotor and shaft torque, without any moving linkages or joints.

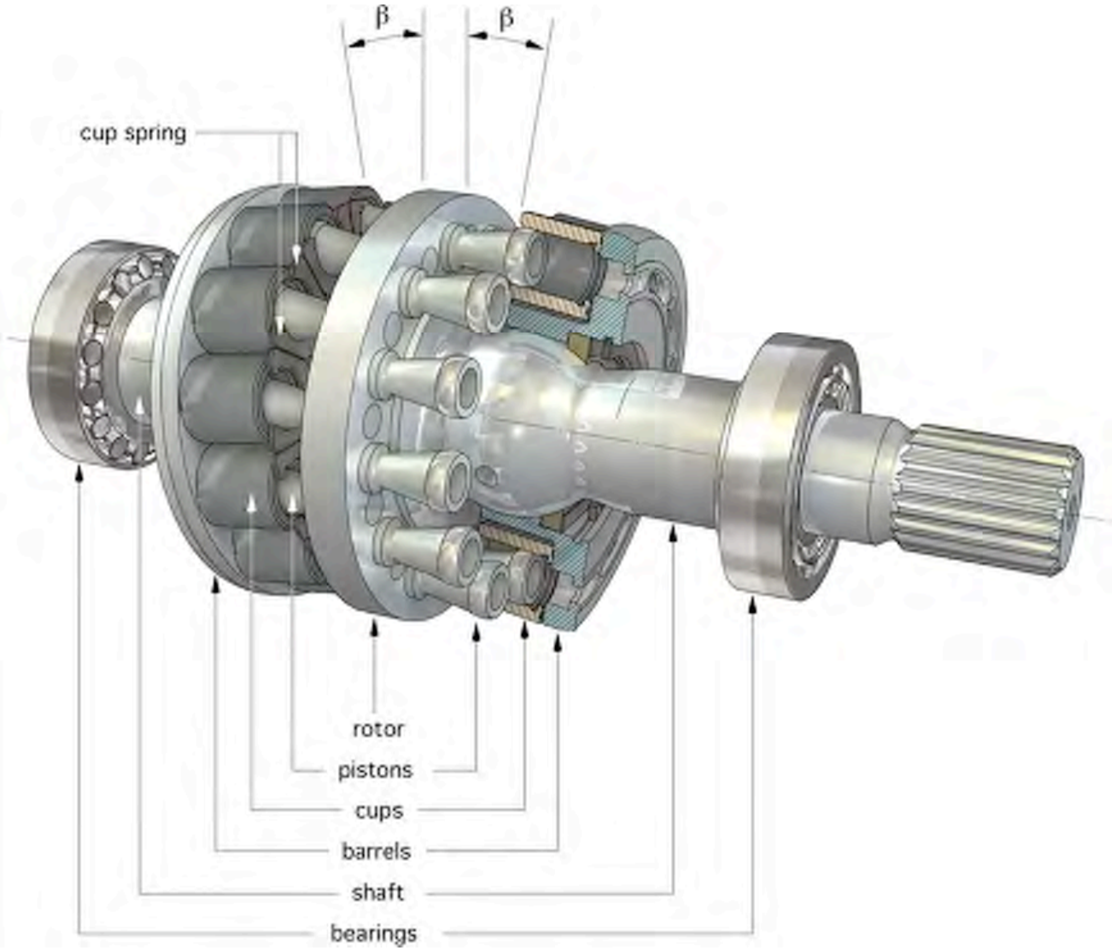

Figure 2.: Rotating parts of the floating cup principle showing a cutaway view of the right barrel.

The figure above shows an assembly of the rotating group of a floating cup machine. The pistons are arranged in a double ring, back-to-back configuration. This mirrored configuration makes the construction to a large extend balanced in the axial direction. In addition, the double ring concept also creates the possibility of having a maximum number of pistons without sacrificing the high power density of axial piston machines [12]. A barrel, one on each side of the machine, supports the cups. Letting the barrels rotate at a tilt angle β creates the positive displacement of the machine.

The high number of pistons is one of the main advantages of the floating cup principle. Figure 2 shows a construction having 2 times 12 pistons. Because the pistons on the left side of the rotor are positioned in between the piston positions on the right side, the total rotating group has a true 24-piston-behavior. The high number of pistons offers the opportunity of a fundamental reduction of pressure and flow pulsations [10]. Furthermore, the torque output is almost constant and remains high, also at starting conditions [11, 14]. The even number of pistons per barrel also results in an almost constant axial load of the barrel on the port plate i.e. swash plate. This improves the possibility to create a complete hydrostatic balance of the swash plate, thereby reducing the friction between the swash plate and its bearing.

The even number of pistons per barrel also has a disadvantage: the swivel torque on the swash plates is twice as high as in case of an odd number of pistons per barrel. This swivel torque is created because of the rotation of the barrel ports on the swash plate annex port plate. As a consequence the resulting barrel load force follows a trajectory on the port plate, which is shown in the figure below. This force creates a swivel torque on the swash plate around the axis of the swash plate bearing, similar to the swivel torque created by the pistons in a slipper type pump [16]. The torque variation creates a small back and forth rotation of the swash plate. The acceleration of this swiveling movement could be too high for the barrel to follow which would result in an increased gap between the barrel and the port plate and thus in high volumetric losses. In case of an even number of pistons per barrel, the frequency of the torque variation is half as large as with an odd number of pistons per barrel, but the amplitude is twice as large.

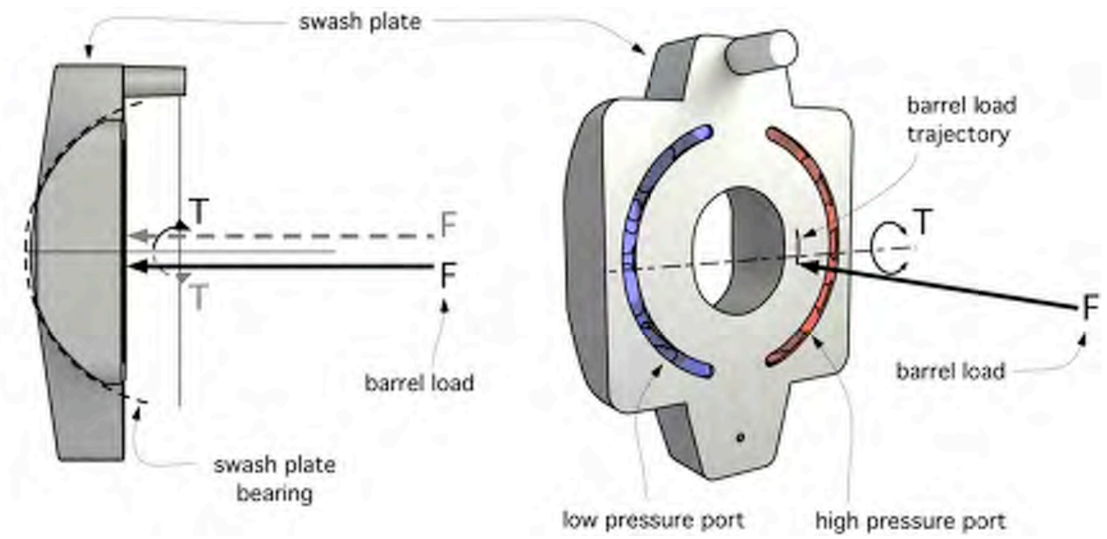

Figure 3: barrel load and swivel torque on the swash plate.

In the floating cup principle, the cups don't run on the stationary port plates but are floating on the barrels. The tilted position of the barrels will make the cups move on the barrel plate on a trajectory, which is not completely circular but slightly elliptical. The synchronization of the rotation of the barrels and the rotation of the shaft, rotor and pistons is realized by means of a small sliding joint (not shown in figure 2). This joint is not a pure uniform, homokinetic coupling. Because of the non-uniformity and the elliptic cup trajectory, the cups must be able to move on the barrel plate [9]. The magnitude of this relative displacement is rather small. For instance for a 28 cc unit having 24 pistons and a barrel tilt angle of 8°, the relative movement of the cups on the barrel plate has an amplitude of around 0,5 mm.

The degree of freedom for the cups also reduces the costs of the new pump concept since it breaks the tolerance chain between the individual pistons. The tolerance chain is now limited to each piston-cup-combination. These core components are designed as such that they can be produced by means of non-machining production technologies like deep drawing, pressing, metal injection molding (MIM) and fine blanking. Most other components of the floating cup principle are also designed as such that they can be produced by applying low cost high volume 'automotive' production technologies.

3. CONVENTIONAL VARIABLE DISPLACEMENT MACHINES

Figures 4 and 5 show two state-of-the art axial piston machines that can be operated in an open circuit: a slipper type or swash plate pump and a bent axis motor. Both machines cannot be operated over-centre. In these machines the control of the displacement volume is realized by means of a control piston and a bias piston, the latter being always loaded with the pressure from the high-pressure side of the machine.

In the slipper type pump the port plate is stationary; the control of the pump is realized on the opposite side, at the swash plate. This has the advantage that the connection between the ports of the port plate and the high and low-pressure ports of the housing is stationary. Due to the high load of the pistons the swash plate deforms. This deformation can be as high as 0,1 mm [17]. The combination of the pistons and the slippers however offers sufficient flexibility to follow these deformations, thereby eliminating the leakage that could occur because of the deformation.

Both the control and the bias piston are positioned parallel to the central axis of the pump. As a result these pistons also generate side forces that could push the swash plate out of its cradle bearing. To compensate for this effect the swash plate must be pushed harder into the bearing, which is realized by reducing the size of the pressure compensating areas at the back of the swash plate. In addition the control piston and the odd number of pistons will further increase the contact force of the swash plate, which increases the friction between the swash plate and its bearing

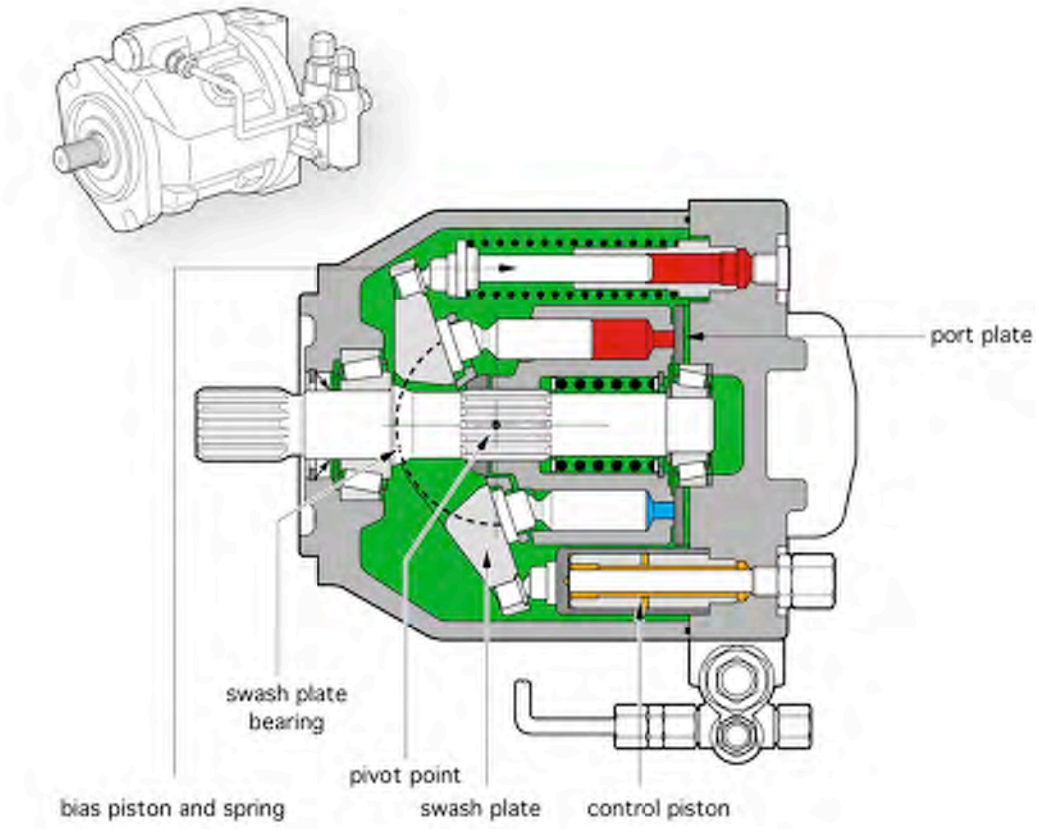

Figure 4: Variable displacement slipper type pump (courtesy Bosch Rexroth)

In the bent axis motor the displacement is varied by means of changing the tilt angle of the barrel and the port plate. Because there is no through shaft, the port plate is relatively small, stiff and well supported. This is important for limiting the deformation of the port plate, thereby avoiding excessive leakage between the barrel and the port plate.

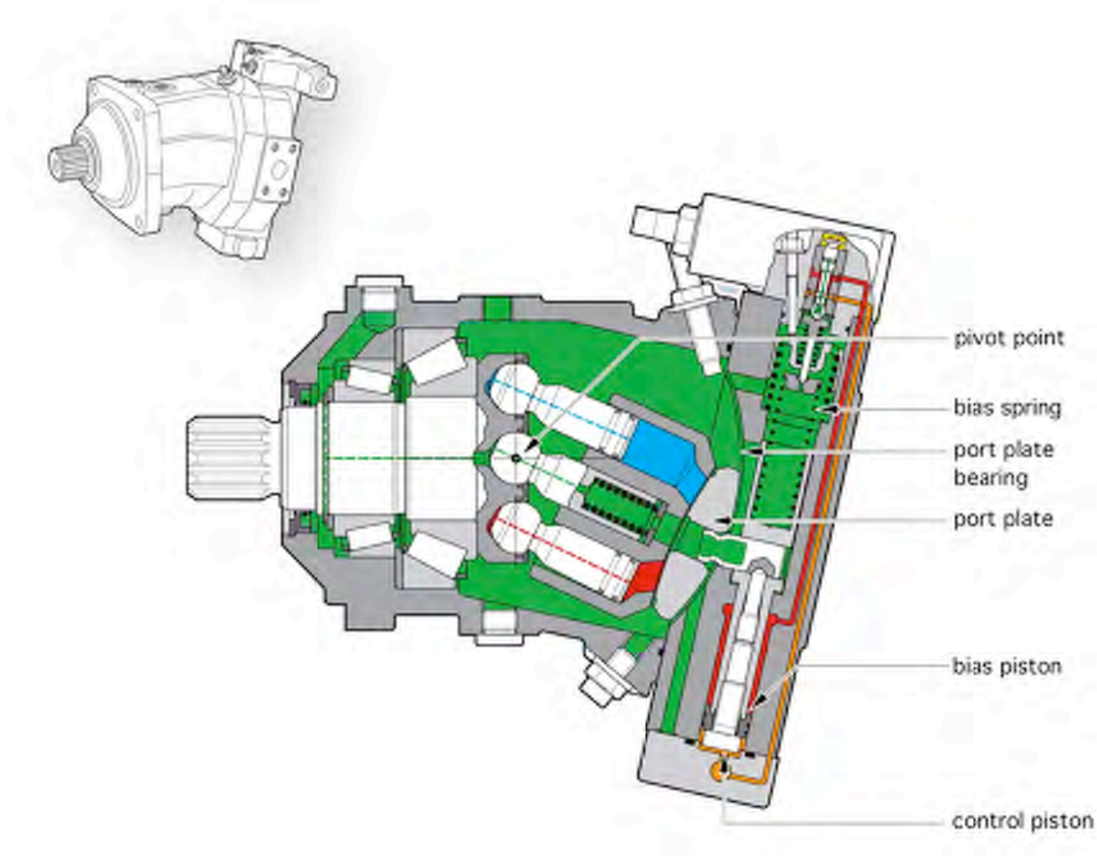

Figure 5: Variable displacement bent axis motor (courtesy Bosch Rexroth)

The high-pressure port in the housing must always remain covered by the port plate in order to avoid leakage to the housing. This restricts the range of the minimum and maximum tilt angles between which the displacement can be varied. Even than, the movement of the port plate is rather large, which is due to the relatively large distance between the line of rotation (indicted by the pivot point in figure 5) and the port plate. Consequently the actuator controlling the port plate and barrel position needs to make a large stroke, which results in a rather massive actuator.

4. THE VARIABLE DISPLACEMENT FLOATING CUP PUMP

The variable displacement floating cup machine (figure 6) has elements of both the slipper type and the bent axis design. Like in the slipper type machine, the barrel diameter and the corresponding size of the port plate annex swash plate is rather large. But in the floating cup design the swash plate has to be much stiffer in order to avoid large gaps between the port plate and the barrel. Similar to the bent axis machine, the port plate moves with the barrel, and the oil is supplied and delivered via the sliding interface between the swiveling port plate and the stationary housing. Due to the relatively small tilt angle of the barrel of 8°, the displacement of the swash plate is however much smaller than in the bent axis machine.

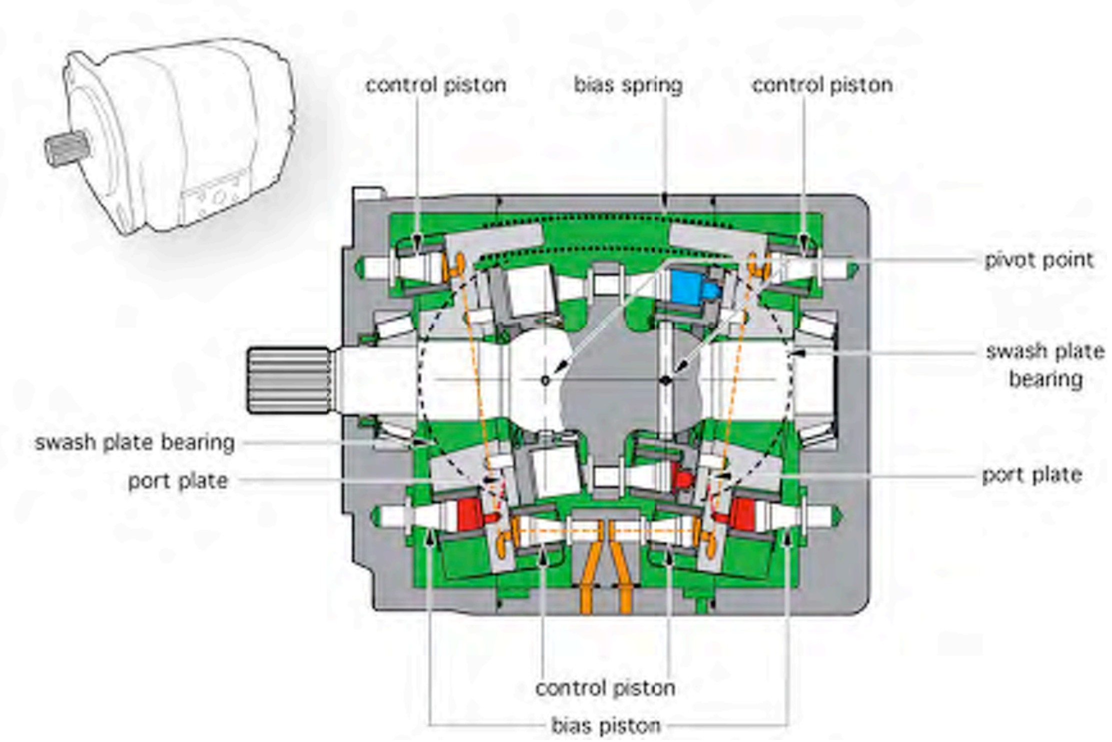

Figure 6: Variable displacement floating cup pump

Distinctive for the floating cup design is the double swash plate, allowing both sides of the pump to be varied. In case of an open circuit pump, the swash plates do not need to be rotated over centre. As an alternative it would also be possible to keep one side of the floating cup pump at a constant displacement in combination with a variable displacement, over centre design at the other side. If the variable displacement side of the unit could be operated between full minus and plus stroke, the total flow of the constant and the variable displacement side of the machine could be varied between zero and maximum displacement. This would however result in large internal flows in the pump and have a negative effect on the efficiency at small displacement.

The design, which is described in this paper, has a variable geometrical displacement on both sides of the machine, having a tilt angle of the barrel that can be varied between 0° and 8°. The two tilt angles are not synchronized mechanically; each barrel has its own set of control and bias pistons. Both barrels however share one single bias spring, which turns the swash plates to the maximum displacement angle at zero pressure conditions. The actuators and bias pistons and cylinders are identical to piston-cup-combinations of the rotating group. This has several advantages:

- The mass produced low cost piston-cup-combinations offer a cost effective solution for actuation and control of the swash plate angle
- The short length of the piston and cup creates a compact design
- The double configuration of two control pistons per swash plate strongly reduces the

contact force between the swash plate and the swash plate bearing
- The swash plate is well supported and the deformation of the swash plate is reduced to a minimum.

5. CONTROL OF THE SWASH PLATE TILT ANGLE

Figure 7 shows a detailed view of one of the swash plates including the actuators. The two pictures show the most extreme tilt positions of the swash plate at zero tilt angle and zero displacement, and at maximum tilt angle and maximum displacement. The pressure level in the cylinders of the control pistons determines in which direction the swash plate will rotate. In the left picture, the control pressure is set at zero. In this case the force from the bias spring and the bias piston create a torque that makes the swash plate rotate counterclockwise. In the right picture the pressure in the cylinders of the two control pistons is high. The oil for the control pistons is supplied at the control piston at the bottom left corner and from thereon via internal lines in the swash plate to the second control piston. The torque created by both control pistons is now twice as high as the torque from the bias piston. As a result the swash plate will rotate clockwise.

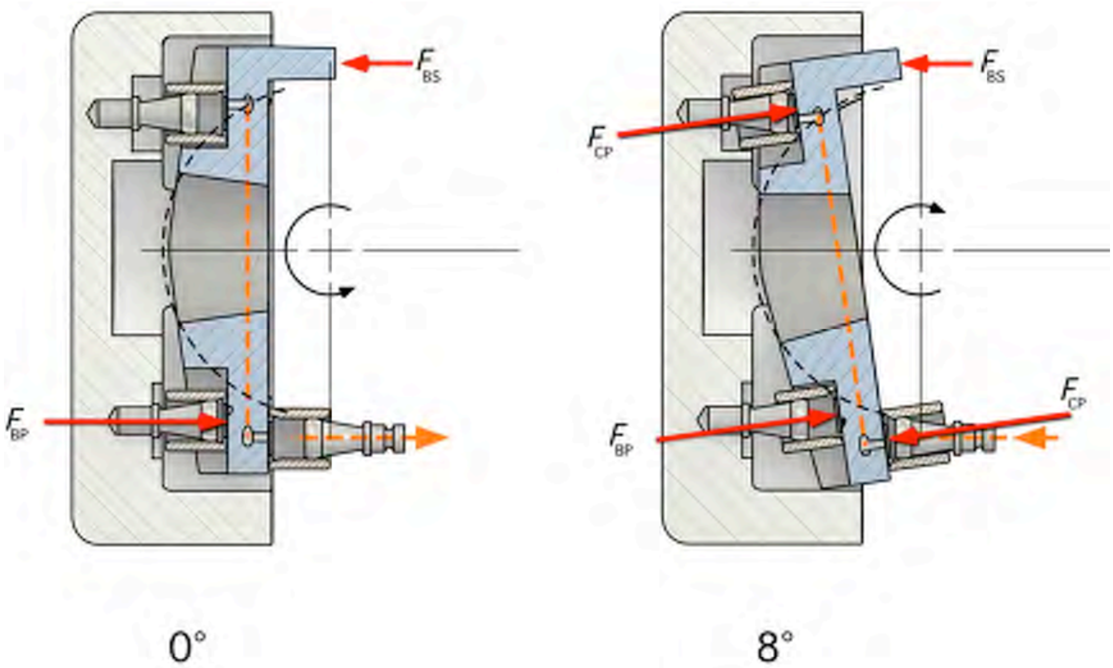

Figure 7: One of the swash plates including the control pistons and the bias piston at 0° angle (left) and 8° angle (right). In the left picture the control pistons are at low pressure and the swash plate will rotate counterclockwise because of the force of the bias piston F_{BP} and the bias spring F_{BS}. In the right picture the control pistons are loaded at a high pressure. Now the double force F_{CP} acts against the bias piston and the spring and the swash plate will rotate clockwise.

The hydrostatic forces of the control and bias pistons are always perpendicular to the port plate side of the swash plate; there are no radial forces like in current slipper type machines. Instead the oil pressure in the control and bias cylinders creates a radial load on the pistons, equally to the pistons of the rotating group (see figure 1) but this force is transferred directly to the rigid connection between the piston and the housing.

The swivel torque, which is created by the rotating barrel ports (see Figure 3) acts around the same axis of rotation as of the swash plate bearing. The bias and control pistons hold the swash plate against these torque variations, similar to the function of the actuators in a slipper type machine.

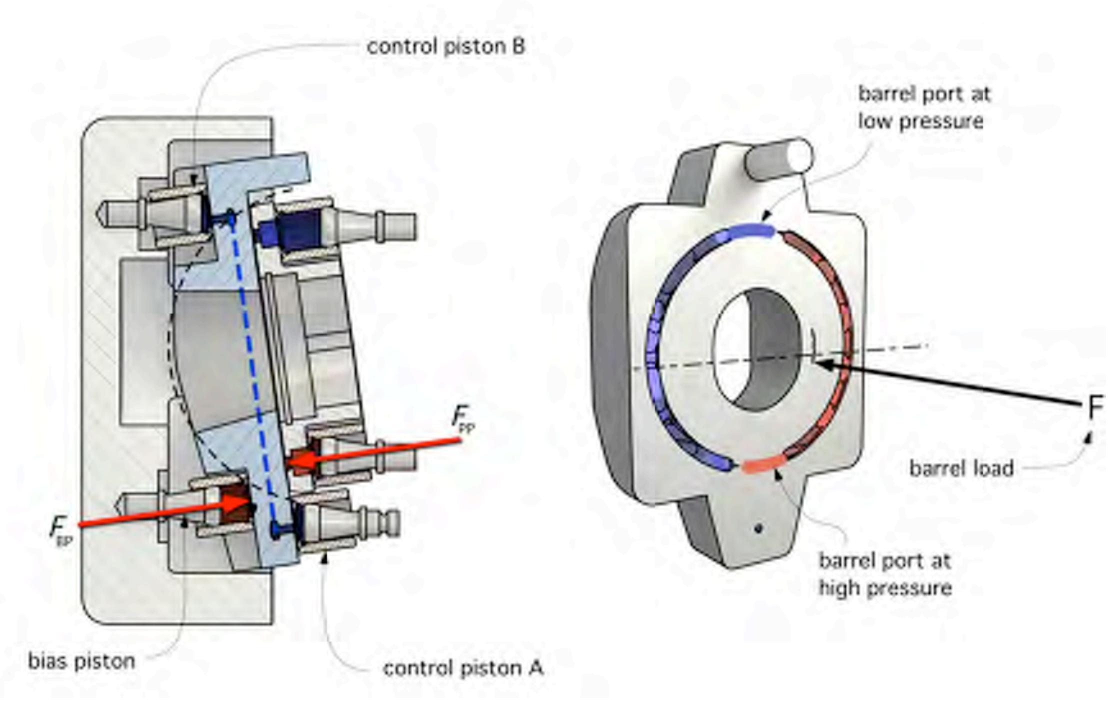

Fig. 8a: Bottom barrel port at high pressure

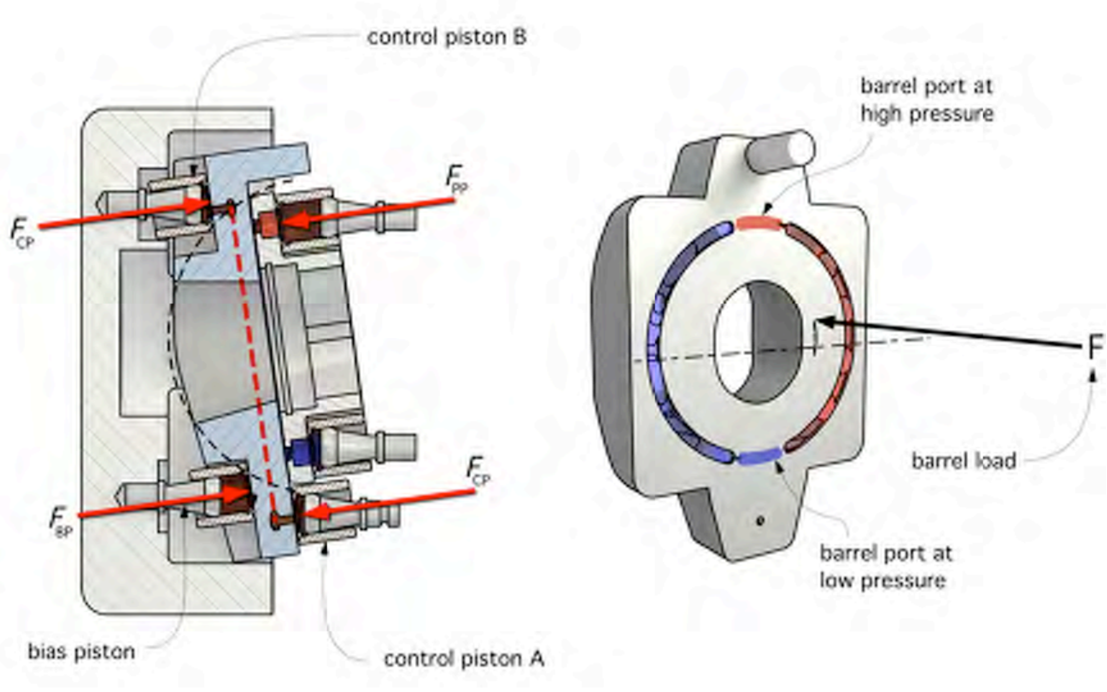

Fig. 8b: Upper barrel port at high pressure

Figure 8: Force balance of the swash plate for two barrel positions

Figure 8 shows the two barrel positions in which the swivel torque has a maximum positive or negative value. In figure 8a the barrel port at the top dead centre (TDC) position is connected to the high-pressure port of the swash plate. The resulting swivel movement of the swash plate lowers the pressure in the cylinders of both control pistons. Now the bias piston holds the force F_{pp} created by the high pressure TDC barrel port. Just a few degrees of barrel rotation further the bottom barrel port is connected to the low pressure side of the pump while the top barrel port (at the bottom dead centre) is connected to the high pressure port. The swash plate now rotates in the opposite direction thereby compressing the oil volumes in the cylinders of both control pistons (see figure 8b). Now the force F_{bp} from the bias piston is counteracted by the force F_{cp} generated by control piston A, whereas the force F_{pp} from the upper high-pressure barrel port is balanced by the force F_{cp} generated by control piston B.

6. SWASH PLATE BEARING

The design of the swash plate has to be as such that the swash plate is always pushed into the cradle bearing. The bearing force should however be as low as possible in order to reduce the friction between the swash plate and the bearing. Figure 9 depicts the pressure fields acting on the swash plate. Some of these fields have the same pressure as the suction or the delivery side of the pump. Others, like the pressure fields of the control pistons and the barrel ports that are commutating, have a varying pressure level. All forces are predominantly acting in a direction perpendicular on the port plate (the z-direction indicated in figure 9). Because the cups of the control and bias pistons always deliver the force entirely in the z-direction, the force balance in the y-direction does not need to be considered. This is an advantage compared to slipper type machines (figure 3) in which the control and bias pistons also create a force in the y-direction.

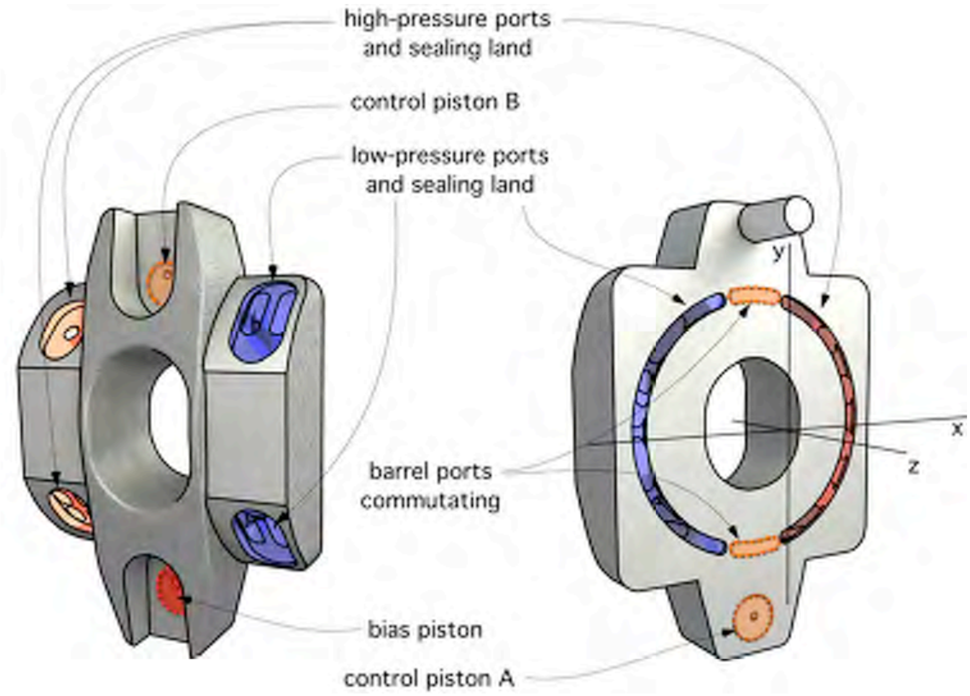

Figure 9: Pressurized areas acting on the swash plate

In the z-direction, the pressure fields are balanced almost completely:

- The pressure field from the high-pressure kidney acting on the front side of the swash plate is balanced with the pressure fields from the high-pressure ports and the bias piston at the backside of the swash plate. These back ports are also needed for the connection to the stationary supply and delivery ports in the pump case.

- In case of a pre-charged suction side, the low pressure kidney on the front is balanced by the low-pressure port and sealing area at the back of the swash plate.

- The two control pistons create forces in opposite z-directions and are therefore also completely balanced.

- The two remaining pressure fields, created by the barrel ports that are commutating from one kidney to the other, are not directly compensated by another pressure field, acting on the back of the swash plate. But having an even number of pistons, the commutation occurs simultaneously: while in one port the oil is compressed, the oil in the other port expands. Hence, during a barrel rotation, the sum of the forces of the two commutating ports is approximately constant. Consequently this force can be balanced with the high-pressure ports at the backside of the swash plate.

Compared to the conventional designs of slipper type machines, the new variable displacement machine utilizes the advantage of having an even number of pistons per barrel. Because of the offset of the piston positions between the left and right side of the machine, the floating cup still has the advantage of an odd number of pistons, without needing to compromise the axial force balance of the swash plate.

The bearing force of the swash plate in slipper type machines is further increased because of the single control piston in these machines. Since the pressure level in the control piston is not directly related to the pump pressure, it cannot be compensated in the design of the swash plate. In the floating cup design this is solved by having two control pistons, one on each side of the swash plate. All in all, the bearing force in the floating cup design is reduced by more than 90%. This of course strongly reduces the friction in the swash plate bearing which also lowers the hysteresis in the pump control and improves the dynamical behavior of the pump.

7. SWASH PLATE DEFORMATION

The swash plate is a heavily loaded component subject to deformation. Figure 10 shows the swash plate of the variable displacement slipper type pump from figure 3. The prime load comes from the pistons that are running on the high-pressure side of the machine. Added to this are the forces coming from the bias and control pistons. At the backside these forces are balanced by two pressurized area, one at each bearing side. The forces are however not in line. The construction is like a bridge bending at the middle of the swash plate [17].

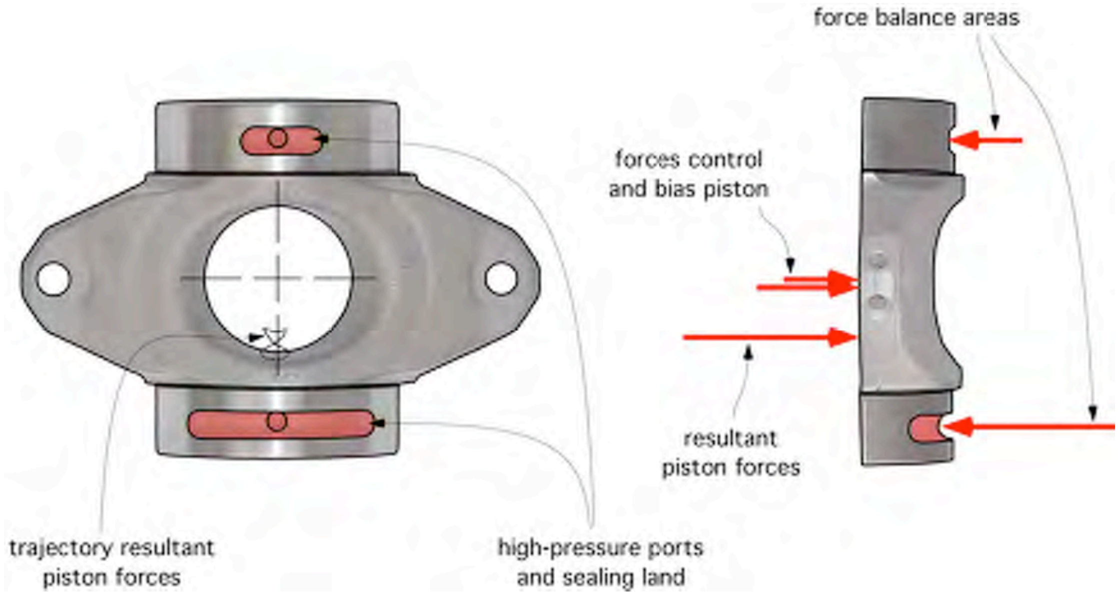

Figure 10: loads on a swash plate of a variable displacement slipper type pump

In a slipper type pump the swash plate supports the piston slippers. These slippers can follow any deformation of the swash plate and the deformation will have little or no effect on the leakage characteristics of the pump. In the floating cup pump however the swash plate supports the barrel. Since the barrel does not allow for any kinematic flexibility it is crucial that the deformation of the swash plate is minimized. This is realized for a part by increasing the stiffness of the swash plate but especially by creating a situation in which the forces that act on the front side of the swash plate are more in line with the forces acting on the backside (see figure 11).

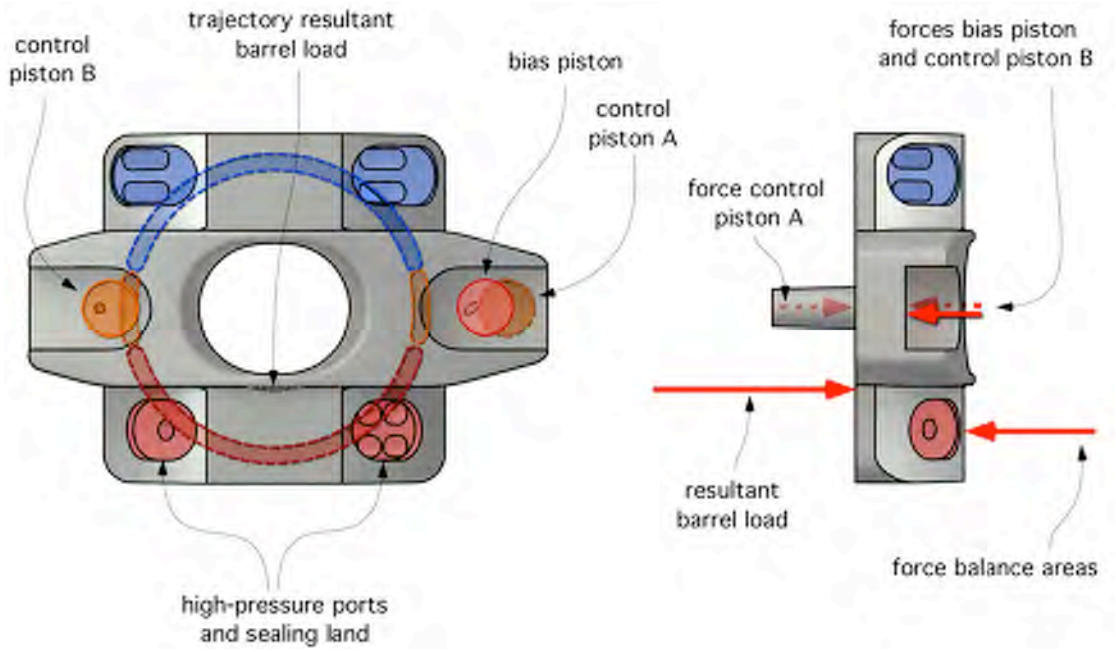

Figure 11: Loads on a swash plate of the variable displacement floating cup pump. The dotted lines refer to pressure fields acting on the port plate side of the swash plate. In the right picture the forces from the areas at low pressure and the bias spring are neglected.

In the final design of the variable displacement floating cup pump, the port plate has been separated from the swash plate. The figure below shows the outcome of the FEM-analysis of the cast iron swash plate (GGG-60), the brass port plate and the steel barrel. The deformation is only shown in the axial z-direction, perpendicular on the plane of the port plate. From the FEM-analysis the gap height between the barrel and the port plate and between the port plate and the swash plate is calculated. The diagrams in figure 12 only show the gap height for one side of the high pressure sealing lands.

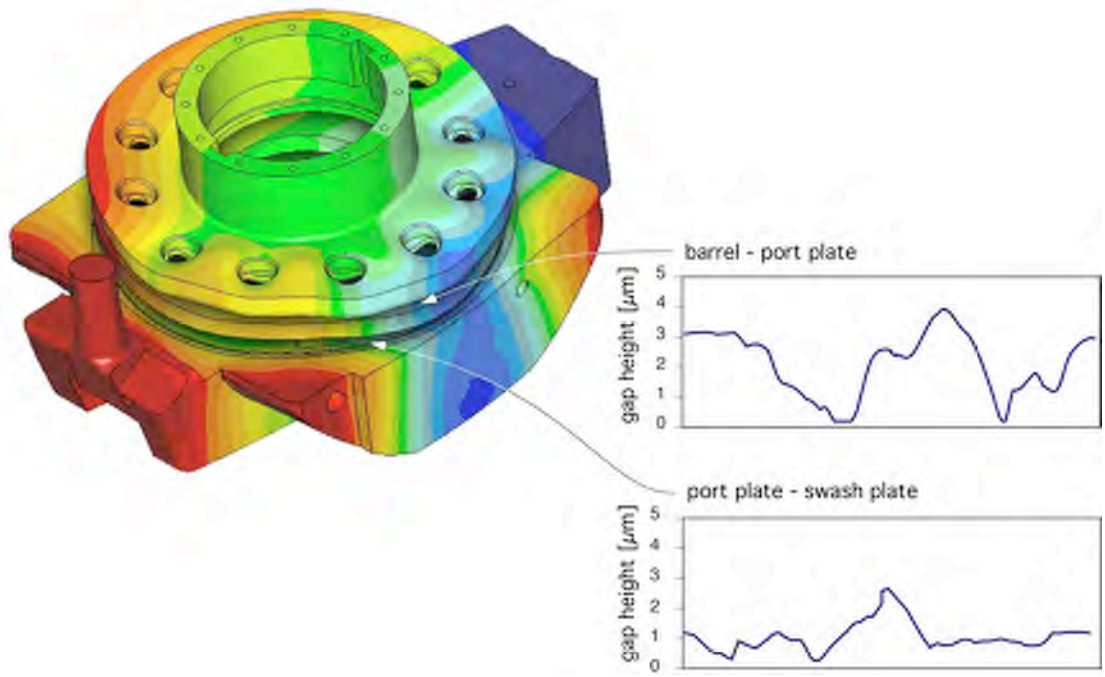

Figure 12: FEM-analysis of the cast iron (GGG60) swash plate, the separate brass port plate and the steel barrel at a pump pressure of 35 MPa showing the deformation in the axial direction and the gap height at the high pressure sealing

On average the gap height caused by the deformation is only a few micron. This is at a pump pressure of 35 MPa. Even at this pressure level the leakage across all high pressure ports is less than 0,1 L/min for a 28 cc pump, taken into account that the floating cup pump has two barrels, two port plates and two swash plates. Compared to a maximum flow output at 3000 rpm of around 84 L/min, this leakage represents a volumetric loss of around 0,1%.

8. CONCLUSION

The design of the new, variable displacement, floating cup pump had quite a few challenges. At a first impression it seems that the need for a double swash plate actuation results in an expensive and large pump, much more complicated, expensive and voluminous than the current slipper type pump. In addition the large diameter of the port plate and the swash plate might easily result in considerable deformations of these components, and it seems quite difficult to control the leakage losses.

The first consideration has been solved by means of applying the same pistons and cups as are applied in the rotating group of the floating cup machine. These components can be produced extremely cost effective in a similar way as many automotive components are produced. The combination of a cup and piston is also about the most compact linear actuator, which is especially important for the floating cup principle since the tilt angle of both swash plates needs to be controlled. Additional advantages of the piston-cup-actuators are the low friction characteristics and the possibility to eliminate side forces acting on the swash plates.

The second consideration, the deformation of especially the swash plate, has been resolved first of all by increasing the stiffness of the swash plate compared to equivalent swash plate designs of current slipper type pumps. But most important the positions of the control and bias pistons and the position of the swash plate bearings has been chosen as such that the hydrostatic forces acting on the front of the swash plate are almost in line with the hydrostatic forces at the backside. The design is realized by having two instead of only one control piston, whereby the two control pistons are positioned as such that the hydrostatic forces are balanced and only a pure torque is created.

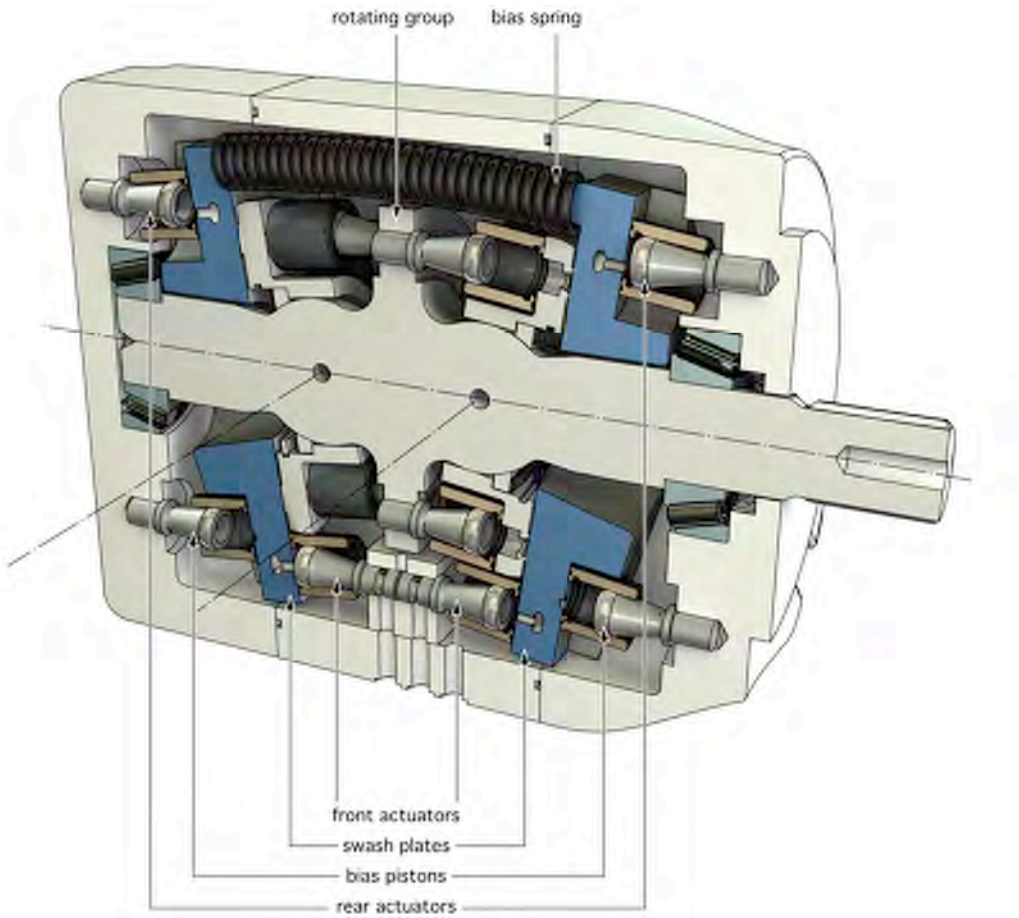

Figure 13: cross section of the open circuit, variable displacement floating cup design.

The figure above shows a cross section of the design. Although most of the actuators are positioned at the backside of both swash plates, the design is still very compact. This is for a part due to the small maximum tilt angle of the floating cup principle i.e. the small swivel movement of the swash plates. Also, the short length of piston-cup-actuators helps to reduce the length of the pump.

Furthermore the new variable displacement principle offers the opportunity to almost completely balance the hydrostatic forces acting on the swash plate. Compared to current slipper type machines the contact force between the swash plate and the bearing is reduced by as much as 90%, which will considerably improve the dynamic behavior of the pump.

Furthermore it is expected that the characteristics that already have been proven with the constant displacement machine are also valid for the variable displacement version of the floating cup principle. This is especially important for the part load efficiency, which will be improved due to the low torque losses that have already been proven with the constant displacement floating cup machines.

REFERENCES

1. Achten, P.A.J., 2002. "Dedicated design of the Hydraulic Transformer", Proc. IFK.3, Vol. 2, IFAS Aachen, pp. 233-248
2. Van den Oever, J.P.J., I. Lopez Arteaga, and P.A.J Achten, 2002, "Voruntersuchungen des Floating Cup Axialkolbenprinzips". Proc. 3. Kolloquium Mobilhydraulik, TU Braunschweig
3. Achten, P.A.J., T.L Van den Brink, T. Paardenkooper, T. Platzer, H.W. Potma, M.P.A. Schellekens and G.E.M. Vael, 2003, "Design and testing of an axial piston pump based on the floating cup principle", Proc. SICFP'03, Vol. 2, Tampere University of Technology, pp. 805-820
4. Achten, P.A.J., 2003, "Designing the impossible pump", Proc. Hydraulikdagar 2003, Linköping University
5. Rob van Malsen, 2003, Floating sensation, IVT International
6. "Der 'floating cup'-Prototyp: Verdrängung ohne Schrägscheibe/-achse", 2003, Fluid, (July/August)
7. Achten, Peter A.J., "Hydraulic device", PTC patent application WO03058035A1 (published 2003)
8. Platzer, T., R.A.H. van Malsen and P.A.J. Achten, 2004, "Floating Cup - Ein neues Konstruktionsprinzip für hydrostatische Maschinen", Ölhydraulik und Pneumatik, (5)
9. Achten, P.A.J., T.L. van den Brink and J.W. Potma, 2004, "Movement of the cups on the barrel plate of a floating cup, axial piston machine", Int. Journal of Fluid Power, 5(2)
10. Vael, G.E.M., I. López Arteaga and P.A.J. Achten, 2004, "Reducing Flow Pulsations with the Floating Cup Pump: Theoretical Analysis", Proc. PTMC 2004, Bath
11. Achten, P.A.J., M.P.A. Schellekens, H. Murrenhoff and M.Deeken, 2004, "Efficiency and low speed behavior of the floating cup pump", SAE 2004-01-2553
12. Achten, P.A.J., 2004, "Power density of the floating cup axial piston principle", Proc. Of the 2004 ASME IMechE Congress and Expo, IMECE2004-59006
13. Murrenhoff, H. and M. Deeken, 2004, "Wirkungsgradmessungen an einer Kolbenpumpe - Abschlussbericht", RWTH-Aachen
14. Murrenhoff, H. and M. Deeken, 2004, "Langsamlaufverhalten einer Axialkolben¬maschine", RWTH-Aachen
15. Achten, P.A.J., 2005, "Volumetric losses of a multi piston floating cup pump", Proc. Of

the 50^{th} National Conference on Fluid Power, NCFP-paper I05-10.2
16. Manring, N.D., R.E. Johnson, 1994, “Swivel Torque within a Variable-Displacement Pump”, Proc. of the Expo and Technical Conference for Electrohydraulic and Electro-pneumatic Motion Control Technology, NFPA-paper 94-1.3.
17. Kudav, G., L. Kasper, H. Zhang, R. Kimpel, 2005, “Development of A Heavy Duty Piston Pump Using Flexible Body Approach”, Proc. Of the 50^{th} National Conference on Fluid Power, NCFP-paper I05-10.3