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POWER DENSITY OF THE FLOATING CUP AXIAL PISTON PRINCIPLE

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ABSTRACT

The floating cup principle is a new concept for hydrostatic pumps, motors and transformers. It features a large number of pistons, which enables a strong reduction of the pressure pulsations and fluid borne noise. The pistons are arranged in a double ring, back-to-back configuration, and are locked onto the rotor. Each piston has a separate, cuplike cylinder, which is floating on a rotating barrel plate.

This article will discuss the consequences of the floating cup design on the power density of pumps and motors. For current axial piston units the trend is towards larger tilt angles of the barrel. In slipper type machines, the tilt angle can be as large as 21°, whereas in bent axis machines the tilt angle can even be increased to 45°. For a 24 piston floating cup machine, the tilt angle of each barrel is however limited to about 12°.

The object of the article is to prove that the reduced tilt angle does not need to have a detrimental effect on the power density of the hydrostatic machine. After giving a brief description of the floating cup principle, the article will focus on the design aspects that limit the barrel tilt angle. After this, the main parameters that govern the power density are discussed. A comparison with a slipper type pump is made.

NOMENCLATURE

D	Piston diameter	[mm]
Db	Barrel diameter	[mm]
F	force	[N]
H	Height of the barrel	[mm]
k	Total number of barrels per pump	[-]
L	Length	[mm]
R	Radius piston pitch circle	[mm]
s	Piston stroke	[mm]
V	Displacement volume	[cm^3]
VR	Volume of the rotary group	[dm^3]
x	Distance between two pistons	[mm]
zk	Total number of pistons per pump	[-]

Greek symbols:

β	Tilt angle of the barrel	[°]
γ	Piston pitch angle	[°]
λ_1	Piston pitch ratio	[-]
σ	Stress	[N/mm^2]

Subscripts:

ax	Axial
FC	Floating Cup
p	pressure
rad	radial
ST	Slipper type

ABOUT ASSUMPTIONS AND CONVENTIONS

Designs are often based on assumptions and conventions: an axial piston pump should have a large tilt angle of the barrel, the number of pistons is limited to nine and the majority of the components has to be manufactured by means of milling, drilling and grinding. This is the way it always has been and – as is believed– always will be.

Innovations by nature offend these unwritten rules. They have to, since the paradigm of conventions and assumptions is nothing more and nothing less than a definition of what already exists. So does the floating cup principle, which has been developed for application in axial piston pumps, motors and transformers [1-10]. The new principle is characterized by a small barrel tilt angle, a large number of pistons, and the components of the rotary group are intended to be manufactured by means of deep drawing, extrusion, sintering and other mass production manufacturing techniques.

The first pump prototype, which has been constructed according to this principle, has 24 pistons, divided in two rings of 12 pistons each (see Fig. 1). The two rings are arranged in a mirrored, back-to-back configuration. Each ring has its own barrel, which has a tilt angle of only 8°. From a production (and cost) point of view the new pump resembles more like a bearing or a set of hydraulic lash adjusters [3]. Unlike current axial piston pumps, many components of the floating cup pump can be made by means of low-cost, high volume production technologies, like deep drawing, extrusion and sintering.

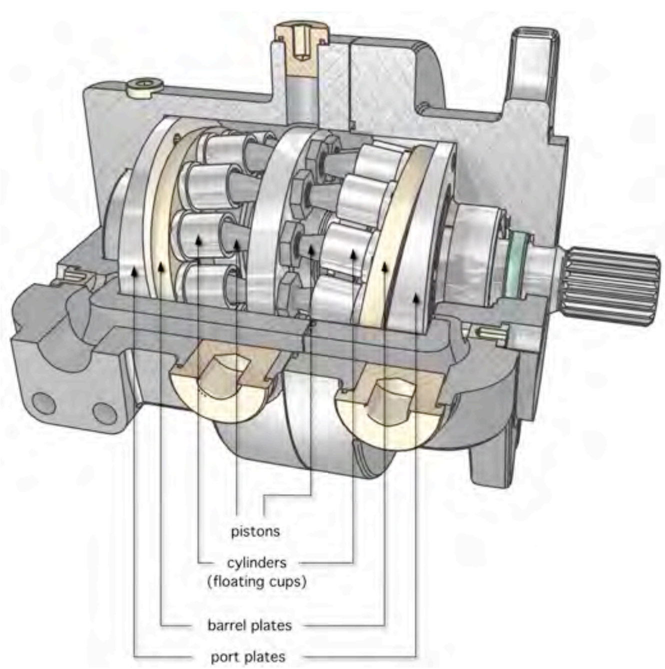
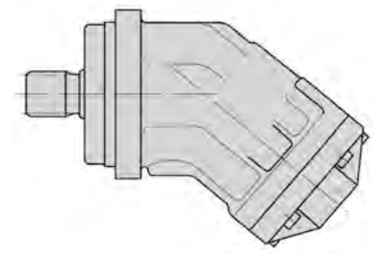


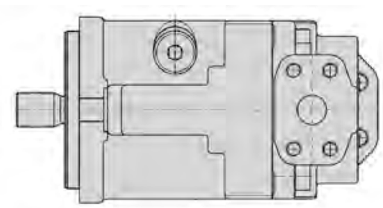
Fig. 1: Cut-away view of the first prototype of a floating cup pump

Yet, it could be argued that the small tilt angle of the barrel in the floating cup design results in a reduction of the power density and thus in a larger pump size. The trend in pump development is clearly in the direction of larger tilt angles. Mayr [11] for instance, reported about an increase of the barrel angle of a slipper type machine from 25° to 40°. As a result the weight of the unit decreased by 30 to 40%. The power density per unit of volume increased according to Mayr by 35 to 60%, depending on the displacement volume of the unit.

On the other hand, the type of pump principle also influences the power density. Figure 2 shows the contours of two axial piston pumps, both having a constant displacement of 28 cc/rev: a bent axis pump and a slipper type pump. The bent axis pump has a tilt angle of the barrel of 40° whereas the tilt angle of the swash plate in the slipper type unit is only 20°. Despite the big difference in tilt angle, the power to weight ratio of the bent axis unit is only 13% better than of the slipper type pump. This is partly due to the higher rotational speed of slipper type pump at self-priming conditions. On the other hand the weight of the slipper type unit is higher because it is designed for high speed, self-priming operation resulting in a large and heavy flange at the suction side of the pump. The motor version of the slipper type pump, which doesn't have this flange, only weighs 11 kg.



Bent axis pump [12]
Tilt angle barrel: 40°
Constant displacement 28.1 cc/rev
Mass: 9.5 kg
Maximum pressure: 400 bar
Maximum rotational speed: 2500 rpm
Maximum power: 47 kW
Power density: $\frac{47\,\text{kW}}{9.5\,\text{kg}} = 4.95$ [kW/kg]



Slipper type pump [13]
Tilt angle swash plate: 20°
Constant displacement 28 cc/rev
Mass: 13.5 kg
Maximum pressure: 400 bar
Maximum rotational speed: 3000 rpm
Maximum power: 56 kW
Power density: $\frac{56\,\text{kW}}{13.5\,\text{kg}} = 4.15$ [kW/kg]

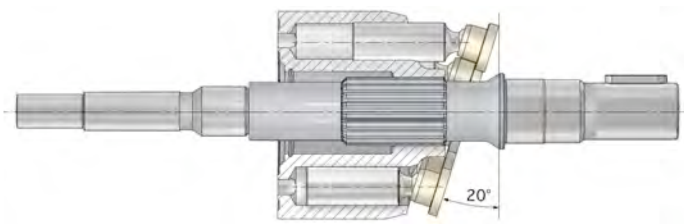
Fig. 2: Comparison of the size, weight and power density of a bent axis pump versus a slipper type pump.

In this respect the question about the power density of the new floating cup principle cannot simply be answered by looking only at the tilt angle of the barrel. A more fundamental analysis is needed to investigate the effects of the floating cup principle on dimensions and power density. This is the topic of this paper. The analysis will take the current prototype as a starting point. As a reference, the factors that determine the size of an equivalent slipper type pump will be examined as well. The analysis is focused on the size and dimensions of the rotary group (especially the pistons and the barrels) since these components are mainly accountable for the size of the entire pump.

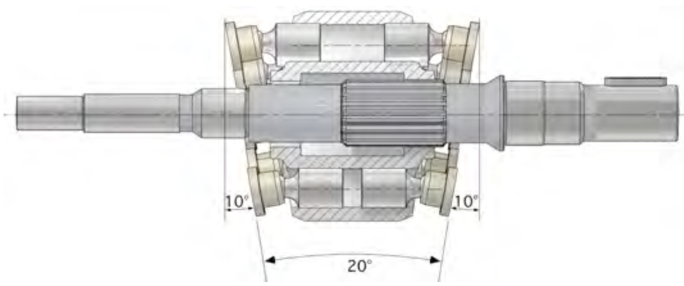
THE FLOATING CUP PRINCIPLE

One of the most important objectives in the design of the floating cup principle was to cut the pressure pulsations by a factor of 4 to 5 by means of increasing the number of pistons. In principle a larger number of pistons could also be realized by creating a mirrored slipper type configuration, as is shown in Fig. 3b. Although the swash plate angle of each half would than be reduced to only 10°, it is obvious that the size of the complete rotating group is nearly the same as for the original design.

The configuration shown in Fig. 3b does however not offer any benefits. Because each pair of pistons shares one common cylinder, the two halves of the machines have to be completely synchronized. Consequently the pump will effectively still behave as if it had the same number of pistons as in the original configuration shown in Fig. 3a.



3a: conventional slipper type



3b: 'split' slipper type

Fig. 3: two possible configurations of a slipper type pump

This is changed in the new floating cup design (Fig. 4). Whereas in the 'split slipper type' design the pistons were broken up into two separate pistons, in the floating cup design the barrels are split. Now the port timing or commutation of the two sides of the pump can be made out of phase, which truly creates a multi-piston machine with low pulsations [2, 8, 10].

The total angle between the two barrels can in theory become as large as 24°, as will be shown in this paper. Contrary to bent axis and slipper type machines, the pistons of the floating cup machine are locked onto the rotor: there is no ball joint or any other kind of linkage between the piston and the rotor. The conversion of hydraulic power to mechanical power is directly realized at the ball shaped piston crown (see Fig. 5).

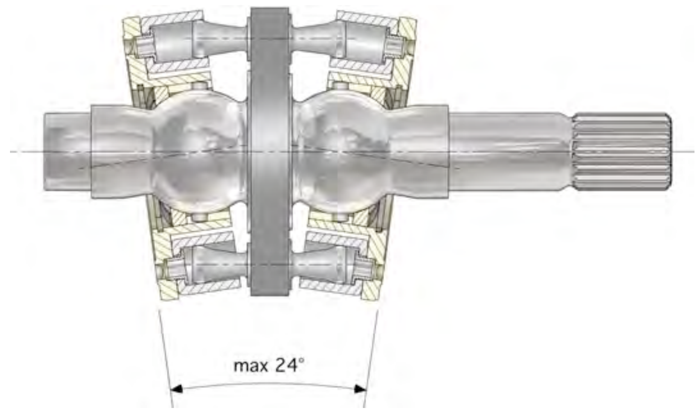


Fig. 4: cross section of a floating cup rotary group

If there had been a linkage between the piston and the rotor, the non-axial forces acting on the piston would have to be counteracted by the cylinders, i.e. the sliding interface between the piston and the cylinder. In the floating cup design however, the cuplike cylinders –which are floating on and supported by the barrel plate– are completely balanced in the radial direction. Consequently there is almost no load between the cups and the pistons and friction and wear are very low.

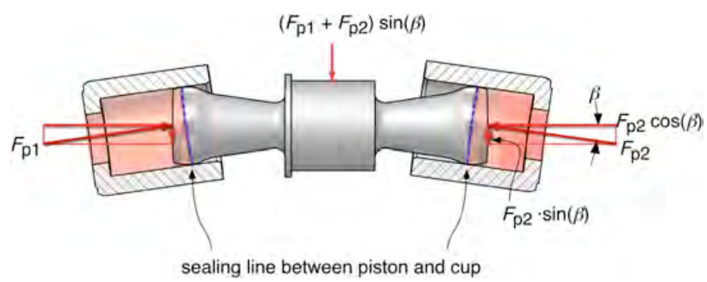


Fig. 5: Force balance on a pair of pistons

Earlier this year, the Institute of Fluid Power Drives and Controls (IFAS) of Aachen University have tested a first prototype of a floating cup pump [8]. The tests have proven the low friction characteristics of the floating cup pump. The total efficiency of the pump is on average 5% higher than of a comparable, modern slipper type pump. Furthermore the pressure pulsations in the output line are reduced by 75 to 80%.

SIZE OF THE ROTARY GROUP

The power density of a hydrostatic pump or motor is dependent on many factors. A higher efficiency of the pump for instance results in a higher hydraulic output for the same size and weight. Concerning the power to weight ratio, the choice of materials, especially for the housing, is of great influence. But this is not a specific concern for the power density of the floating cup pump: in principle the floating cup machine can use the same materials as any other hydrostatic machine.

In the end the power density of a hydrostatic machine is most and for all determined by its core: the rotating group. And even than, an evaluation of the power density is not unambiguous. The size and weight of the rotating group of a bent axis machine are for example largely affected by the size of its bearings. In this paper the influence of the bearing structure will not be studied since it involves parameters, like the loading and lifetime of the bearings, which are difficult to assess. For this reason the evaluation of the power density will be limited to two pump principles, which have more or less the same bearing demands: a slipper type pump and a floating cup pump. In both pumps the large hydraulic forces acting on the barrels and the pistons are hydrostatically compensated.

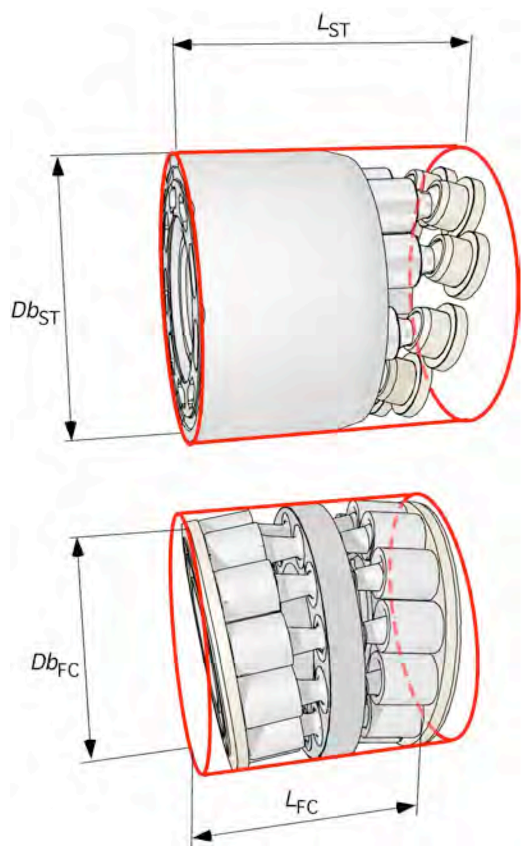


Fig. 6: Definition of the volume *VR* of the rotary group of an in-line slipper type machine (above) and a floating cup machine (below)

For both principles, the volume VR of the rotary group is defined by a cylinder that fits around the rotary group (see Fig. 6):

$$VR_{ST} = \frac{\pi}{4} \cdot Db_{ST}^{\;2} \cdot L_{ST} \tag{1}$$

and

$$VR_{FC} = \frac{\pi}{4} \cdot Db_{FC}^{\;2} \cdot L_{FC} \tag{2}$$

A consequence of this rather simple definition is, that for both machines, some void volume is also included, especially the wedge shaped volume to the right and below of the slippers (in case of the in-line pump), and the two wedges at the left and the right side of the reference cylinder of the floating cup machine. But for practical reasons the housing will be more or less cylindrical too, and the void spaces will therefore also contribute to the final dimensions of the hydrostatic machine.

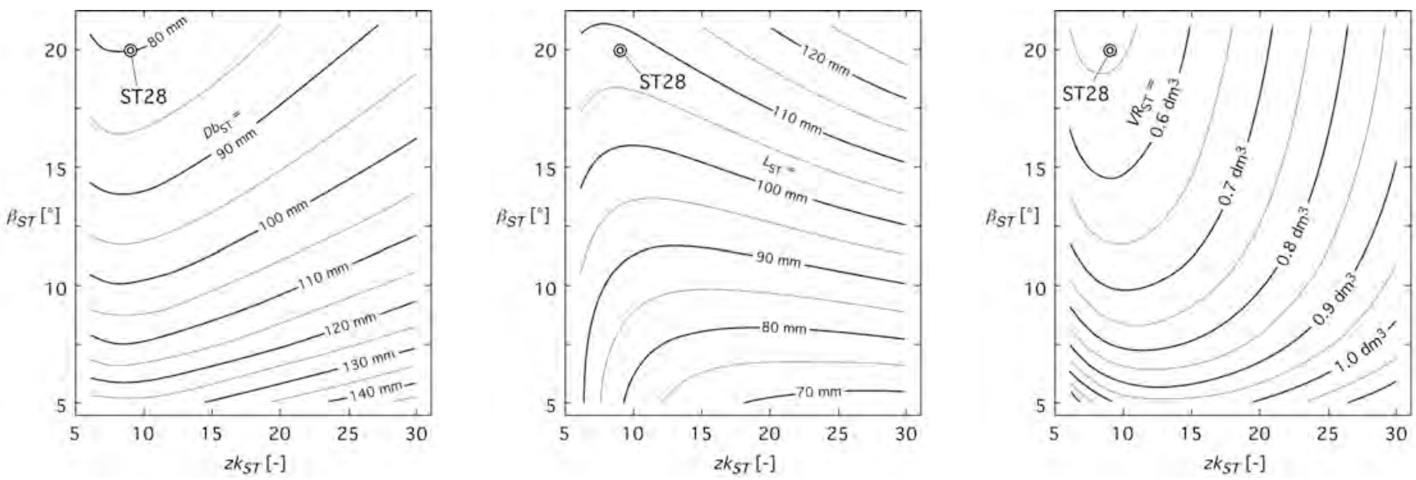
The shaft is not included in the evaluation. Furthermore, the evaluation is based on a single size i.e. 28 cc/rev, which is the displacement volume of the first prototype of the floating cup pump. As a reference the A4FO28 slipper type pump from Bosch Rexroth is chosen, having also a displacement volume of 28 cc/rev. Both machines have the same specifications for maximum pressure level and maximum speed (also at self-priming conditions). Experiments have already proven that the floating cup pump is self-priming up to a rotational speed of 3000 rpm.

The floating cup concept has three important characteristics that could have a large influence on the dimensions of the rotary group and hence on the power density of the total pump:

- the double barrel configuration
- the high number of pistons
- the small tilt angle of the barrels

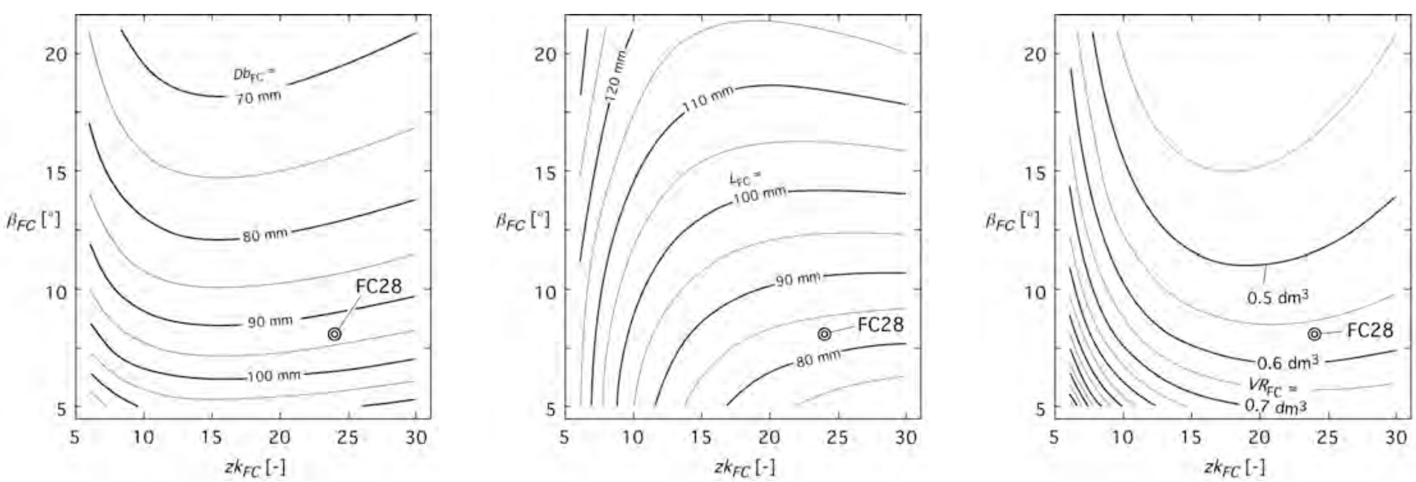
Although it is in principal possible to conceive the floating cup concept with a single barrel [7] many of the advantages of a double, mirrored configuration would get lost. Therefore, only the double version of the floating cup principle will be evaluated. The other two parameters, the total piston number zk and the tilt angle β of the barrel are the main parameters of the evaluation in this paper. For the slipper type pump the tilt angle β_{ST} is defined as the angle between the central axis of the barrel and the axis of the swash plate (see also Fig. 18 in appendix A). In the floating cup machine β_{FC} represents the angle between the central axis of the barrel and the shaft (Fig. 19 in appendix A).

In appendix A the equations for the dimensions of both machines are derived as a function of β and zk. The results are shown in Fig. 7 and 8. For all combinations of β and zk the calculated dimensions result in a total displacement volume of 28 cc/rev. Aside from efficiency effects, the power output of the machines is the same.



7a: Barrel diameter Db_{ST} 7b: Length L_{ST} of the rotary group 7c: Volume VR_{ST} of the rotary group

Fig. 7: Calculated dimensions of the rotary group of a 28 cc/rev slipper type pump as a function of the total number of pistons zk_{ST} and the tilt angle of the swash plate β_{ST}. Point ST28 in the diagram indicates a reprensentative slipper type pump having 9 pistons and a swash plate angle of 20°.



8a: Barrel diameter Db_{FC} 8b: Length L_{FC} of the rotary group 8c: Volume VR_{FC} of the rotary group

Fig. 8: Calculated dimensions of the rotary group of a 28 cc/rev floating cup pump as a function of the total number of pistons zk_{FC} and the tilt angle of the barrel β_{FC}. Point FC28 in the diagram indicates the current prototype of the floating cup pump having 24 pistons and a tilt angle of each barrel of 8°.

Figures 7a and 8a show the calculated barrel diameter Db for both units. For the floating cup machine the barrel diameter is about equal to the diameter of the rotor on which the pistons are mounted. As can be seen the diameter of floating cup rotary group is generally smaller than of the slipper type unit. This despite the fact that in the floating cup machine, the distance in between the pistons is larger, resulting in higher values for λ_1. It should however be noticed that zk represents the total number of pistons. That is to say, for the floating cup pump zk stands for the total number of pistons for the two barrels together. This explains why, for the same values of β and zk, the barrel diameter of the floating cup machine is generally smaller than of the slipper type machine. At a total piston number of (2 x 9 =) 18 pistons and a barrel angle of (20°/2 =) 10° the piston diameter of the floating cup machine should be about the same as of a 9 piston slipper type, and also the number of pistons per barrel would the same. For these conditions, the barrel diameter of the floating cup pump is calculated to be 86 mm. This is slightly larger than for a slipper type having 9 pistons and a barrel angle of 20°, which has a barrel diameter of about 80 mm. The difference between the two diameters is due to the larger distance between the pistons of the floating cup machine. With a total number of 24 pistons and a barrel angle of only 8° (the parameters for the floating cup prototype) the difference is even larger. In that case the diameter of the rotary group is calculated to be 93 mm.

Fig. 7b and 8b show the contour plots for the calculated length of the rotary group. This time the length of both principles varies in the same range, between 70 and 120 mm. The dependency to the barrel angle and the number of pistons is however very different. For the slipper type unit the length is predominantly dependent on the barrel angle. Yet, with the floating cup machine the number of pistons has a much larger influence, especially at small piston numbers.

Comparing a standard slipper type machine, having 9 pistons and a barrel tilt angle of 20°, to a configuration similar to the floating cup prototype, it becomes clear that the floating cup rotary group is shorter (83 mm for the floating cup machine against 108 mm for the slipper type).

The total volume of both rotary groups is depicted in Fig. 7c and 8c. Again the values for the two machines are in same range: between 0.55 and 1.1 dm^3. The volume of the rotary group becomes smaller (and hence the power density higher) by increasing the tilt angle β of the barrel. This fits with the current trend towards larger tilt angles of the barrel.

The relationship between the volume *VR* and the total piston number *zk* is however more complicated. At very small and at relatively high number of pistons the volume of the rotary group is large. In between, a minimum value of *VR* can be found. For the slipper type principle the volume of the rotary group is smallest if the total number of pistons is relatively small. In case of a slipper type machine with a tilt angle β = 20°, the optimum number of pistons is 9, which –not surprisingly– corresponds to the current state-of-the-art of axial piston slipper type pumps and motors.

For the new floating cup principle however the situation is quite different. Now the volume of the rotary group is smallest if the total number of pistons is somewhere between 18 and 24, which is much higher than in current axial piston pumps and motors.

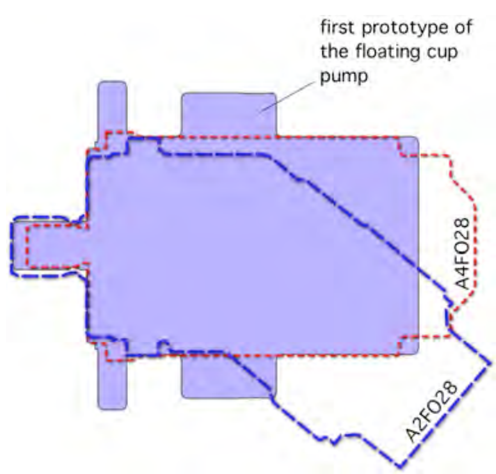


Fig. 9: Comparison of the shapes and sizes of three axial piston pumps, all of them having a constant displacement of 28 cc/rev.

A floating cup pump having 24 pistons and a barrel angle of 8° (indicated with point FC28 in Fig. 8) has a volume VR_{FC} of 0.57 dm^3. This is about the same as of the standard slipper type machine (indicated with point ST28 in Fig. 7) having a volume of the rotary group VR_{ST} of 0.54 dm^3. The floating cup rotary group is shorter, but has a larger diameter than the rotary group of the slipper type machine. This is also reflected in the dimensions of the complete pumps. Figure 9 shows the contours of the first prototype of the floating cup pump, compared to the shape of a slipper type pump (Bosch Rexroth A4FO28) and a bent axis pump (Bosch Rexroth A2FO28). Again the floating cup pump is shorter, but also wider than current pumps.

STRENGTH CONSTRAINTS

The first prototype of the floating cup pump (FC28 in Fig. 8) has not been designed for having an optimum power density. Figure 10 shows the sensitivity of the volume *VR* for a change of the barrel angle β, the total number of pistons *zk* and the piston pitch λ_1. Reducing the piston pitch i.e. the distance between two neighboring pistons is one of the best strategies to reduce the volume of the rotary group of the floating cup pump. This can only be achieved if the thickness of the cup wall can be reduced. In this paragraph, this option will not be discussed any further.

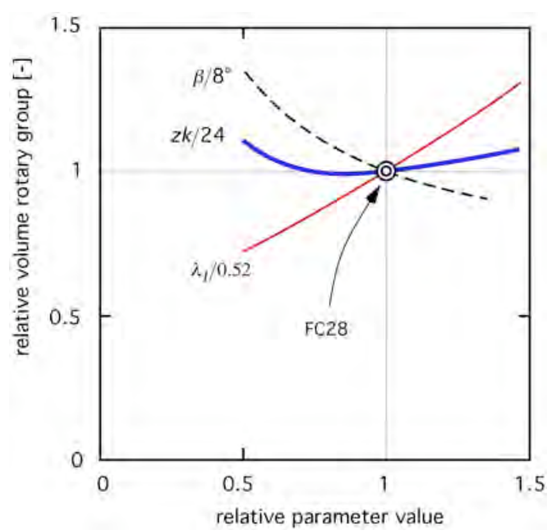


Fig. 10: Influence of design parameters on the volume of the rotary group of the floating cup machine compared to the reference configuration (FC28)

Another option to reduce the volume of the rotary group is to increase the tilt angle of the barrels in the floating cup machine. For instance, by increasing the barrel angle from 8 to 11° the volume of the rotary group decreases from 0.57 to 0.51 dm^3. Yet, there are some physical and geometrical constraints that limit the tilting of the barrel angle. In a recent publication [9] the relative movement of the cups on the barrel plate is discussed. At large barrel tilt angles, this relative movement can become too large to seal off the leakage between the cup and the barrel plate. More stringent demands are however set by the strength of the piston neck and the shaft diameter (see Fig. 11).

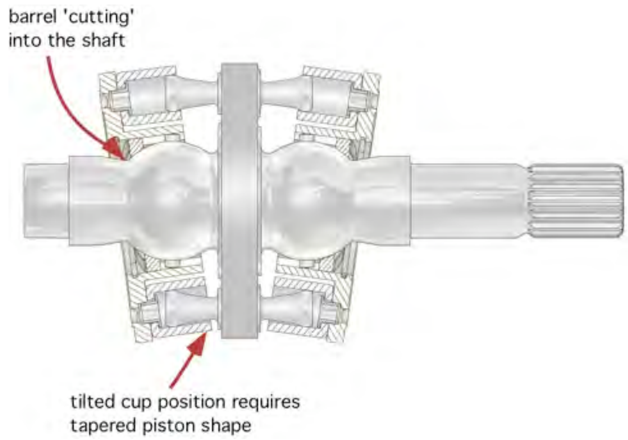


Fig. 11: Constraints of the tilt angle of the barrel.

Due to the tilted cup position the piston has to be more or less tapered, having the smallest diameter at the neck close to the rotor. Furthermore the tilted barrel 'cuts' into the shaft. In the end the shaft could become too small to carry the radial load and the torque created by the pistons. Both strength constraints are discussed in this paragraph.

<u>The piston neck</u>

The tilt angle β_{FC} of the barrel has several effects on the loading of the piston neck (see Fig. 12). An increase of the tilt angle for instance results in:

- an increased radial load on the piston, which results in a higher bending stress at the piston neck
- a reduced axial load on the piston
- longer pistons (provided the radius of the piston pitch circle stays the same), which also results in a higher bending stress at the piston neck
- a reduction of the diameter of the piston neck

The combination of a smaller cross section of the piston neck, a longer piston and an increased load causes a higher local stress in the piston neck. In Appendix B the maximum stress at the cross section of the piston neck is calculated as a function of the total number of pistons zk and the tilt angle of the piston b. Given a maximum stress value of 600 N/mm^2, it can be calculated for which configurations this demand is fulfilled. The result is shown in Fig. 13. Eventually the (calculated) diameter of the piston neck has to become zero or even negativ in order to allow for the tilted position of the cup.

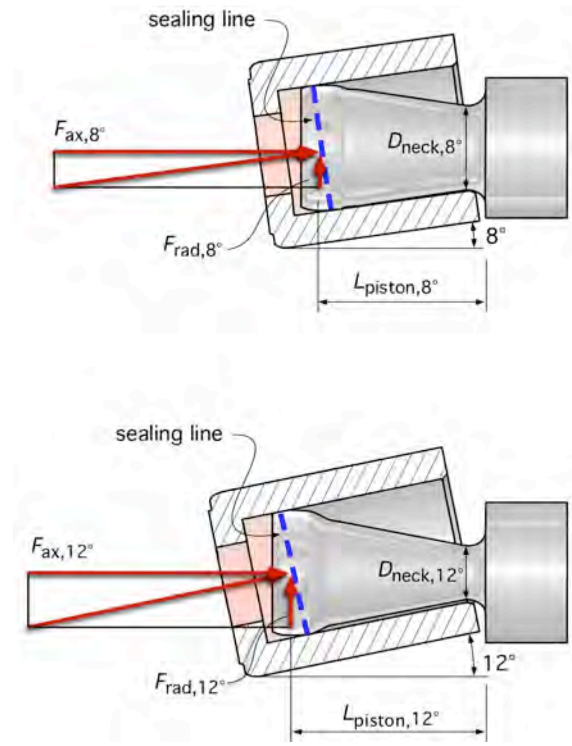


Fig. 12: Piston-cup-combination at β_{FC} = 8° (above) and β_{FC} = 12° (below)

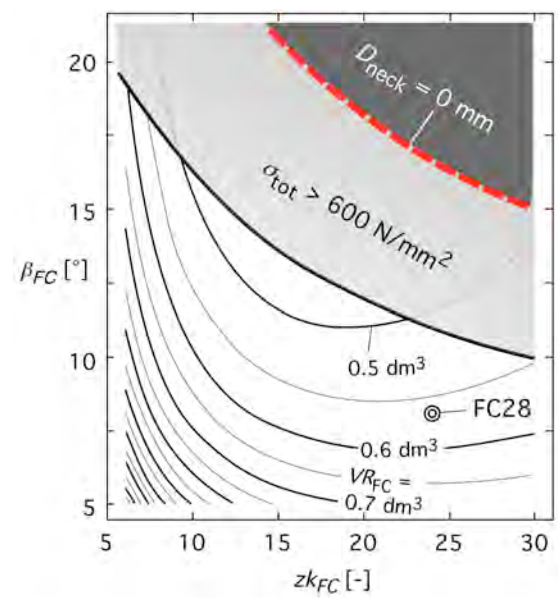


Fig. 13: Design constraints for the floating cup principle concerning the total stress σ_{tot} and the diameter D_{neck} of the piston neck (p = 400 bar, V = 28 cc/rev, λ_1 = 0.52)

Shaft diameter

In Appendix C the influence of the number of pistons and the tilt angle of the barrel on the maximum allowable (local) shaft diameter is calculated. For a load condition of 400 bar and a displacement volume of 28 cc/rev it can be calculated that the shaft needs to have a diameter of at least 20 mm. This results in a second constraint for the possible configurations of the floating cup principle. Figure 14 shows the calculated results.

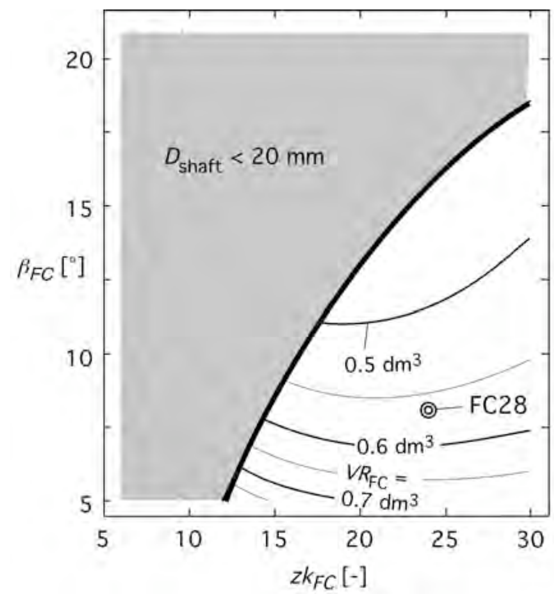


Fig. 14: Design constraints for the floating cup principle concerning the minimum shaft diameter D_{shaft} (p = 400 bar, V = 28 cc/rev, λ_1 = 0.52)

It is clear that a combination of a small tilt angle of the barrel and a high number of pistons leave more room for the shaft in the centre of the barrel. More pistons per barrel result in a larger piston pitch circle, forcing the pistons on a larger radius. On the other hand a larger tilt angle tips barrel more towards the shaft, thereby reducing the shaft diameter.

CONCLUSIONS

In Fig. 15 the contour plots of Fig. 13 and 14 are combined into one plot. The tilt angle β_{FC} of the barrel can be increased to a maximum value of 12°. For the two barrels together this results in a total barrel angle of 24° (see also Fig. 4). The volume of the rotary group of the floating cup machine can be reduced to about 0,5 dm3, at which point the volume VR_{FC} is about 10% smaller than of a comparable slipper type unit. Although in this paper the comparison is limited to a rather small pump, having a displacement of only 28 cc/rev, the equations show that these conclusions are also valid for larger pump sizes.

As was already concluded from the comparison between slipper type and bent axis machines, the barrel angle is not the only parameter that determines the power density of a hydrostatic machine. The kind of displacement principle has apparently a dominant effect on the geometry and dimensions. Each principle has its own constraints: whereas the current axial piston principles have a built-in tendency to limit the number of pistons to about 9, the new floating cup principle is by its nature a multi-piston machine. The volume of the rotary group, and hence the power density of the new design is equivalent or even better than current slipper type machines.

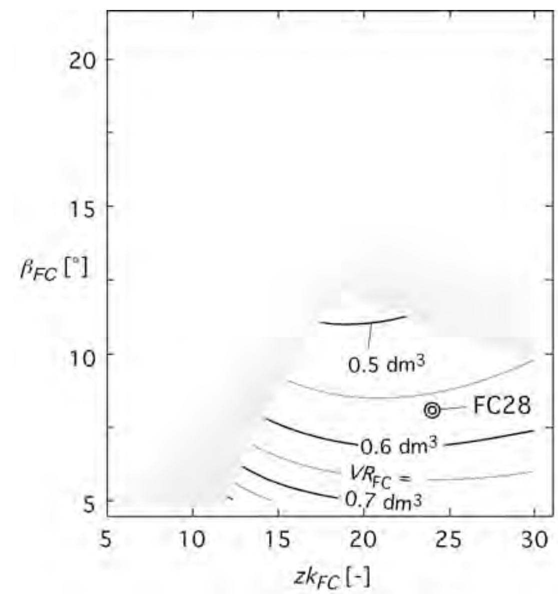


Fig. 15: Volume of the rotary group of a 28 cc/rev floating cup machine as a function of the total number of pistons zk_{FC} and the barrel tilt angle β_{FC}, also showing the constraints for the diameter of the shaft and the piston neck.

The Floating Cup principle is in the early stages of its development. Up till now, only a constant displacement pump and motor have been built. For this reason the analysis in this paper is also limited to constant displacement machines. It remains yet to be seen how the control of the two barrels in the floating cup machine will be realized and how this will affect the power density.

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APPENDIX A: CALCULATION OF THE DIMENSIONS OF THE ROTARY GROUP

Figure 16 shows the barrel of a slipper type pump having 9 pistons. The figure indicates the most important parameters of the barrel with respect to the volume of the rotary group. The equivalent parameters for the floating cup machine are shown in Fig. 17.

For calculating the dimensions of the rotary group as a function of the barrel angle and the number of pistons some dimensionless parameters have to be defined. The first of these parameters is the piston pitch:

$$\lambda_1 = \frac{x}{D} \tag{3}$$

or

$$x = \lambda_1 \cdot D \tag{4}$$

From current designs of slipper type pumps it follows that $\lambda_{1,ST} \approx 0.33$. In the floating cup design, each piston has it's own cylinder. Because of the double cylinder wall between two neighboring pistons, the piston pitch for the floating cup machine is larger. From the floating cup prototype it follows that $\lambda_{1,FC} \approx 0.52$. From the following equation

$$\frac{zk}{k}\arcsin\left(\frac{D+x}{2\cdot R}\right) = \pi \tag{5}$$

an equation for R can be derived

$$R = \frac{\left(\lambda_1 + 1\right)\cdot D}{2\cdot\sin\left(\gamma\right)} \tag{6}$$

In this equation, γ represents the piston pitch angle:

$$\gamma = \frac{\pi \cdot k}{zk} \tag{7}$$

wherein k represents the number of barrels over which the total number of pistons is divided. For a slipper type pump k =1, for the floating cup pump k = 2. Furthermore zk represents the total number of pistons of the pump.

Given the radius of the piston pitch circle the diameter of the barrel Db can be determined:

$$Db = 2\cdot R + D + x \tag{8}$$

By substituting Eq. 4 and 6 the latter equation can be written as:

$$Db = \frac{\left(\lambda_1 + 1\right)\cdot\left(\sin\left(\gamma\right)+1\right)\cdot D}{\sin\left(\gamma\right)} \tag{9}$$

The displacement volume V of the pump is equal to:

$$V = \tfrac{1}{2}\pi \cdot R \cdot zk \cdot D^2 \cdot \sin\left(\beta\right) \tag{10}$$

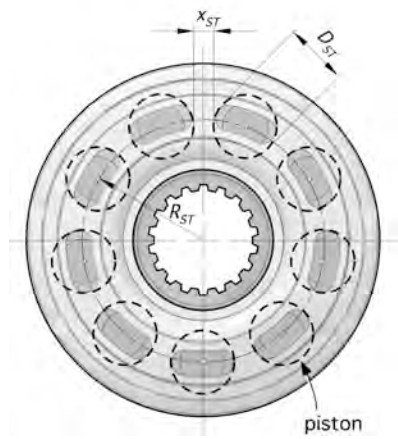

Fig. 16: Rotor parameters of the slipper type pump

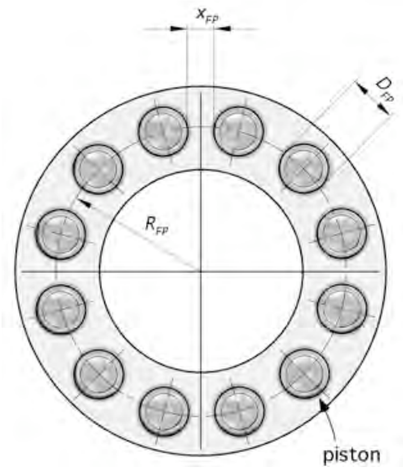

Fig. 17: Rotor parameters of the floating cup pump

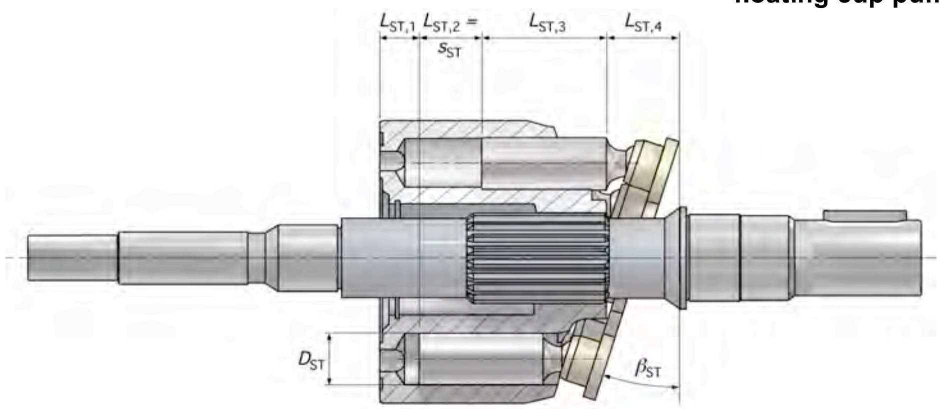

Fig. 18: Parameters defining the length of the rotary group of the slipper type machine

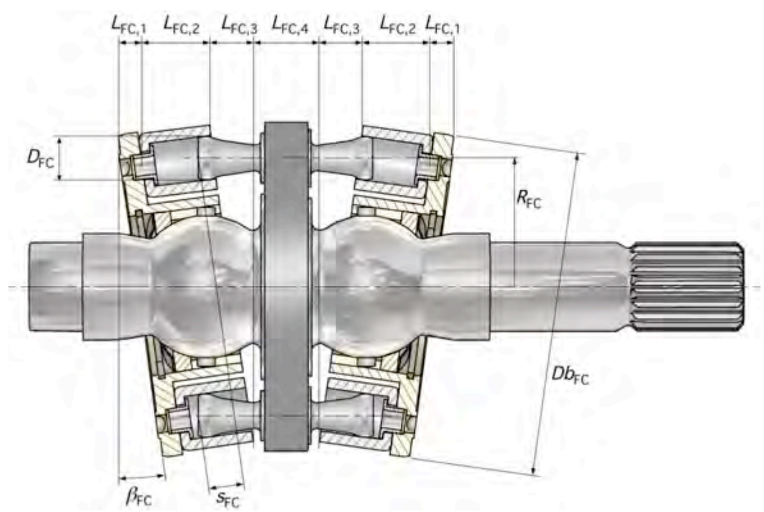

Fig. 19: Parameters defining the length of the rotary group of the floating cup machine

After substituting Eq. 6 into Eq. 10 the latter formula can be rewritten in order to obtain an equation for the piston diameter D:

$$D = \sqrt[3]{\frac{4 \cdot V \cdot \sin(\gamma)}{\pi \cdot (\lambda_1 + 1) \cdot zk \cdot \sin(\beta)}} \tag{11}$$

The piston stroke s can be calculated from:

$$s = 2 \cdot R \cdot \sin(\beta) \tag{12}$$

After substituting Eq. (6) and (11) in Eq. 12 it follows:

$$s = \sqrt[3]{\frac{4 \cdot V}{\pi \cdot zk}} \cdot \left[\frac{(\lambda_1 + 1) \cdot \sin(\beta)}{\sin(\gamma)}\right]^{2/3} \tag{13}$$

The structure of the rotary group of the floating cup pump is in the longitudinal direction very different from that of a slipper type pump. Figures 18 and 10 show a cross section of both principles, indicating the most important parameters defining the length of the rotary group. For the slipper type pump the length of the rotary group can be estimated by:

$$L_{ST} = L_{ST,1} + L_{ST,2} + L_{ST,3} + L_{ST,4} \tag{14}$$

in which:

$$L_{ST,1} = 0.8 \cdot D_{ST} \tag{15}$$

$$L_{ST,2} = s_{ST} \tag{16}$$

$$L_{ST,3} = 2.0 \cdot s_{ST} \tag{17}$$

$$L_{ST,4} = 1.4 \cdot D_{ST} \tag{18}$$

For the floating cup machine the length of the rotary group can be described by:

$$L_{FC} = 2(L_{FC,1} + L_{FC,2} + L_{FC,3}) + L_{FC,4} \tag{19}$$

In this equation:

$$L_{FC,1} = 0.65 \cdot D_{FC} \cdot \cos(\beta_{FC}) \tag{20}$$

$$L_{FC,2} = Db_{FC} \cdot \sin(\beta_{FC}) \cdot \cos(\beta_{FC}) \tag{21}$$

$$L_{FC,3} = 1.4 \cdot s_{FC} \cdot \cos(\beta_{FC}) \tag{22}$$

$$L_{FC,4} = 1.1 \cdot D_{FC} \tag{23}$$

APPENDIX B: CALCULATION OF THE STRESS IN THE PISTON NECK

In order to allow for the tilted position of the cup, the piston must have a (more or less) tapered shape. At the neck of the piston, where the diameter of the piston is smallest, the diameter can be calculated with the following equation:

$$D_{neck} = D_{FC} - 2 \cdot s_{FC} \cdot \tan(\beta_{FC}) \tag{24}$$

The piston is loaded in the axial direction:

$$F_{ax} = p \cdot \frac{\pi}{4} \cdot D_{FC}^2 \cdot \cos(\beta_{FC}) \tag{25}$$

and in the radial direction:

$$F_{ax} = p \cdot \frac{\pi}{4} \cdot D_{FC}^2 \cdot \sin(\beta_{FC}) \tag{26}$$

The axial force causes a compressive stress in the piston:

$$\sigma_{compr} = \frac{F_{ax}}{\frac{\pi}{4} D_{neck}^2} \tag{27}$$

The radial force creates a torque load at the cross section of the piston neck (see Fig. 12) over an arm length:

$$L_{piston} = Db_{FC} \cdot \sin(\beta_{FC}) \tag{28}$$

The maximum bending stress created by the torque load equals:

$$\sigma_{bent} = \frac{Frad}{\frac{\pi}{32} D_{neck}^3} \cdot L_{piston} \tag{29}$$

By combining the two loads the maximum stress in the piston neck can be calculated:

$$\sigma_{tot} = \sigma_{compr} + \sigma_{bent} \tag{30}$$

By substituting the equations for D_{neck} (Eq. 24), and the values for Db (Eq. (9)), D (Eq. (11)), and γ (Eq. (7)) for the floating cup, the maximum total stress can be written as a function of the total piston number zk and the tilt angle β of each barrel. The calculated results are presented in the following contour plot (Fig. 20).

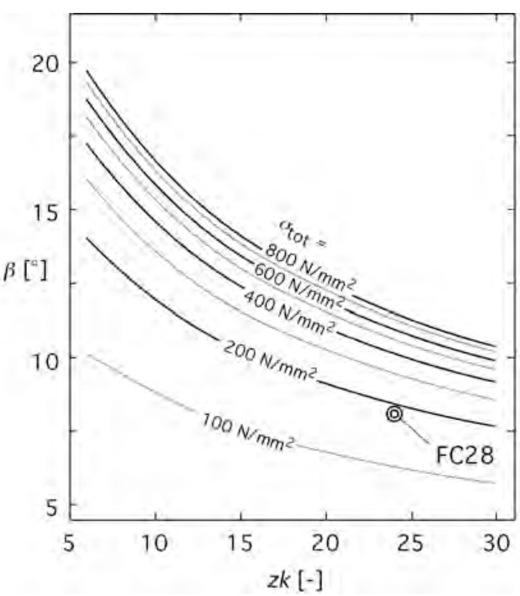

Fig. 20: Maximum stress σ_{tot} of the piston

Assuming a pure alternating load and high-grade steel as the base material of the piston, the stress in the piston may not become higher than 600 N/mm^2. This limits the maximum tilt angle of the barrel in the floating cup design. A larger number of pistons thereby results in a smaller permissible barrel angle.

APPENDIX C: SHAFT DIAMETER

In the design shown in Fig. 1 the rotor is connected to a central shaft, which is supported on both ends by means of bearings. It is also possible to support the rotor directly by means of bearings at the rotor. Also the torque in- and output could be realized directly at the rotor, for instance by means of a gear transmission. Although this would certainly increase the width of the rotor, it would avoid the necessity for having a central shaft. If however, a central shaft is needed or preferred the diameter of this shaft is limited at the place where the barrels 'cut' into the shaft, due to their tilted position (see Fig. 11). Figure 22 shows a detailed cross section of the current design of the floating cup machine.

Similar to other types of axial piston machines, the barrel is pushed to the port plate by means of a spring (in case of the floating cup design a wave ring is applied). This spring pushes against a support ring which running on the ball shaped extensions of the shaft. Although more constructions can be conceived to push the barrel against the port plate, the place that is needed for the support ring maximizes the local diameter of the shaft.

The shaft diameter at this position can be calculated by means of the following equation:

$$D_{shaft} = 2 \cdot \left(R_{FC} - \left[\frac{1+\lambda_1}{2} D_{FC} + 0.005 \right] \cdot \cos\left[\beta_{FC}\right] \right) - 2 \cdot H_{FC} \cdot \sin\left[\beta_{FC}\right] \tag{31}$$

In this equation H_{FC} is defined as the height of the barrel, including the cups:

$$H_{FC} = 0.65 \cdot D_{FC} + 1.2 \cdot s_{FC} \tag{32}$$

Figure 21 shows in a contour plot the resulting shaft diameter as a function of the total number of pistons zk and the barrel tilt angle β.

For a maximum load condition of 400 bar, it can be calculated that the shaft diameter at least has to be around 20 mm. In this calculation the shaft material is 42CrMo4, or an equivalent material. The load is not completely alternating. A safety factor of 3 has been applied to account for the combined torsion and bending load of the shaft.

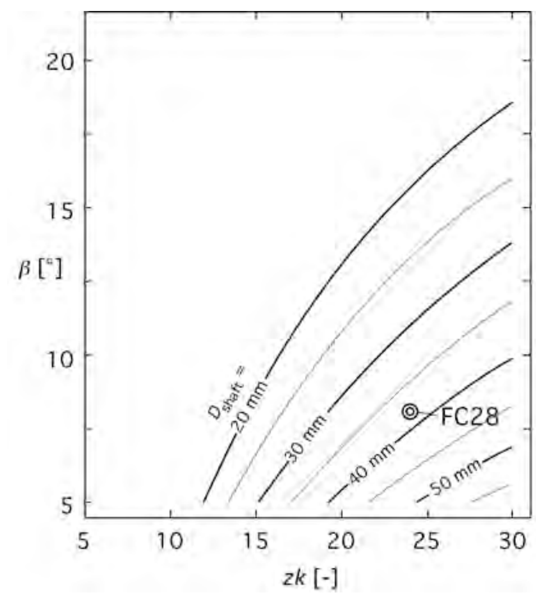

Fig. 21: Maximum allowable shaft diameter at the position where the barrel cuts into the shaft

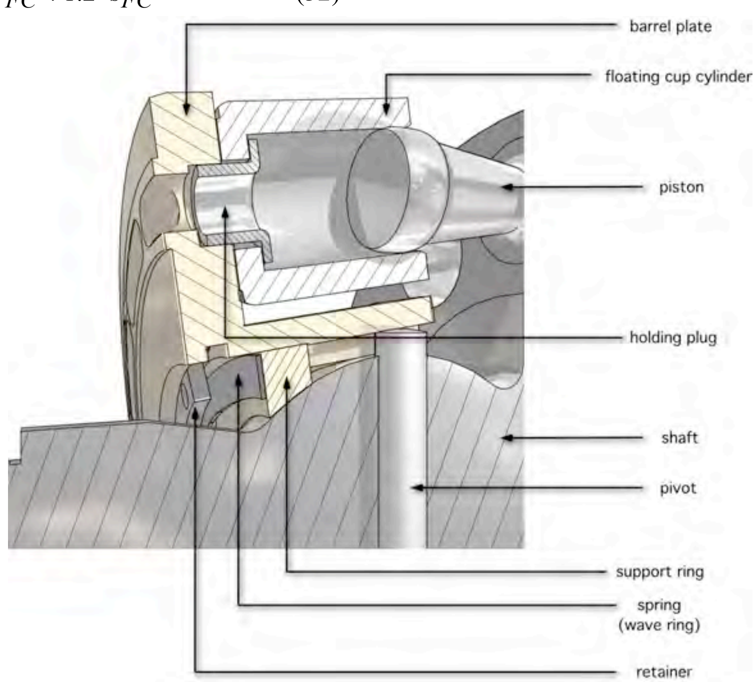

Fig. 22: Cross section of a part of the current design of the floating cup machine