Designing the impossible pump.

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Abstract

The development of products is following a life cycle. After their establishment and introduction into the market a new development phase starts in which the product is improved incrementally. This goes on until the full potential of the design concept has been utilized. After that it is only a matter of time awaiting a new concept that offers new and better perspectives.

This paper is about the design of such a new concept, a new principle for hydrostatic pumps, motors and transformers. The so-called floating cup principle belongs to the family of axial piston machines. This paper will describe the similarities and differences compared to bent axis and slipper type machines.

The new pump concept features a high number of pistons, about three times as much as in current pumps. Consequently the flow and pressure pulsations are strongly reduced. It is also expected that the high number of pistons will improve the start-up behaviour in motor operation. The new concept also features a through drive. This is important for pumps and motors for it offers the opportunity for piggybacking another pump or a brake. All these advantages are quite obvious and hardly need any prove.

Not as obvious however are the low torque losses of the floating cup principle. Measurements performed by the IFAS of the University of Aachen have already proven the high volumetric and hydro-mechanical efficiency of the floating cup pump. Especially at low rotational speeds and high operating pressures the torque losses are much lower than of comparable bent axis and slipper type machines. This paper will especially analyse the reasons for the reduction of the torque losses, in particular of the friction losses.

Keywords: axial piston pump, floating cup principle, torque losses, efficiency

1 Nomenclature

a	dimension
b	dimension
c	dimension
D	diameter
F	force
M	torque
p	pressure
R	radius of the piston pitch circle
s	piston stroke
v	velocity
z	number of pistons
□	barrel tilt angle
□	angular barrel position
□	friction coefficient
□	rotational speed

Subscripts

c	cylinder
f	friction
min	minimum
N	normal
p	Piston

2 Off the beaten track

In his book "The Innovator's Dilemma: When New Technologies Cause Great Firms to Fail" Clayton Christensen analyses a number of technology breakthroughs in the recent history of product development [1]. One of them is the appearance of hydraulic excavators in the fifties of the last century. In less than a decade the market of cable-actuated excavators was almost completely gone, and only a few excavator producers survived the change. This was the short innovation era of the hydraulic industry.

Looking back it is hard to believe that fluid power as such has ever been a disruptive technology. Moreover most people working in the hydraulic industry have completely forgotten about it (who remembers the 'giant Bucyrus-Erie'?). Instead of a continuation of the role of innovator, the hydraulic industry has focussed on sustaining the existing products and technologies: to improve incrementally the performance of established products in existing markets.

Yet, this paper is not following the same road; this paper is about an innovation, an entirely new concept of hydrostatic machines. It is called the floating cup principle [2, 3, 4] and has been introduced for the first time in March 2002. The floating cup principle is a disruptive technology in the market of hydrostatic pumps and motors. It will enable the pump industry to create new market opportunities and to change the rules of the game; at least for those industries that are still open for innovations. At other industries however, the new innovation will automatically start off their immune response systems. In the end the new technology is a mixed blessing. It will jeopardise the old production lines. And it threatens the know-how status of current pump experts, designers and producers. This is especially of relevance for current market leaders in the hydrostatic pump and motor business. They have a difficult decision to make for not only the concept is new and different, but it also involves a new production technology, that is to say, new for the hydraulic industry.

But there is something to gain here: a pump that has a high efficiency, extremely low pulsations and low noise levels. If applied in a motor the new principle improves the start-up and the low speed behaviour. And due to the kind of production technologies the new principle is designed for, it offers the opportunity for further cost reductions.

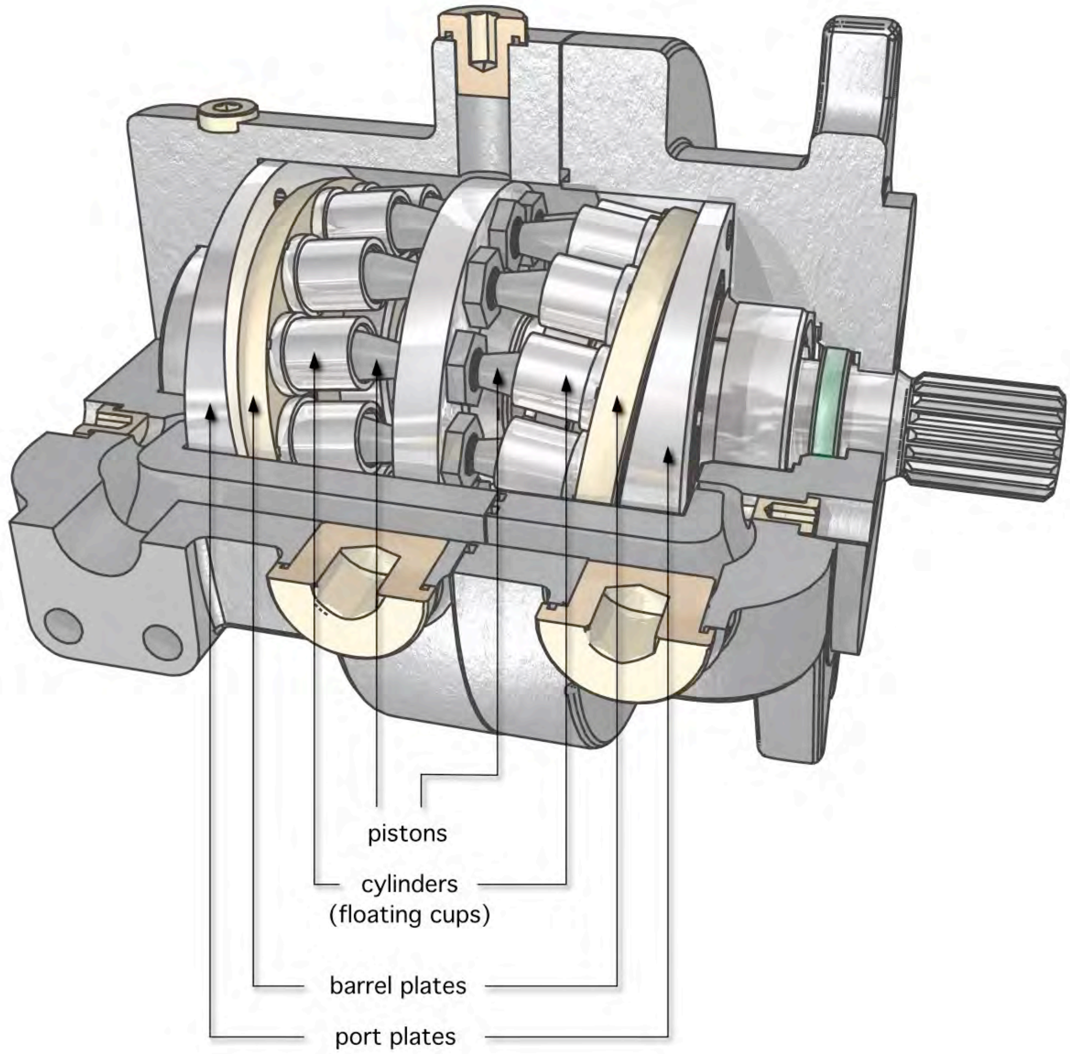
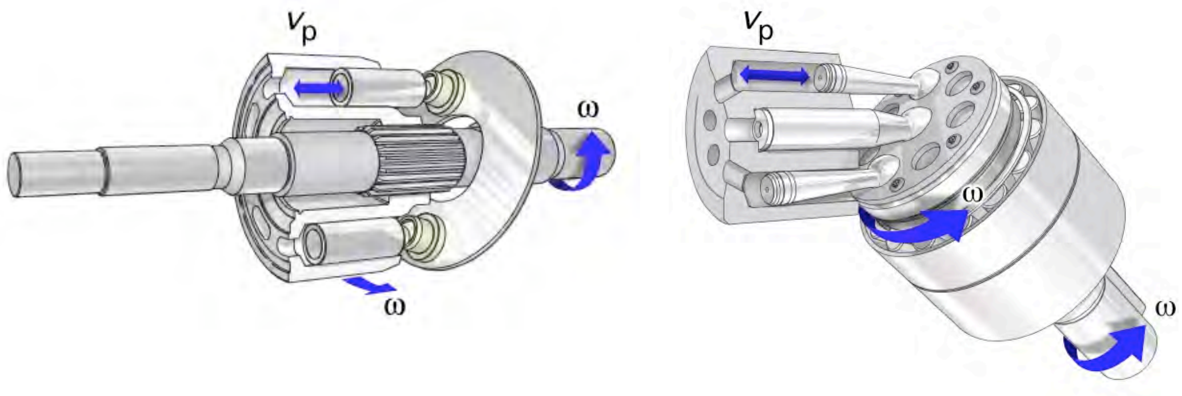


Figure 1: Cut-away view of the first prototype of a floating cup pump

This paper is focussed on the hydro-mechanical efficiency of the new principle. Most losses of hydrostatic machines are caused by friction and other hydro-mechanical losses. Aside from the energy losses the friction will also result in a reduction of the starting torque, which is especially important for motor operation.

3 Kinematics of axial piston machines

The floating cup principle belongs to the family of axial piston machines. The pistons are arranged in a circle around a central axis and the displacement is (for the most part) realized in the axial direction i.e. parallel to the axis of the barrel. In slipper type (figure 2a) and bent axis pumps (figure 2b) –currently the most important members of the axial piston family– the cylinders are all part of a solid barrel: they can only move together with the barrel in the radial direction. In both pump types, the pistons have a spherical joint at the opposite end of the piston sealing. As a result the pistons can only transfer loads in the axial direction of the piston. All other piston loads that will create a torque at the spherical joint will have to be taken by the barrel.



a) slipper type   b) bent axis

Figure 2: Kinematical principles of current axial piston machines. Only two pistons are shown.

In the floating cup principle (figure 3) this configuration is changed completely. Now the pistons are locked onto the central rotor and their movement is confined to the rotation of the rotor and shaft. Consequently the rotor takes all loads acting on the pistons. This creates the possibility to convert the hydraulic pressure forces directly into a driving torque on the drive shaft and vice versa.

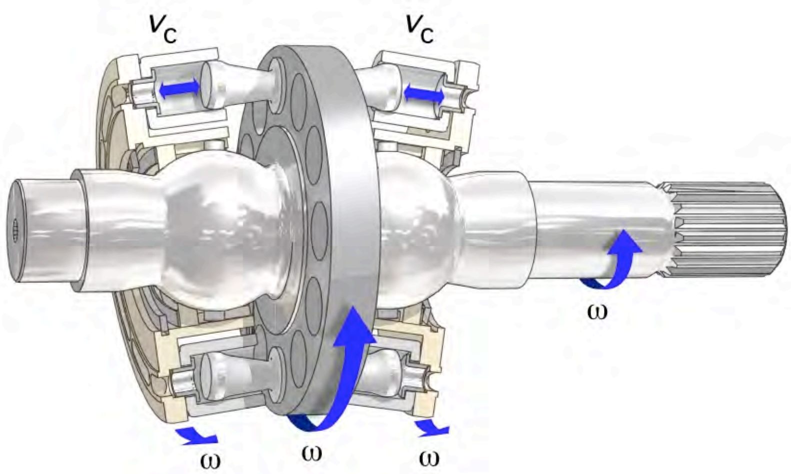


Figure 3: Kinematics floating cup principle showing only 4 of the 24 pistons.

But there are more differences. Whereas the number of pistons in bent axis and slipper type machines is limited to generally 7 or 9 pistons, the floating cup principle is specifically designed to reduce the noise level and the output pulsations by increasing the number of pistons. In the first prototype of a floating cup pump [4] in total 24 pistons are applied. The pistons are arranged in two rings of 12 pistons each. The two rings are connected to the central rotor in a mirrored back-to-back configuration.

In all three axial piston concepts shown above, the displacement is created in a similar way: part of the rotary group is rotating around a different axis than the drive shaft. In case of the bent axis machine the barrel axis has a 'bent' position. As a result this principle doesn't allow a through drive. In the slipper type machine the barrel axis is in-line with the drive shaft, thereby permitting a through drive to piggyback another pump or to connect a brake at the other end of the shaft. The tilted swash plate, on which the slippers of the pistons are running, now creates the displacement. The floating cup concept is a hybrid of the two previous principles. As in the bent axis design the barrel axis has a tilted position. Only now the angle between the central shaft and the barrel axis is so small that the shaft can run through the central hole of the barrel, thereby creating again the possibility of a through drive.

4. Barrel angle limitations

In bent axis and slipper type pumps the spherical joint at the end of the piston creates the possibility to position the piston in line with the cylinder axis, independently of the tilt angle of the barrel or the swash plate. The floating cup principle misses this extra degree of freedom and consequently the angle between the barrel and the drive shaft also determines the tilt angle between the piston and its cylinder.

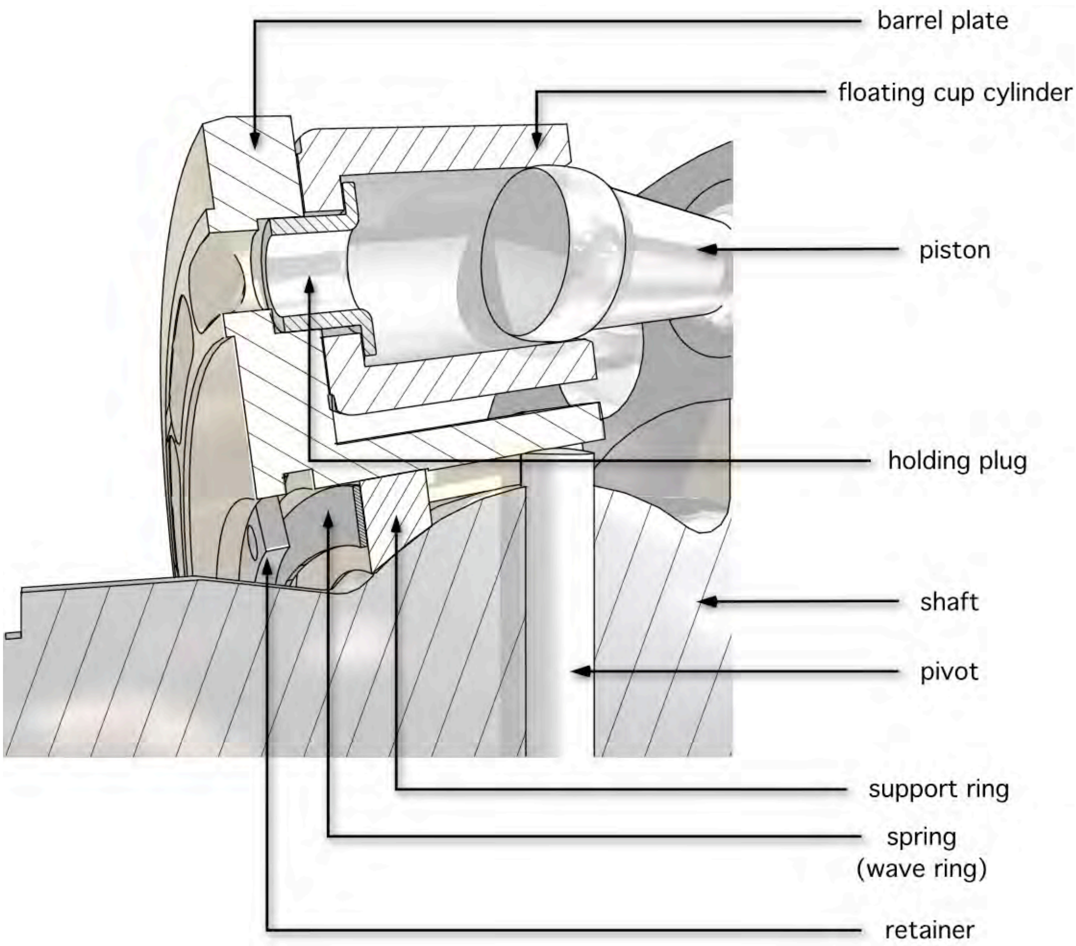


Figure 5: Detailed cross section of the floating cup concept

The lack of a swivel joint at the piston end is to some extend compensated by giving each piston a separate, cup-like cylinder which is supported by and floating on a barrel plate. The 'floating cups' are hydrostatically balanced in a similar way as the barrels in current axial piston machines. The movement of the cups on the barrel plate is very small since both the cylinders and the barrel are rotating at approximately the same speed. But because of the tilted position of the barrels the circular movement of the pistons will create an elliptic movement of the cylinders on the barrel plate. Separating the cups and allowing them to float on the barrel plate provides the extra degree of freedom that can take this elliptic deviation.

But the extra degree of freedom of the floating cups does not change the angle between the pistons and the cylinder as such. Consequently there are other limitations set by the new concept than in current bent axis and slipper type units. One of the most obvious restrictions is set by the strength of the piston neck $D_{p,min}$. Due to the angle β between the barrel and the drive shaft axis i.e between the cup and the piston, the piston must be tapered in order to give room for the cylinder when the piston is in the top dead centre position. As a result the tilt angle of the barrel cannot be larger than around 10° (assuming the pump has to operate at pressures up to 400 bar). It must be emphasized that the tapered pistons do not deliver the driving torque for the barrel rotation. Instead the barrels are taken by a pivot, which is mounted in the central drive shaft (see figure 5).

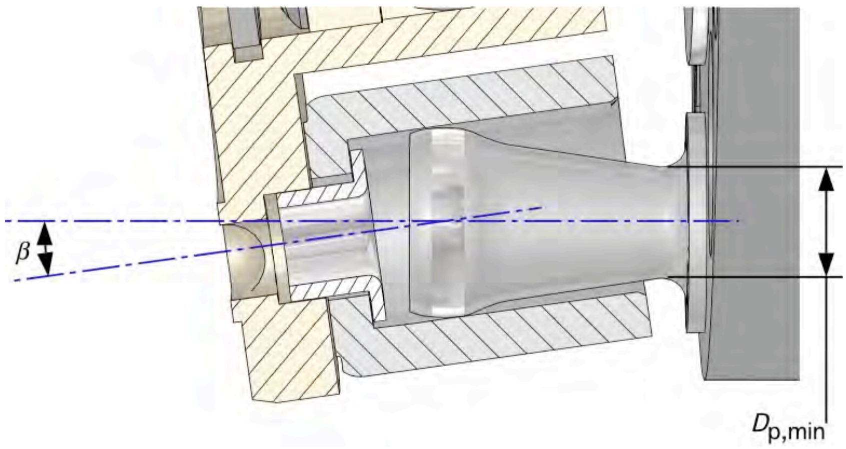


Figure 6: Cross section of one of the piston cup combinations of the floating cup pump.

Another reason for limiting the tilt angle of the barrel could be the need for having the shaft going through the central hole in the barrels. A large tilt angle would result in a small diameter of the drive shaft and would eventually even reduce the shaft size to a point where the strength is insufficient to supply the necessary torque. Aside from the tilt angle, the amount of room for the drive shaft is also dependent on the number of pistons, that is to say on the diameter of the piston pitch circle and the outer diameter of the cups. A smaller number of pistons will result in a larger piston diameter in order to keep the displacement of the pump the same. This in turn would increase the outer diameter of the cups and would consequently reduce the interior circular space between the cups. However, with 24 pistons and a barrel angle of around 10° the drive shaft diameter is not the critical parameter. The shaft can even be made much stronger and stiffer than in a comparable sized slipper type pump.

Finally there is the risk of the cups being locked between the barrel plate and the holding plug (see figure 7). A larger tilt angle of the barrel results in a longer piston stroke. Since the friction between the cup and the barrel plate has to be taken by the piston there is a torque created on the cup, which will tend to rotate the cup away from the barrel plate.

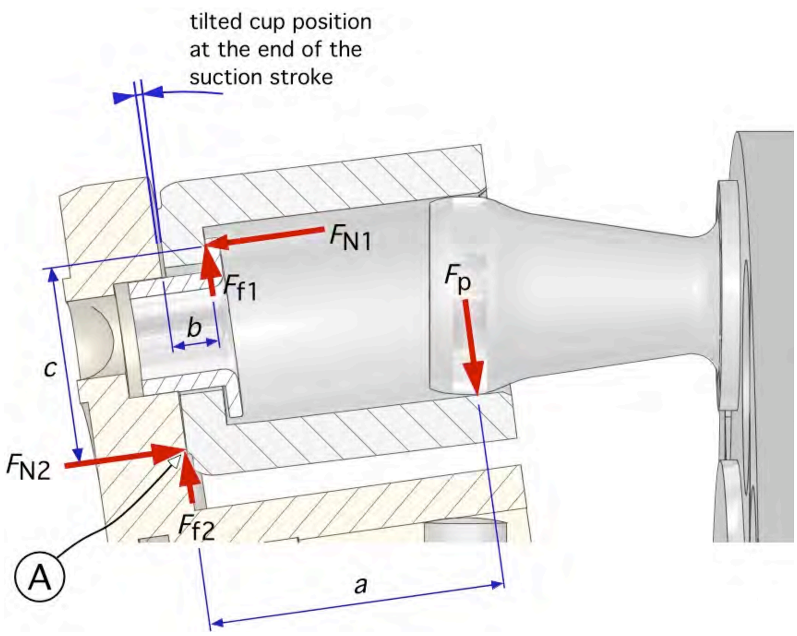


Figure 7: Tilted cup position at the end of the suction stroke

This is especially the case during the suction stroke when there is no pressure available to press the cups against the barrel plate. Due to the centrifugal forces the cup will be in a slanted position at the end of the suction stroke. The holding plug will prevent the cups from turning away completely. At the beginning of the pump stroke the cups have to get positioned again to the barrel plate. The friction between the cup and the barrel on the one hand and the friction between the cup and the holding plug on the other are counteracting this repositioning. The torque equation around point A (in figure 7) and the force equations in the axial and radial direction result in the following requirement for avoiding a lock-up of the cup movement:

$$\frac{a}{2 \cdot c - b} > \mu$$

In this equation, the friction coefficient μ between the cup and the barrel on the one hand and between the cup and the holding plug on the other are assumed to be equal.

The dimensions a, b and c are related to the piston stroke s and piston diameter D. Assuming the following relationships:

$$a = D$$
$$b = 0.2 \cdot s$$
$$c = 1.1 \cdot s$$

and

$$\mu = 0.2$$

it can be calculated that:

$$D > 0.4 \cdot s$$

To stay on the safe side the piston diameter D should not be smaller than halve the piston stroke s. However, giving a barrel angle of around 10° the bore to stroke ratio is close to 1. The lock-up criterion does therefore not determine the maximum barrel tilt angle. The most important parameter that limits the barrel tilt angle is consequently the piston strength i.e. the diameter of the piston neck.

5. Friction in axial piston machines

The simplest form of a piston pump requires a cylinder, a piston and a valve system. In axial piston the non-return valves have been replaced by a port system in which the port plate forms the stationary part of the valve and the barrel the moving part.

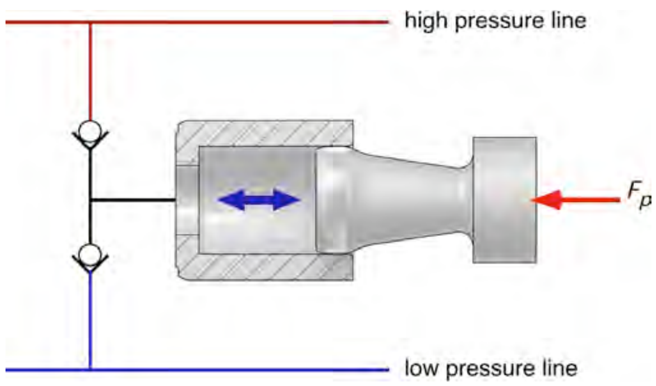
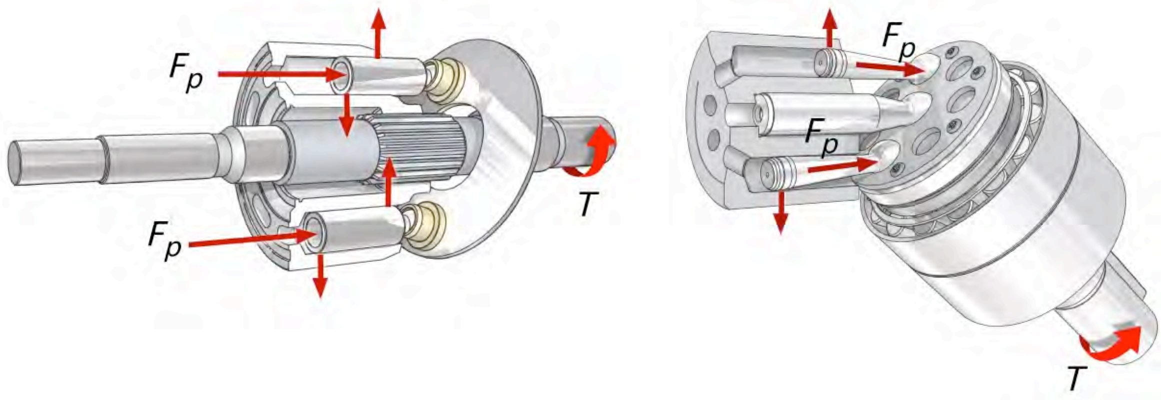


Figure 8: piston pump principle

There are hardly any friction losses in this simple pump configuration. Nevertheless the hydro-mechanical losses in current pumps amount to about 8% or more of the input power for the largest part operating conditions of the pump. The reason for this becomes clear if we consider the way in which torque is converted into hydraulic power in these pumps. In slipper type pumps (figure 9a) the shaft torque is transmitted to the barrel. The torque allows the barrel to drag the piston against the high side forces that are generated by the piston-slipper-mechanism. In case of the bent axis machine, these high side forces are for the most part avoided. Instead the axial piston forces are not anymore hydrostatically balanced and the load has to be taken by the ball joints and the bearings.

In both cases the path of energy conversion from mechanical input power to hydraulic output power involves high contact forces in combination with high sliding velocities; the ideal combination for high friction losses. Apart from friction losses, the high load that current designs put on joints, cylinders and bearings also has detrimental effects on structure borne noise and the durability of these components.



a) slipper type b) bent axis

Figure 9: Main forces current axial piston machines. Only two pistons are shown.

This is fundamentally different in the floating cup design. Figure 10 shows one of the double acting pistons of the floating cup pump, in combination with the accompanying cups. Because of the tilted position of the cups the pistons must have a ball shaped sealing. Consequently the sealing line between the piston and the cup is always standing perpendicular on the cup axis and the cup is fully balanced in all radial directions.

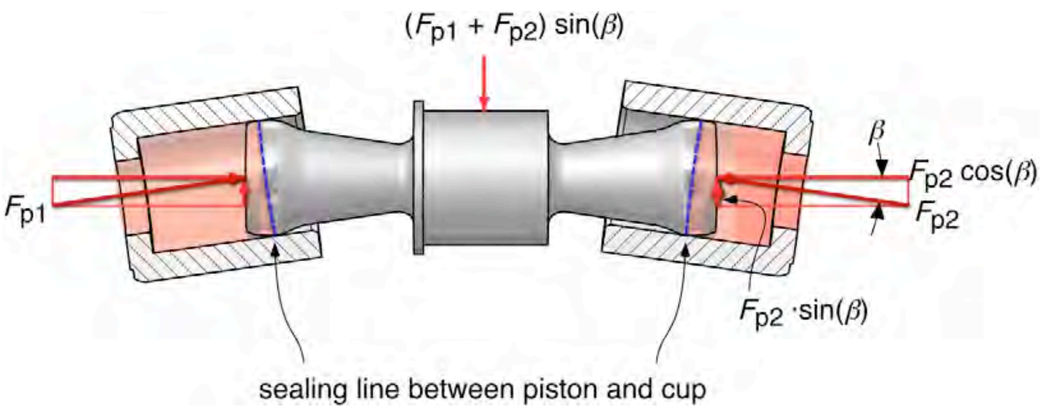


Figure 10: forces acting on a piston pair of the floating cup pump

Conversely the pistons are not balanced, at least not in the radial direction. In the axial direction the hydrostatic forces are entirely balanced as long as the cup pressure on the left side equals the cup pressure on the right side (which is the case for most of the rotor rotation). But in the radial direction the pistons are not hydrostatically balanced and a pressure force is generated on the piston. This is the force that in the end creates the torque on the shaft (see figure 11):

$$T = \frac{R \cdot z}{2\pi} \int_0^{2\pi} F_p \cdot \sin(\beta) \cdot \sin(\varphi) \; d\varphi$$
$$= \tfrac{1}{4} D^2 \cdot R \cdot \Delta p \cdot z \cdot \sin(\beta)$$

In this equation D is the piston diameter, R is the radius of the piston pitch circle, Δp is the pressure difference across the pump and z is the total number of pistons. This is the same basic equation as for a bent axis machine. Port opening and closing effects as well as compressibility effects and other torque losses are neglected in this simplified torque calculation.

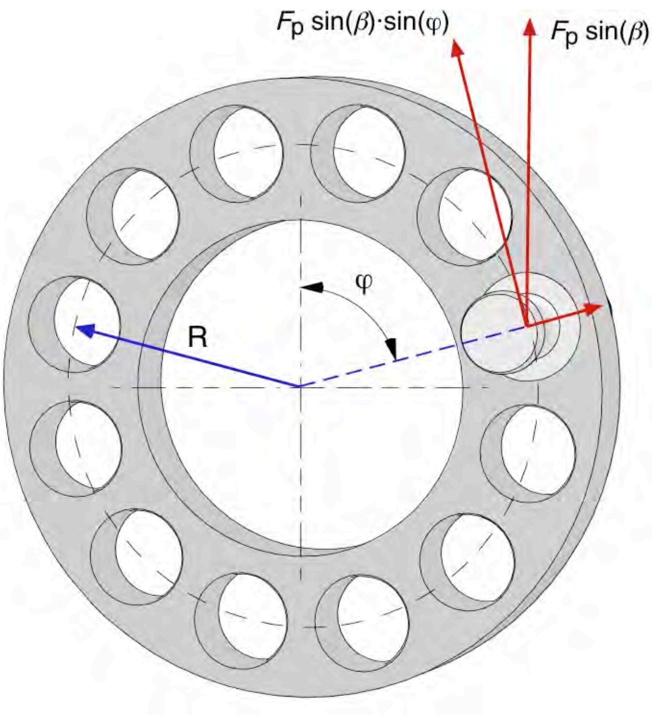


Figure 11: Piston forces acting on the rotor

The floating cup principle combines the advantages of the slipper type and the bent axis machine. It avoids the high bearing load of the bent axis machine by introducing a hydrostatically balanced construction. In this respect the floating cup pump is like a double slipper type pump, having the tilt angle of the swash plate divided in two and having a pump action on both sides of each piston. But it also avoids the high radial force on the sliding contact between the piston and the cylinder. In that respect it is more like a bent axis machine.

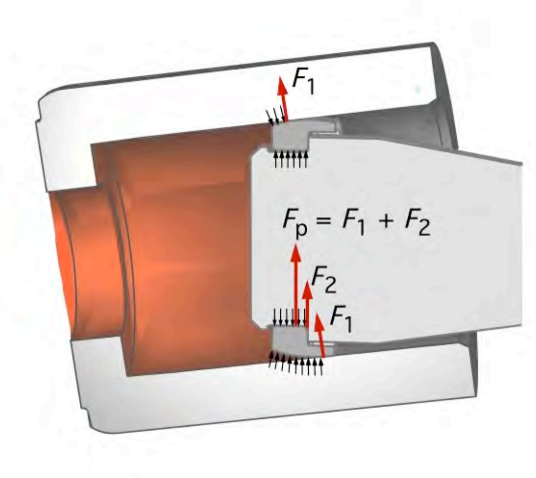
6. Piston sealing

The principle advantage of the bent axis concept is that the pistons have a ball joint in the drive shaft. This avoids the high radial force between the piston and the cylinder of a slipper type unit. But there are still considerable contact forces between the piston and the cylinder, also in the bent axis design. First of all, the pistons in a bent axis pump are not completely in line with the cylinders and as a result the hydrostatic piston load also has a radial component. Added to this are the centrifugal forces. But more important are the forces that are created by the piston rings.

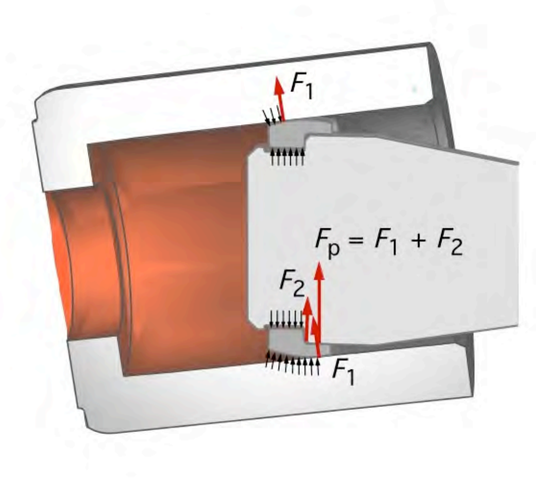
Pistons rings are not completely balanced. They are flexible elements that are pressed against the cylinder wall by the hydraulic pressure as well as by the spring action of the ring. The design of a piston ring is always a compromise between a good sealing function on the one hand and low friction behaviour on the other [5]. But in the floating cup design the requirements are even more difficult:

- the high number of pistons implies also a high number of piston ring slots
- the relatively large angle between the piston and the cylinder axis results for most piston ring designs in higher friction and volumetric losses [5]

To fulfil the specific requirements of the floating cup principle a new type of piston ring has been introduced [2, 3]. Since the cups are free to move in the radial direction, any radial load will be transferred to the piston. The point of contact between the piston ring and the piston can be selected in the design of the ring and the piston. Two design options are shown in figure 12. In the left picture the ring is in contact with the piston at the thick part of the ring. The force F_1 created by the top halve of the piston ring is pushing the entire cup in an upward direction until the cup is stopped by the piston ring and piston combination. The contact between the cup and the ring occurs at the bottom side of the cup where the force F_1 is counteracted by the ring.



a) the ring is supported by the piston
on the inner shoulder of the ring

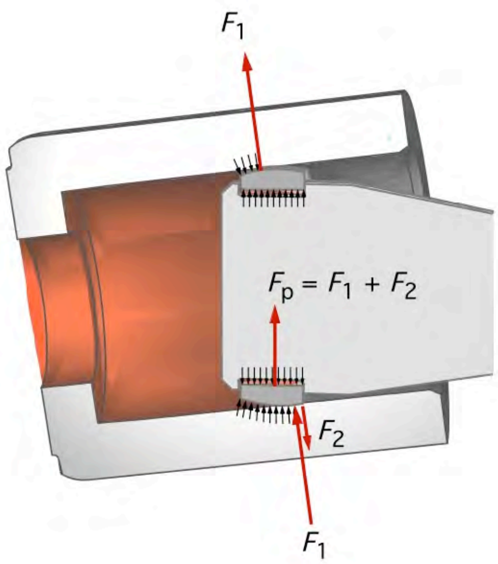


b) the ring is supported by the piston
on the ring skirt

Figure 12: New balanced piston ring design with two design options for transferring the cup and ring forces to the piston

The bottom halve of the ring is also to some extend unbalanced, but the hydrostatic force F_2 is now in the direction of the piston. The sum of F_1 and F_2 is in the end taken by the piston. This is the force that creates the load torque on the drive shaft of the pump. The disadvantage of the left option (figure 12a) is that the contact between the cup and the piston ring is at the thin, unsupported part of the ring, and the force F_1 will create a bending torque on this skirt. Experiments have shown that this in the end increases the width of the piston ring slot, thereby increasing the leakage losses through this gap. The problem can be solved by supporting the piston ring on the skirt, as is shown in figure 12b.

The benefit of the new, stepped piston ring design is the improved ring balance, which reduces the contact force between the piston and the cup, thereby reducing friction and wear. Due to the relatively large tilt angle between the piston and the cylinder of the floating cup machine the application of a conventional piston ring (figure 13a) would result in much larger friction forces.



a) conventional piston ring

b) direct sealing on the piston

Figure 13: Sealing of the piston by means of a conventional piston ring or without a piston ring.

Nonetheless, the best solution, from a friction point of view, is to eliminate the piston ring completely (figure 13b). In that case the hydraulic pressure is directly acting on the piston and the 'detour' over the piston ring is completely avoided. The contact forces between the cup and the piston are limited to the low friction between the cup and the barrel plate and the centrifugal forces of the cup (including the oil contents).

8. Test results

The configuration without piston rings has been tested at the "Institut für fluidtechnische Antriebe" (IFAS) of the Technical University in Aachen. The efficiency measurements have proven the feasibility of the ringless sealing concept: the volumetric efficiency is about the same as of conventional bent axis and slipper type machines. Furthermore the measurements have proven the high hydro-mechanical efficiency of the floating cup concept, thereby giving evidence of the low friction losses.

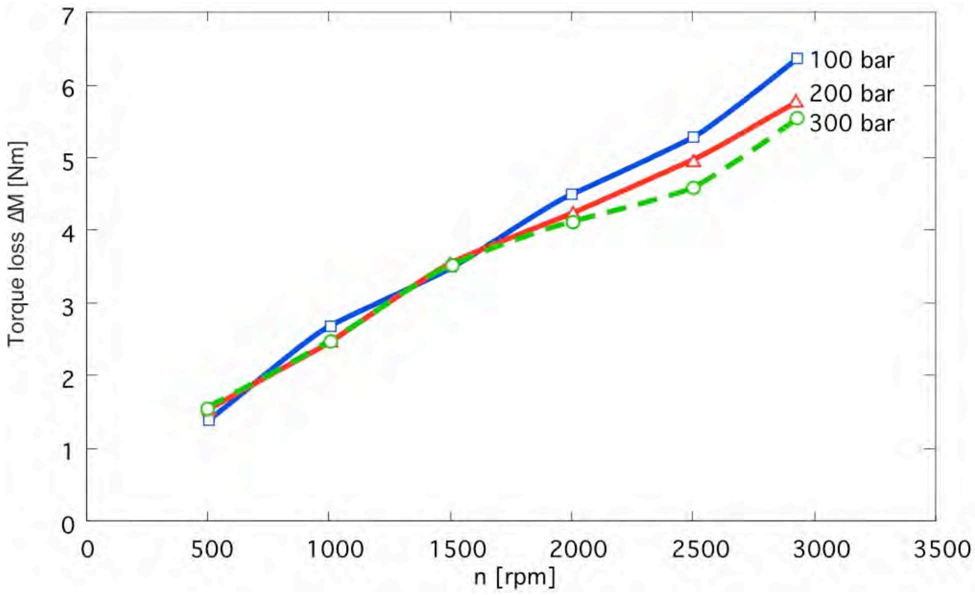


Figure 14: Measured torque losses of the floating cup pump (40°C HLP46)

Part of the results is shown in figure 14. In this diagram the torque losses are calculated from the measured input torque and the calculated theoretical torque. As would have been expected the torque losses are directly proportional to the speed, mainly as a result of the increased viscous losses in the gap between the barrels and the port plates. More remarkable is the small effect of the pump pressure on the torque losses. Figure 15 shows a comparison of the torque losses at a rotational speed of 500 rpm for three different axial piston pumps, all having a constant displacement of 28 cc/rev.

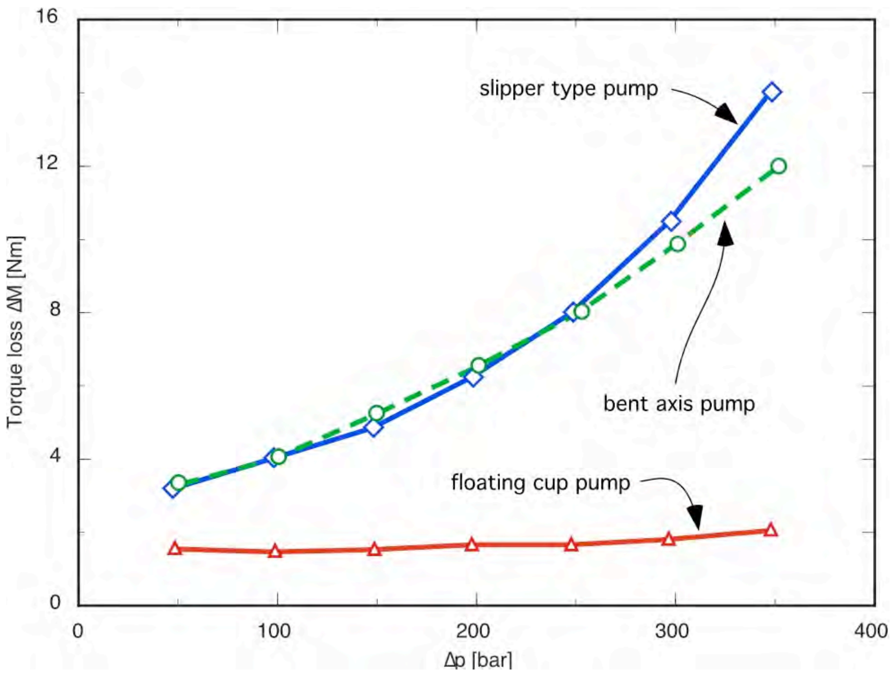


Figure 15: Torque loss for three different constant displacement machines (28 cc/rev) at a rotational speed of 500 rpm (40°C, HLP46)

Whereas the torque losses of current bent axis and slipper type machines strongly increase at higher pump pressures, the torque loss of the floating cup pump almost remains constant. As has been mentioned before, in the floating cup pump the viscous friction between the barrels and the port plates mainly causes the torque loss. Compared to a bent axis pump the floating cup pump has the disadvantage of having two barrels instead of only one. Moreover the diameter of the barrel port seal lands is larger than of a comparable bent axis pump. In this respect the floating cup pump is more like a slipper type pump. Although a conventional slipper type has only one barrel the slippers are also running on a stationary plate and will have similar viscous friction behaviour.

9. The impossible pump

Well-managed companies listen carefully to their customers. If possible they team up with them in the development of new products and solutions to their problems. But the customers are not very demanding. Most of the time their perspective is limited, driven by the problems of today and not those of tomorrow. As a consequence, the view of the manufacturing industry is also rather bounded. This counts also for the hydraulic pump industry. Giving the choice between the troubles they have with current pumps or the unknown problems of a new line of pumps the logical response is to go for certainty. If a pump should be improved, than only within the boundaries of the current concept and the current production facilities.

Considering this, the new floating cup pump is an impossible pump. Although it improves the efficiency and strongly reduces the pressure pulsations, and despite the good perspectives for noise reduction and lowering the production costs, it is clear that the new pump does not fulfil the unwritten rules of sustaining, incremental product development.

But this last barrier is a weak one, especially for the floating cup concept. The design structure of the floating cup machine is quite conventional. It utilizes the same basic techniques as in current hydrostatic pumps and motors. The mechanism is even less complex and demanding, and the mechanical loads are lower than of bent axis or slipper type machines. In that sense the floating cup concept can even be considered to be an incremental development. Only not for the production technologies it is designed for. If being produced with standard pump manufacturing techniques the floating cup pump will not offer a cost benefit.

The floating cup principle however is specifically designed for being produced by means of deep drawing, extrusion and other mass production technologies. Although largely unknown to the hydraulic pump industry there is nothing new about these methods. Bearings, hydraulic lash adjusters and even automotive pumps have been produced only by means of these methods; they even completely depend on them (try to imagine to produce a bearing completely by machining). The requirements for these components are the same as for hydrostatic pumps and motors: high load capability, close tolerances (often below 1 μm) and smooth, clean surfaces. The non-machining production technologies offer a potential, which is not utilized so far in the hydraulic pump and motor industry, simply because there was no pump concept which could close the gap. The floating cup concept will.

References

1 Clayton Christensen "The Innovator's Dilemma: When New Technologies Cause Great Firms to Fail" (1997) Harvard Business School Press, ISBN 087584585

2 Peter Achten, et al, "Dedicated design of the Hydraulic Transformer", Proc. IFK.3, Vol. 2, IFAS Aachen, ISBN 3-8265-9901-2 (2002), p 233-248

3 Johan van den Oever, et al, "Voruntersuchungen des Floating Cup Axialkolbenprinzips", Proc. 3. Kolloquium Mobilhydraulik, TU Braunschweig (2002)

4 Peter Achten, et al, "Design and testing of an axial piston pump based on the floating cup principle", Proceedings SICFP'03, Vol. 2, Tampere University of Technology, ISBN 952-15-0972-4 (2003) p. 805-820

5 Ingvar Hydrén, "Liniendichtung für Kolben in Axialkolbenmaschinen", Proc. AFK.8, RWTH Aachen (1988)