VOLUMETRIC LOSSES OF A MULTI PISTON FLOATING CUP PUMP

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ABSTRACT

The floating cup principle is a new axial piston concept for hydrostatic machines. Unlike slipper type and bent axis machines, the floating cup design features a double barrel configuration with two rings of pistons. The number of pistons is also much higher than that of conventional pumps and motors. Moreover each piston has a separate cup-like cylinder, which is floating on a rotating barrel plate.

It is obvious that the floating cup principle has more leakage gaps than conventional axial piston pumps and motors. Nevertheless, measurements have already proven that the volumetric losses of the floating cup principle are not higher than of current axial piston principles. This paper describes the design aspects that are related to the control of the volumetric losses in the floating cup design, especially the leakage losses from the gaps between the pistons and the cups.

INTRODUCTION

Axial piston pumps and motors have the highest efficiency of all hydrostatic machines, even at high pressure levels of 40 MPa and higher. Although most losses are friction related, the volumetric losses cannot be neglected. The high volumetric efficiency is especially important for hydrostatic motors in applications where a load must be held at zero speed with a minimum slippage. But also at higher oil temperatures i.e. lower viscosity of the oil, the volumetric losses can become dominant.

This paper discusses the volumetric losses of a new type of axial piston principle, called 'Floating Cup'. The principle is described in a number of recent papers [1-13] and will only be described briefly in the next paragraph.

Volumetric losses can be divided in external and internal leakage losses. According to the ISO-definition for efficiencies of hydrostatic machines [14] the difference between the compressed and the expanded oil is also regarded as a volumetric loss. Yet, since the energy content of the compressed oil does not represent a real energy loss we will not discuss this any further, especially since it is not dependent on the design of a hydrostatic machine.

The internal losses are predominantly caused when a barrel port crosses over from one kidney to the other, especially if there is a negative overlap, which causes a short circuit flow from the high-pressure kidney to the low-pressure kidney.

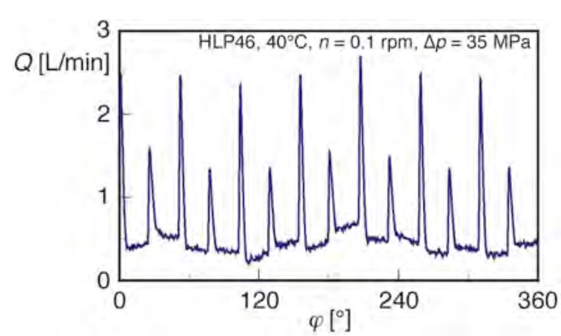

Fig. 1: Cross-port leakage bent axis pump.

The diagram presented in figure 1 shows the leakage of a 28 cc bent axis pump as the Institute of Fluid Power Drives and Controls (IFAS) at Aachen University [15] measured it. The spikes in this graph occur at the rotational positions where a barrel port is passing a top or bottom dead center, thereby creating a short circuit flow from the high pressure port to the low pressure port.

The port-to-port leakage is of course strongly dependent on the dimensions and geometry of the pressure relief grooves of the port plate and is not as much dependent on the pump principle. For that reason this paper will focus only on the leakage flow to the pump house.

FLOATING CUP PRINCIPLE

The core of the new floating cup machine consists of a shaft, a rotor and a ring of about 12 double-sided pistons (see figure 2). The assembled combination can be considered as one single rigid body: there are no sliding joints. The mirrored back-to-back configuration with two rings of pressurized piston surfaces makes the rotor to a large extend hydrostatically balanced.

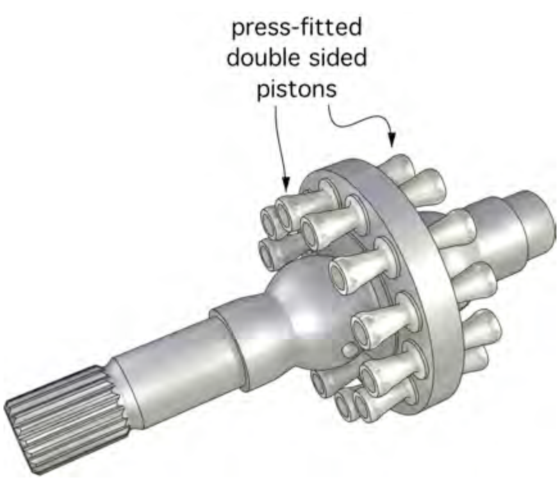

Fig. 2: Shaft, rotor and pistons.

In bent axis and slipper type machines the cylinders are machined in a barrel. Therefore they have a fixed position in the barrel. In the floating cup design however, the cylinders are 'detached' from the barrel (see figure 3): each piston has its own cup-like cylinder.

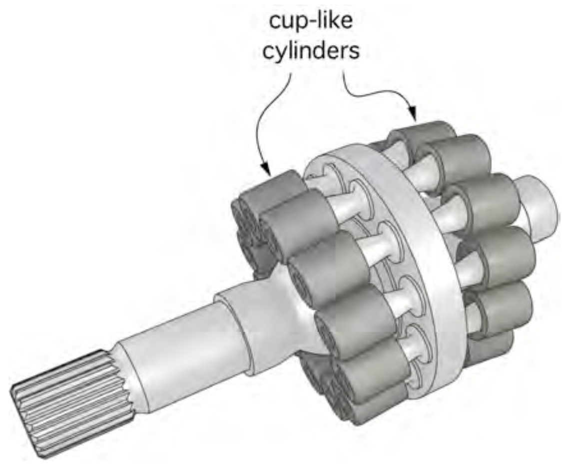

Fig. 3: Cuplike cylinders.

The cylinders or cups are floating on and supported by a barrel, one on each side of the pump (see figure 4). The cups need to be free to move on the barrel plane because of the elliptic deviation caused by the tilted position of the barrel [9]. But the floating position of the cups also eliminates the need for tight tolerances, thereby reducing manufacturing costs.

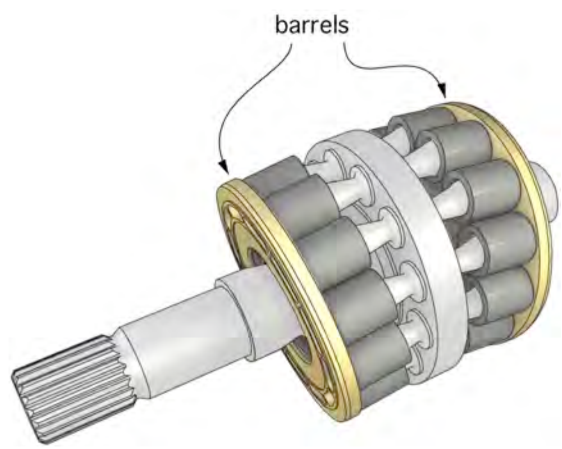

Fig. 4: Rotating group of the floating cup principle.

Figure 5 shows a cross section of the assembly of the rotating group. Only two of the twelve double-sided pistons are shown. The two barrels are positioned in the radial direction by means of a ball shaped extension of the shaft. In the axial direction the barrels are pushed to the port plates by means of a wave spring. When pressurized the hydrostatic balance is chosen as such that the resulting hydrostatic force also helps to push the barrel against the port plate, in a similar way as in conventional axial piston machines.

The barrels are connected to the shaft by means of a simple sliding joint. Since the cups are free to move on the barrel plate they cannot transfer any forces to the barrel. As a result the torque needed to drive the barrel is very low and is limited to the friction between the barrel and the port plate and the torque created by the eventual inertia of the barrel.

An important feature of the floating cup principle is the direct transfer of hydrostatic forces to the piston. The ball shaped piston head always creates a sealing line between the piston and the cup, which stands perpendicular to the centre line of the cup. Therefore the cylinder wall of the cup is completely balanced in the radial direction. The piston however is not balanced and the hydraulic pressure directly creates a radial force on the piston. This force is then transferred to the rotor and from thereon to the shaft, all without any moving linkage.

In order to keep the cups on the surface of the barrel plate, several design options are possible. One solution is to create a slight axial hydrostatic pressure unbalance. This however will only work if the resulting axial force is strong enough to counteract the centrifugal forces acting on the cup. Since the hydrostatic force is pressure dependent, a holding plug (as shown in figure 5) or another type of retainer is needed to keep the cups on the barrel plate at low pressure conditions.

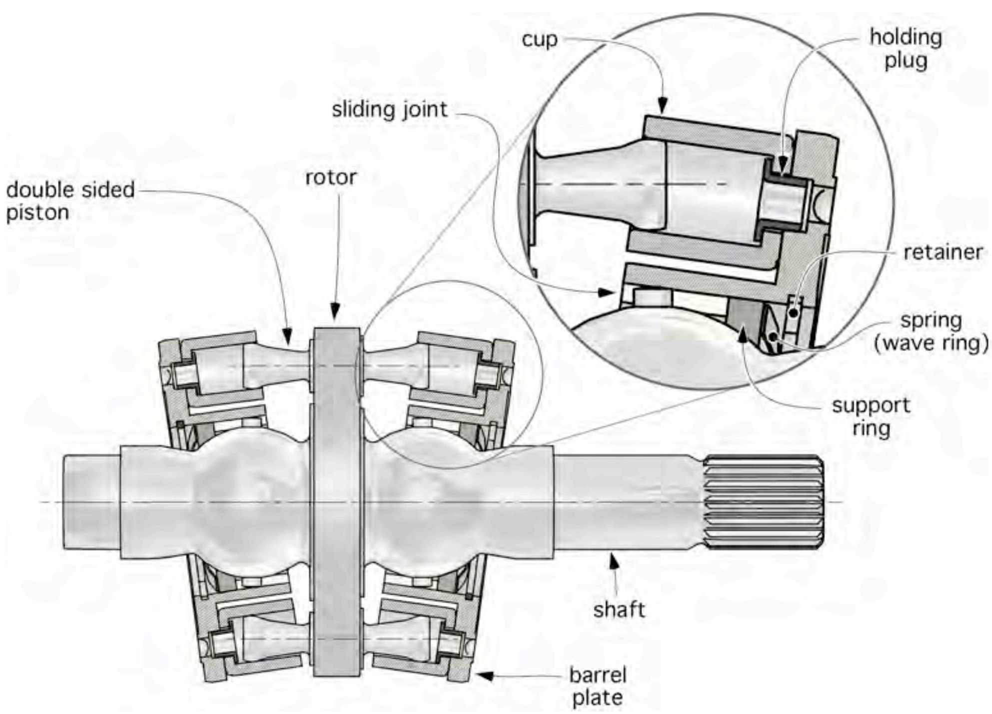

Fig. 5: Cross section of the floating cup principle showing only 2 of the 12 double-sided pistons.

LEAKAGE GAPS

The floating cup principle has a high number of leakage gaps, certainly higher than the average axial piston pump. Figure 6 shows a single piston segment of the floating cup principle illustrating the principle dynamic gaps of the machine; the other gaps, like between the port plate and the house, are stationary and the leakage from these gaps can be neglected.

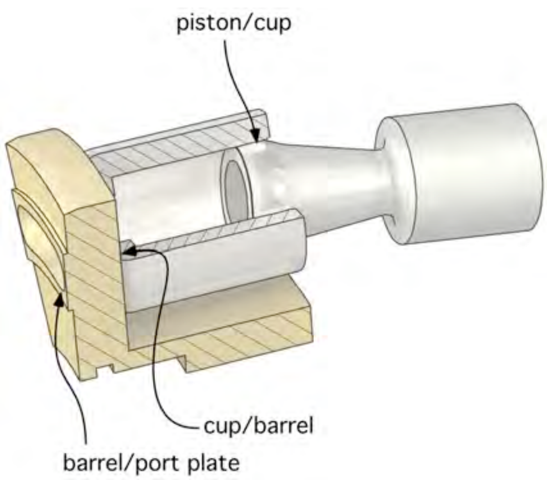

Fig. 6: Leakage gaps of one segment of the floating cup principle.

It is also possible to apply piston rings [2], but compared to a direct sealing the application of piston rings only has disadvantages: higher friction losses, higher leakage and higher costs. Therefore the floating cup designs shown in the previous figures are without piston rings: the ball shaped piston head seals directly on the inner cylinder wall of the cup.

In a similar way as for the floating cup principle, three leakage gaps can be identified for a piston segment of a bent axis machine:

1. Between the barrel and the port plate
2. Between the piston and the piston ring
3. Between the piston and the drive shaft

The second gap also includes the leakage via the slot of the piston ring. Since the piston ring is pressed to the cylinder wall, the circumferential gap between the ring and the cylinder wall is sealed off almost completely. Therefore the piston ring slot is probably the most important source of leakage of the piston ring.

For a swash-plate slipper type machine four leakage gaps per piston segment can be identified:

1. Between the barrel and the port plate
2. Between the piston and the cylinder
3. Between the piston shoe or slipper and the piston
4. Between the slipper and the swash-plate

Assuming a total number of 24 piston segments for the floating cup machine, and 7 and 9 segments for the bent axis and the slipper type principle, a comparison can be made of the total number of leakage gaps (see table 1).

Table 1: Comparison of the number of leakage gaps per machine.

	Gaps per piston segment	Number of piston segments	Total number of gaps
Floating cup	3	24	72 (100%)
Bent axis	3	7	21 (29%)
Slipper type	4	9	36 (50%)

The table above shows what already was obvious: the floating cup principle has 2 to 3.4 more leakage gaps than other axial piston machines. This however does not prove that the volumetric losses of the floating cup principle are also higher. The gaps mentioned above are very different in terms of sealing area and sealing height and many of these factors are still a subject of research, even for conventional axial piston machines [15-21]. This paper will only focus on the three gaps of the floating cup pump mentioned in figure 6, with a special emphasis on the gap between the pistons and the cups.

PISTON SEALING

In the floating cup machine, the spherical piston head combines several functions:

1. Hydraulic power is converted right at the piston head to mechanical power and vice versa.
2. The spherical piston head is also part of the kinematical system. During a barrel rotation the cup makes a combined translating and swiveling motion around the piston. To allow the swiveling motion the piston head must be spherical.
3. And finally the piston head is part of the sealing between the piston and the cylinder.

For sealing the contact between the piston and the cylinder wall, two options have been developed and tested (see figure 7): with, and without a piston ring.

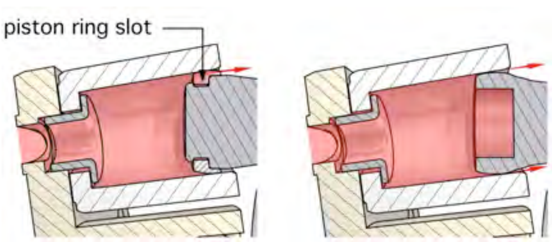

Fig. 7.: piston sealing with or without a piston ring.

Piston rings are delicate components. They have the advantage of being pressurized against the cylinder wall, thereby compensating more or less for deformation of the cylinder due to pressure or temperature effects. However, any enlargement of the outer diameter of the piston ring results in an increased gap at the slot of the piston ring. Depending on the design of the piston ring and the number of piston rings per piston, the slot leakage can be as high as the circumferential leakage around the piston if no piston ring would have been applied. In order to avoid excessive leakage via the piston ring slot, the piston ring must fit precisely in the cylinder. As a consequence piston rings are precision made, expensive components. Furthermore, the radial expansion force of the piston rings creates an extra friction between the piston and the cylinder.

Both the friction and the costs of the piston rings can be avoided if the piston is sealed directly on the cylinder wall. The thin walled cup avoids the temperature differences that occur in the barrel-and-piston constructions of conventional axial piston machines. Furthermore the pistons can be made hollow (as is shown in the right picture of figure 7). By means of this cavity, the piston head expands if pressurized, simultaneously with the pressure expansion of the cup.

FEM-analyses have been performed in order to analyze the deformation of both the cup and the piston. The most important variables to be considered are the wall thickness of the cup, and the diameter and depth of the piston cavity. The FEM-analysis has been performed for the prototype of the floating cup pump, having a piston (head) diameter of 12.2 mm and a maximum pressure load of 40 MPa.

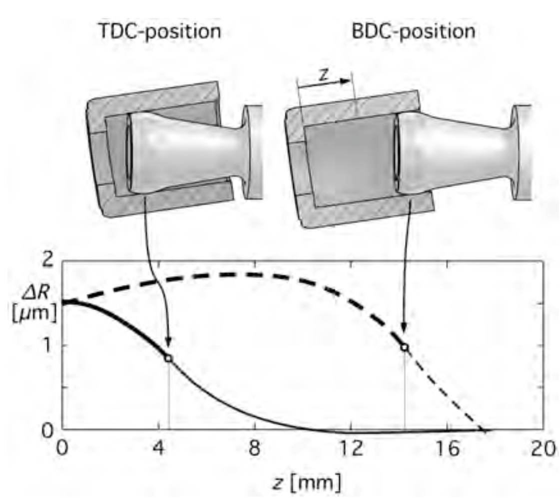

Fig. 8: Radial deformation of the cup due to the internal oil pressure (p = 20 MPa) for the TDC- and BCD-position of the piston. The thick part of the line is where the cup is pressurized. The thin part of the line represents the part where the cup is not pressurized.

The deformation of the cup varies with the piston position. In the Top Dead Centre (TDC) position only a small part of the cylinder is pressurized, whereas in the Bottom Dead Centre (BDC) almost the complete cylinder is in contact with the oil pressure. Figure 8 shows the calculated radial expansion of the cylinder wall for the two extreme piston positions at a pressure level of 20 MPa. In this calculation the cup has a constant wall thickness of 2.95 mm.

For both piston positions the location of the sealing line between the piston and the cup is also indicated in the diagrams of figure 8. Although the deformation of the cup varies with the piston position, the radial expansion is almost constant at the location of the sealing line. At the given conditions the expansion at the sealing line is about 1 μm.

The more or less constant expansion of the cup at the location of the sealing line is important since the expansion of the piston is also independent of the piston position. Now it is possible to tune the diameter and depth of the piston cavity in such a way that the piston expansion follows the cup expansion. For the sealing of a ball in a cylinder, Müller [22] has derived the following dependency of the leakage flow as a function of the gap height:

$$Q_{leak} \equiv h^{2,5}$$

If for instance the radial difference h between the piston head and the cup cylinder would increase from 2 to 3 μm, this would almost triple the leakage flow. Since the cup expansion is pressure dependent, the leakage increases exponential. It is not possible to eliminate this increased leakage by applying piston rings. Although the piston ring would expand with the cup, the gap at the piston ring slot would be widened, and as a result the leakage would be increased as well.

A better solution is to have a direct sealing of the piston on the cup cylinder. By means of a cavity in the piston head, the pressure expansion of the cup can be compensated by a similar expansion of the piston.

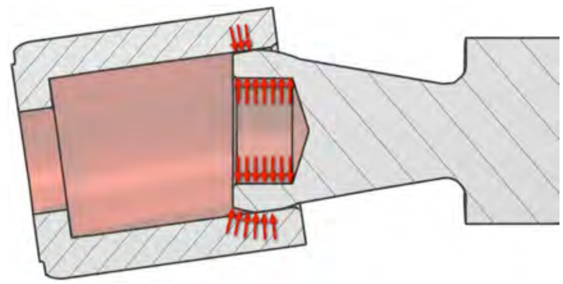

Fig. 9: Hydrostatic load distribution on the piston.

The hydrostatic load on the outside perimeter of the piston head is not rotation symmetrical (see figure 9). Therefore the piston head becomes somewhat oval, having the largest diameter in the plane of the cross section shown in figure 9.

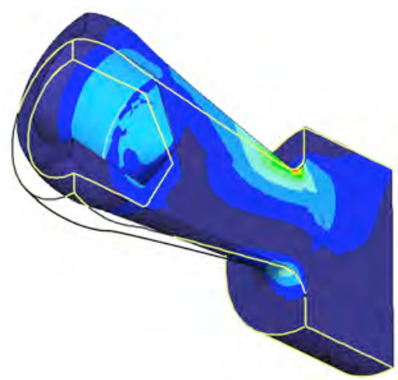

Fig. 10: FEM-analysis of a piston at a cylinder pressure of 20 MPa showing the piston deformation 100x enlarged.

Furthermore, due to the net resulting hydrostatic load on the piston, the whole piston bends upward (figure 10). The bending of the piston does not influence the sealing or the load on the piston since the cups are floating on the barrel plate and will follow the bending of the piston.

The oval deformation ΔR related to the hydrostatic expansion of the piston is shown in figure 11 for different combinations of diameter D_c and depth h_c of the piston cavity. Because the piston expands symmetrical around the vertical axis, only the expansion of the first half of the piston head is shown in the diagram below. From the calculation it can be concluded that the radial variation of the piston expansion is only a second order effect. The variation is largest for large cavities, resulting in a thin walled piston head, which consequently results in a larger variation of the deformation i.e. in a more oval shaped piston head.

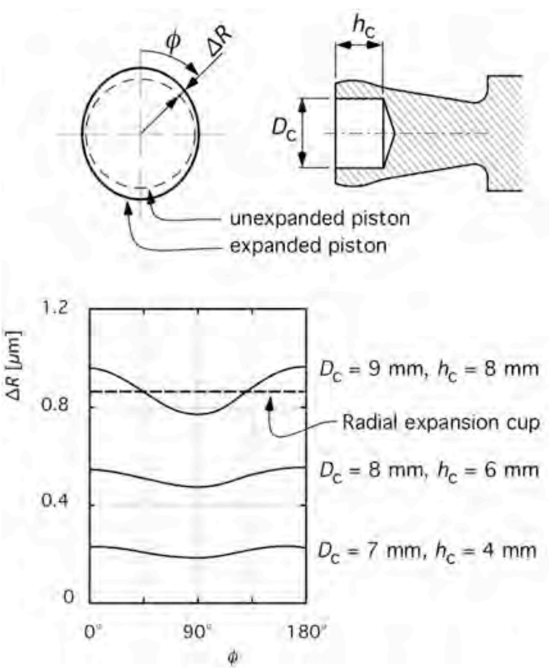

Fig. 11: Calculated radial expansion of the piston (p = 20 MPa) for different combinations of D_c and h_c. The ball shaped piston head has a diameter of 12.2 mm.

The calculated expansion of the cup at the location of the sealing line is also indicated in the diagram of figure 11. According to these calculations, a piston cavity having a diameter of 9 mm and a depth of 8 mm would result in too much radial expansion of the piston, and the piston would get locked in the cup. Therefore, in the prototype of the floating cup, a smaller cavity is chosen (D_c = 8 mm, h_c = 6 mm).

For measuring the leakage flow of each piston cup combination a separate test bed has been constructed. The leakage is measured under dynamic conditions, while the piston is making the same stroke as in the real pump. The stroke frequency is equivalent to a pump speed of about 2 rpm. Both the pressure and the temperature of the oil can be controlled and varied. Furthermore the piston can be positioned under the same angle (i.e. 8°) as in the real pump.

Figure 12 shows the outcome of the test of two pistons:

- solid piston without a piston cavity
- hollow piston in with a cavity having a diameter of 8 mm and a depth of 6 mm.

Both piston-cup combinations are manufactured with a diametral difference at the sealing line between the piston and the cup of about 2 to 3 μm.

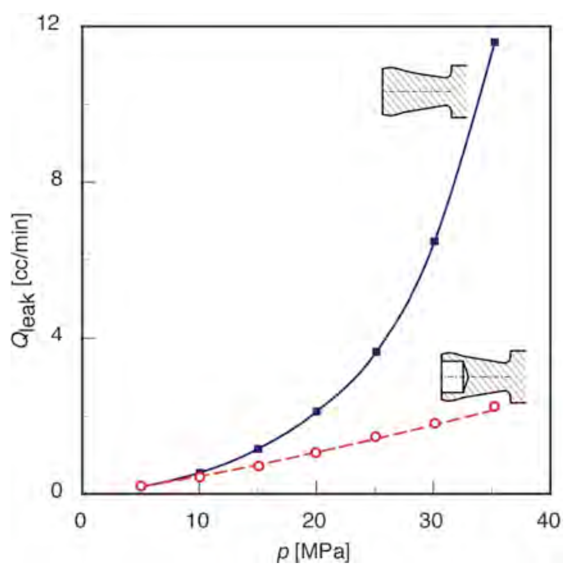

Fig. 12: Measured leakage via the piston sealing as a function of the cylinder pressure p for a solid piston and a hollow piston. (HLP46 at 40°C)

The experiment clearly shows the strong reduction of the leakage if a hollow piston is used. As expected with the solid piston, the leakage increases exponentially due to the expansion of the cylinder wall of the cup and the lack of expansion of the piston. Conversely, with the hollow piston the leakage increases about linearly with the pressure. This indicates that the gap between the piston and the cup does not change and that therefore the cup expansion is compensated by a similar expansion of the piston head.

The reduction may not seem relevant if the leakage flow is compared to the total flow. Even with solid pistons at a pump pressure of 40 MPa, the total leakage of 12 pressurized piston-cup-combinations amounts to 280 cc/min. Compared to the total flow output of a 28 cc pump at 1000 rpm (i.e. 28000 cc/min), the total piston leakage only represents a volumetric loss of 1%. However, this is only true for rather viscous oil (HLP 46 at 40°C having a viscosity of 46 cSt). For many applications the viscosity is much lower. Since the leakage flow between the piston and the cup is laminar the leakage increases strongly at lower viscosity levels. Furthermore, in case of motor operation, the possibility to hold a load with minimum slippage if often necessary. This condition cannot be satisfied with the solid pistons.

PRODUCTION

The direct sealing of the piston head on the cup cylinder requires a small diametrical difference between the piston and the cup of about 3 µm. Combined with the rather high number of components, these strict tolerance requirements provoke the question about the costs of the floating cup principle. Yet, there are many well-known components with the same requirements, being produced at very low manufacturing costs.

Figure 13 shows, as an example, a hydraulic lash adjuster or bucket tappet of an internal combustion engine. As in the floating cup principle, there are many of these components in one single engine. A typical 6-cylinder engine with 4 valves per cylinder requires 24 of these tappets. Inside each tappet there is a small hydraulic piston, a cylinder and a ball valve. These tappets operate under harsh conditions: high accelerations, high load and high temperatures. Nevertheless they need to be very reliable. Furthermore, being an automotive component, the manufacturing costs have to be low. This is realized by applying non-machining production technolo–gies in combination with a classification process.

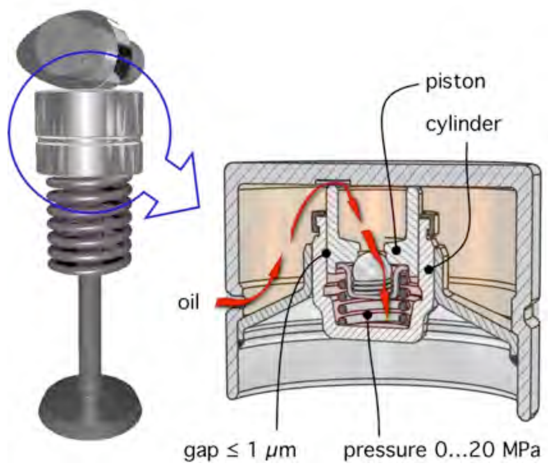

Fig. 13: Cross section of a hydraulic bucket tappet.

The floating cup principle is designed as such that it can be produced by applying the same kind of manufacturing technologies: extrusion, deep drawing, fine blanking, sintering, etc.

DURABILITY

To this point no tests have been performed to verify the durability of the ball shaped pistons. Ball shaped sealings are however not new in hydrostatic pumps. In most bent axis pumps for instance the piston rings have a ball shaped sealing area (see figure 14). Piston rings are hydrostatically unbalanced components. The pressure on top of the piston ring pushes the ring in the axial direction onto the bottom of the piston ring groove. In the radial direction, the outside area of the piston ring is only pressurized as far as the sealing line, whereas on the inner side of the piston ring the hydraulic pressure stands on the full ring area. Consequently the piston ring is expanded and pushed to the cylinder wall.

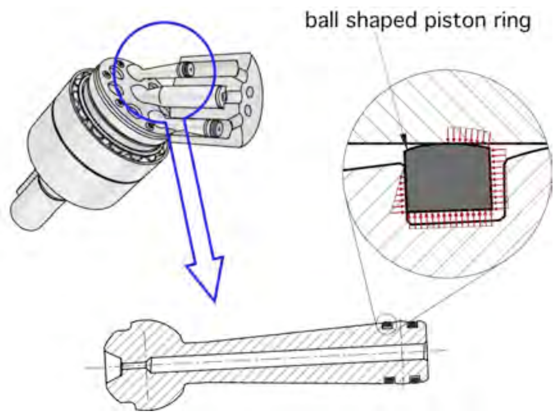

Fig. 14: Spherical piston rings as applied in a bent axis machine.

The expansion of the piston ring creates a substantial friction between the piston and the cylinder wall. Furthermore the slightly tilted position of the piston inside the cylinder creates another radial load on the piston, pushing it to the cylinder wall.

The contact load between the piston and the cylinder is quite different in the floating cup design. Figure 15 shows the radial hydrostatic load acting on the piston and the cup. The ball shaped piston creates a sealing line, which stands perpendicular on the axis of the cup. Consequently the cup is loaded equally in all radial directions and the cup is balanced completely.

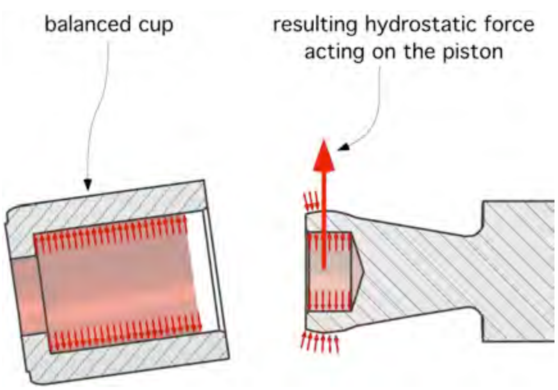

Fig. 15: Radial hydrostatic load on the piston and the cup.

The hydrostatic load on the piston is however not constant in the radial direction. The resulting radial force is transferred to the rotor where it creates the driving torque, together with the other pistons.

Unlike the bent axis design, the hydrostatic force acting on the piston does not create a contact force between the piston and the cylinder. What remains is the low centrifugal force of the cup and the friction force between the cup and the barrel plate. The strong reduction of forces in the sliding contact between the piston and the cup not only results in a high efficiency of the floating cup machine [11, 23], but will also reduce the wear of the cup and the piston.

TEST RESULTS

The case leakage of a floating cup pump has been measured by Innas and compared to the leakage of a state-of-the-art bent axis and a slipper type pump. All pumps have a constant displacement of 28 cc/rev. Figure 16 shows the results of measurements in which the pressure is kept constant at 20 MPa (supply pressure 1 MPa) and the speed is varied.

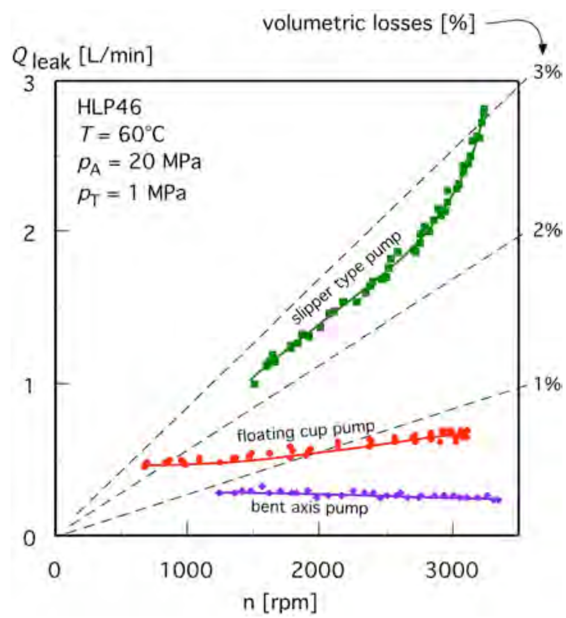

Fig. 16: Measured leakage from the pump house.

The diagram also shows three lines indicating the volumetric losses relative to the maximum theoretical pump output. Of the three pumps, the slipper type pump has the highest leakage. The leakage also increases with the speed and amounts up to about 3% volumetric loss. The bent axis pump has the lowest leakage. This was expected: the bent axis pump has the smallest number of leakage gaps. Also the leakage between the pistons and the drive plate can be neglected since the pistons are pushed strongly into the ball joint sockets of the drive plate.

As has already been shown in table 1 the floating cup pump has twice as much leakage gaps as the slipper type pump. Yet the leakage is much lower, especially at high rotational speeds. Compared to the bent axis pump however, the leakage of the floating cup pump is twice as high. But even then the leakage is still very low and amounts to 1 to 2% of the theoretical output flow.

In order to investigate the influence of the piston leakage all piston-cup-combinations have been tested separately. Figure 17 shows the result for the 12 double pistons applied in the pump for which the total leakage (fig 16) was measured. The average leakage of each double piston equals to 2,5 cc/min at a pressure level of 20 MPa.

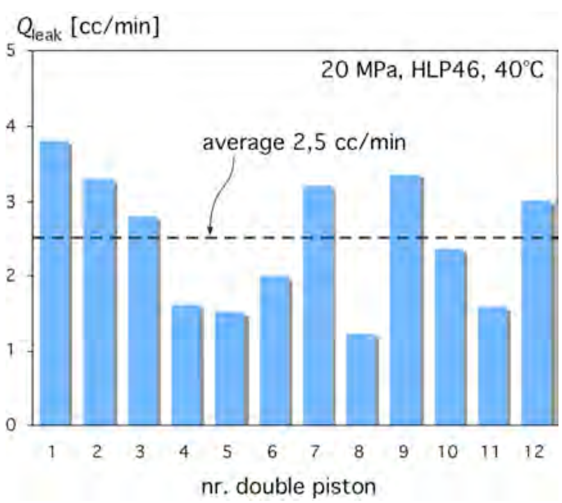

Fig. 17: Measured leakage for each double piston

Taking into account that at 60°C the viscosity is only half of the value at 40°C and assuming a laminar leakage flow around the pistons the total leakage for all 12 double pistons adds up to 60 cc/min or 0,06 L/min at 60°C and 20 MPa. The total leakage of the floating cup pump (as shown in figure 16) is however much higher:

- 0,35 L/min at low rotational speeds
- 0,67 L/min at high rotational speeds.

From these measurements it can be concluded that the piston leakage only contributes to 9 to 17% of the total leakage of the floating cup pump. In other words, 80 to 90% of the external leakage of the floating cup pump has to come from other sources i.e. the leakage between the cups and the barrel plate and the leakage between the barrels and the port plates.

CONCLUSION

The new floating cup principle has a high number of leakage gaps. Compared to an equivalent slipper type unit the number of gaps is twice as high. In comparison with a bent axis machine the number of gaps is even more than three times as high. Another difference between the floating cup principle and the other axial piston principles is the method of sealing the pistons against the cylinders. Like in bent axis machines, the pistons in floating cup machines have a spherical piston head. But unlike bent axis machines, the floating cup principle can refrain from using piston rings. As an alternative, the piston head is made hollow, which allows the expansion of the piston when being pressurized. This would probably not be possible in a bent axis machine due to the different expansion of the barrel cylinders. Furthermore the relatively large barrel mass will also introduce differences in thermal expansion.

Tests have been performed at a prototype floating cup pump having 24 pistons and a total geometrical displacement of 28 cc per revolution. In a separate test bed the leakage of the individual pistons has been measured. At a pump pressure of 20 MPa and an oil temperature of 60°C, the leakage between the piston and the cups amounts up to 0.06 L/min. Compared to a total flow output of the pump of 84 L/min at 3000 rpm, the leakage from the piston-cup-gaps can be almost neglected.

The total leakage from the piston-cup-combinations adds up to 9 to 17% of the total case leakage of the pump. The rest of the leakage has to come from the two remaining sources of leakage: the gaps between the cups and the barrels and the gaps between the barrels and the port plates. Further research is necessary to investigate the leakage from these two remaining sources of leakage. However, it can already be conclu–ded that the leakage of the floating cup pump is within the range of comparable state-of-the-art axial piston pumps.

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ADDITIONAL SOURCES

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

D_c	Diameter piston cavity [mm]
h_c	depth piston cavity [mm]
h	gap height [μm]
n	rotational speed [rpm]
p	pressure [MPa]
Q	flow [L/min]
T	oil temperature [°C]
z	axial position inside the cup [mm]
ΔR	radial cup deformation [μm]
ϕ	angle [°]
φ	angle [°]