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DESIGN AND TESTING OF AN AXIAL PISTON PUMP BASED ON THE FLOATING CUP PRINCIPLE

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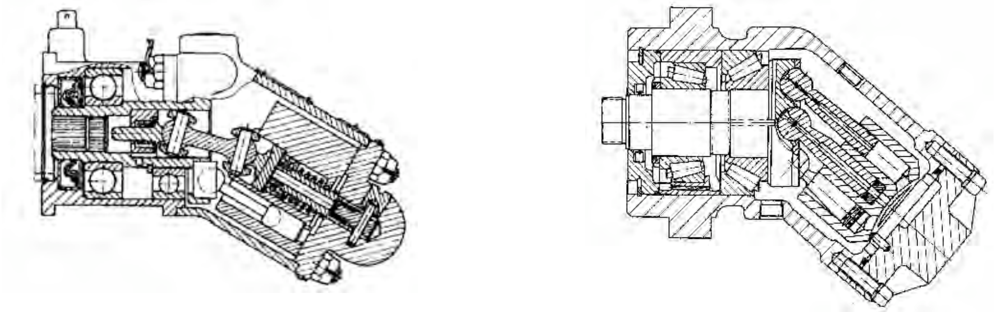
ABSTRACT

A first prototype based on the new floating cup principle has been designed, built and tested. The new pump features a high number of pistons arranged in a double ring, back-to-back configuration. Each piston has a ball shaped end, which is sealing directly on the cylinder wall. Experiments have proven the viability of the new concept. The floating cup principle has demonstrated to be stable in a wide range of pressures and rotational speeds. Furthermore, in a series of tests conducted by the IFAS of the University of Aachen, the efficiency of the floating cup pump was measured. It has been proven that the floating cup pump has a high efficiency in a wide range of operating conditions, with a maximum efficiency of around 97%. In addition, the hydro-mechanical losses are very low at the operating condition of low speeds in combination with high loads. This makes the floating cup principle also very attractive for application in hydrostatic motors. Further research needs to be done especially regarding pulsations, noise and costs. It is expected that the floating cup pump will decrease the pressure pulsations in the output line by a factor of 4 to 5. Moreover a reduction of fluid borne and structure borne noise is expected. Finally, contrary to current axial piston machines, the new pump design can be produced by utilizing modern, low cost production techniques like extrusion and deep drawing.

KEYWORDS: Floating Cup principle, axial piston pump, design, measurements

1. AND NOW FOR SOMETHING COMPLETELY DIFFERENT

Technical progress is mostly achieved by means of small incremental steps. The hydraulic industry is no exception to this rule. As an example figure 1 shows, side by side, the design of a bent axis pump from Thoma (1930) and a modern bent axis pump. Although the similarity between the two designs is striking, there are numerous small design differences, which have resulted in an improvement of performance and characteristics.



First bent axis pump (Thoma 1930)   Modern bent axis pump (Bosch Rexroth)

Fig. 1: Bent axis axial piston machines

Some of these improvements can be seen in the graphs of figure 2, showing the development of variable displacement, bent axis pumps over the years [1]. A closer look at these graphs however reveals that most of the development has occurred in the sixties and seventies. In the last twenty years the development has almost come to a hold.

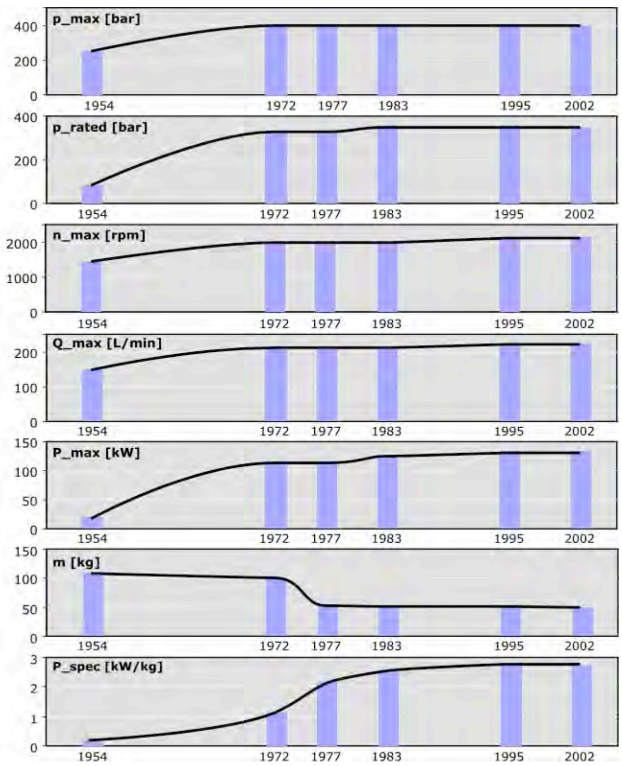


Fig. 2: Development of variable displacement, bent axis pumps [1]

On the other hand, this display of maintaining the status quo could also imply that we are dealing with a very mature technology. In that case the current design of hydraulic pumps and motors is in principle perfect and flawless. The only progress to be expected is in small details like a change of materials or a minor modification of the pressure relieve grooves in the port plate. But if we consider for instance the long struggle for reducing noise and pulsation levels of pumps and motors [2] it becomes clear that more needs to be done than simply detailed variations on the same theme. It's time for something completely different.

2. FLOATING CUP PUMP

On occasion of the Aachener IFK.3 a new axial piston principle for hydrostatic machines was presented [3]. The new principle, called 'floating cup', has the potential to combine the advantages of several different designs of current hydrostatic machines:

- The pressure pulsations are strongly reduced by increasing the number of pistons;
- The bearing load is reduced by introducing a mirrored configuration;
- The increased number of displacement volumes and the mirrored configuration will result in a reduction of structure borne and fluid borne noise;
- The reduced bearing load, the reduced contact force between pistons and cylinders and the short stroke-to-bore-ratio will improve the hydro-mechanical efficiency;
- The short piston stroke will improve the self-priming performance;
- The increased number of pistons reduces the torque variations, which will improve the start-up behaviour of hydraulic motors and transformers;
- Contrary to bent-axis units the new principle offers the possibility of a through drive;
- The new principle can be largely produced by means of deep drawing, extrusion and fine blanking, thereby reducing the production costs of hydrostatic machines.

But there are also some concerns. The new principle has a relatively large number of leakage gaps. This could reduce the volumetric efficiency of the machine. Furthermore, the new principle has two barrels and two port plates instead of only one, and the seal lands of the ports have a rather large diameter. This could result in an increased friction and consequently in a reduction of the mechanical efficiency. Moreover it could be argued whether the floating cup principle itself will be stable under all conditions. Finally it is hard if not impossible to make any quantified predictions about the noise level, durability or production costs of the new principle if these predictions are only based on theoretical analysis.

The only convincing way to determine the characteristics of the new principle is to design a prototype and test it. In this article we will discuss the design and the preliminary experimental results of a first prototype of the floating cup (FC) pump. The prototype and its main parts are illustrated in figures 3 and 4.

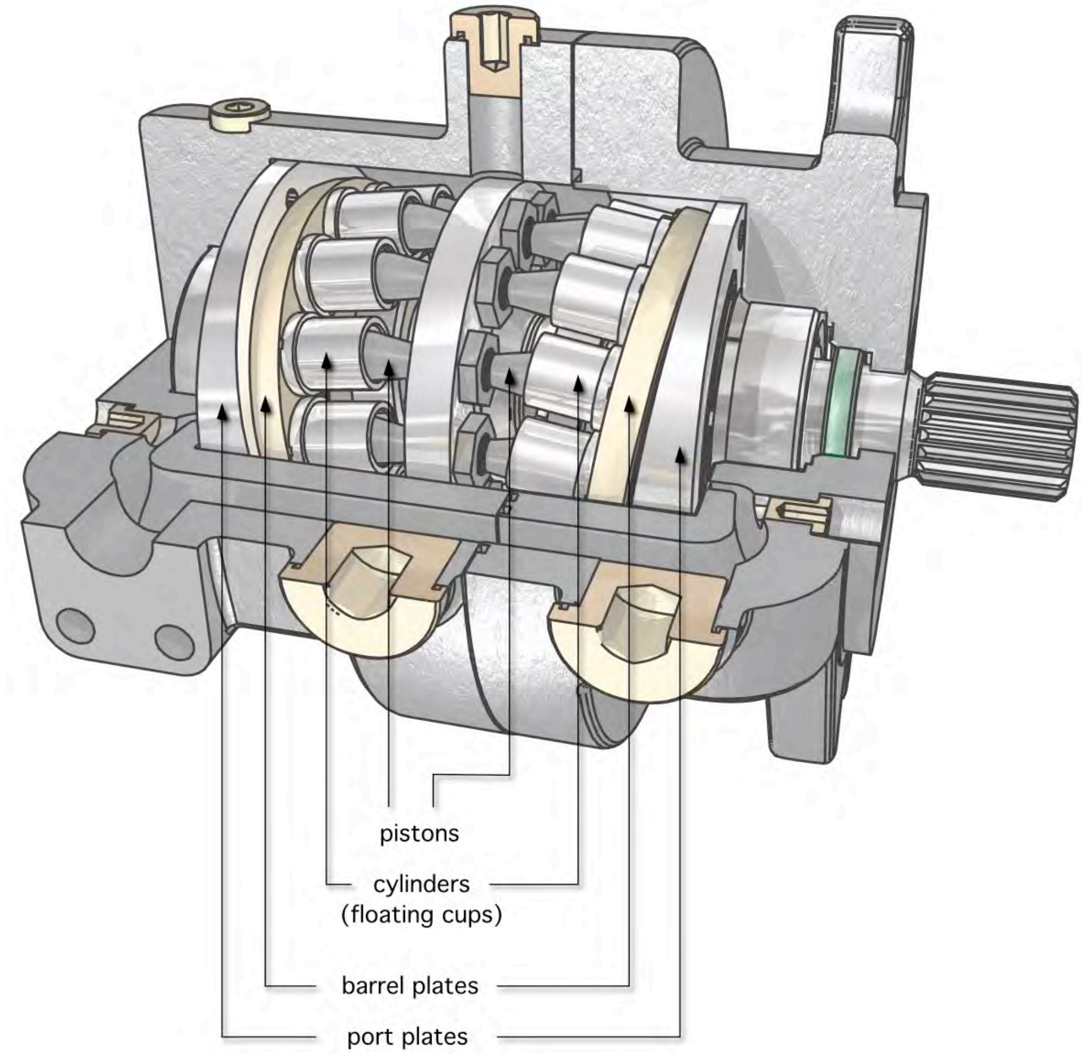


Fig. 3: Cut-away view of the first prototype of a floating cup pump

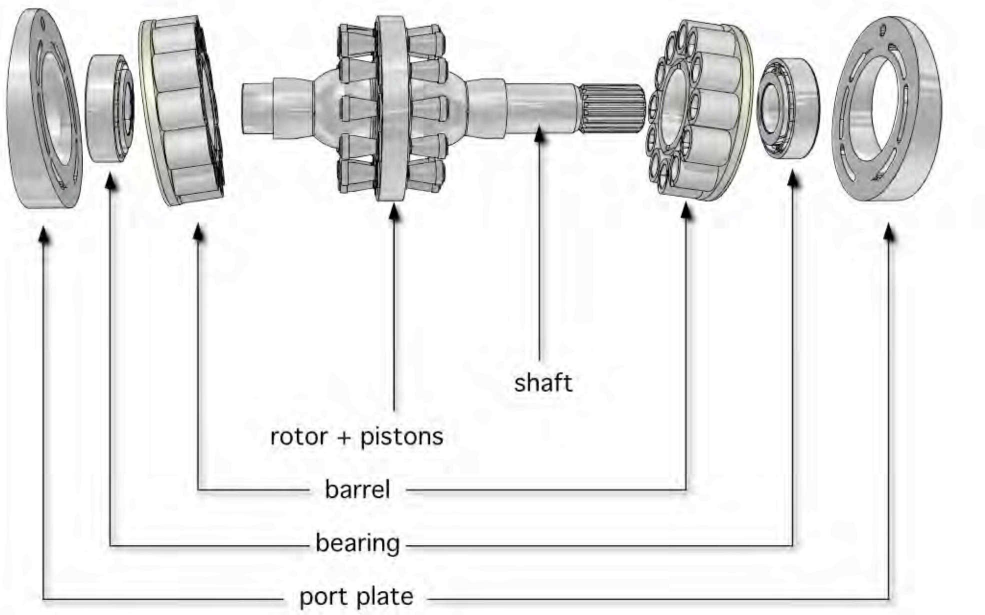


Fig. 4: Exploded view of the main parts of the rotating group of the floating cup pump

The main specifications of the FC-pump prototype are:

- 2 x 12 pistons in an out-of-phase configuration [3] i.e. 24 displacement volumes per revolution
- inner diameter of the cup cylinder: 12,2 mm
- piston stroke: 10,0 mm
- displacement volume = constant = 28 cc/rev at a tilt angle of the port plates of 8°
- maximum pressure 400 bar
- maximum rotational speed ≈ 5000 rpm
- suction side needs to be pressurized to about 2 to 4 bar for rotational speeds above 1500 rpm

Unlike other axial piston pumps, the pistons are rigidly connected to the rotor: there is no linkage between the pistons and the rotor. In the FC-pump, each of the pistons has its own separate cylinder (see figure 5). The cylinders are supported by a rotating disc, which is running on the port plate. This disc, which we call the barrel plate, allows the commutation of the cylinders between the low and high-pressure ports.

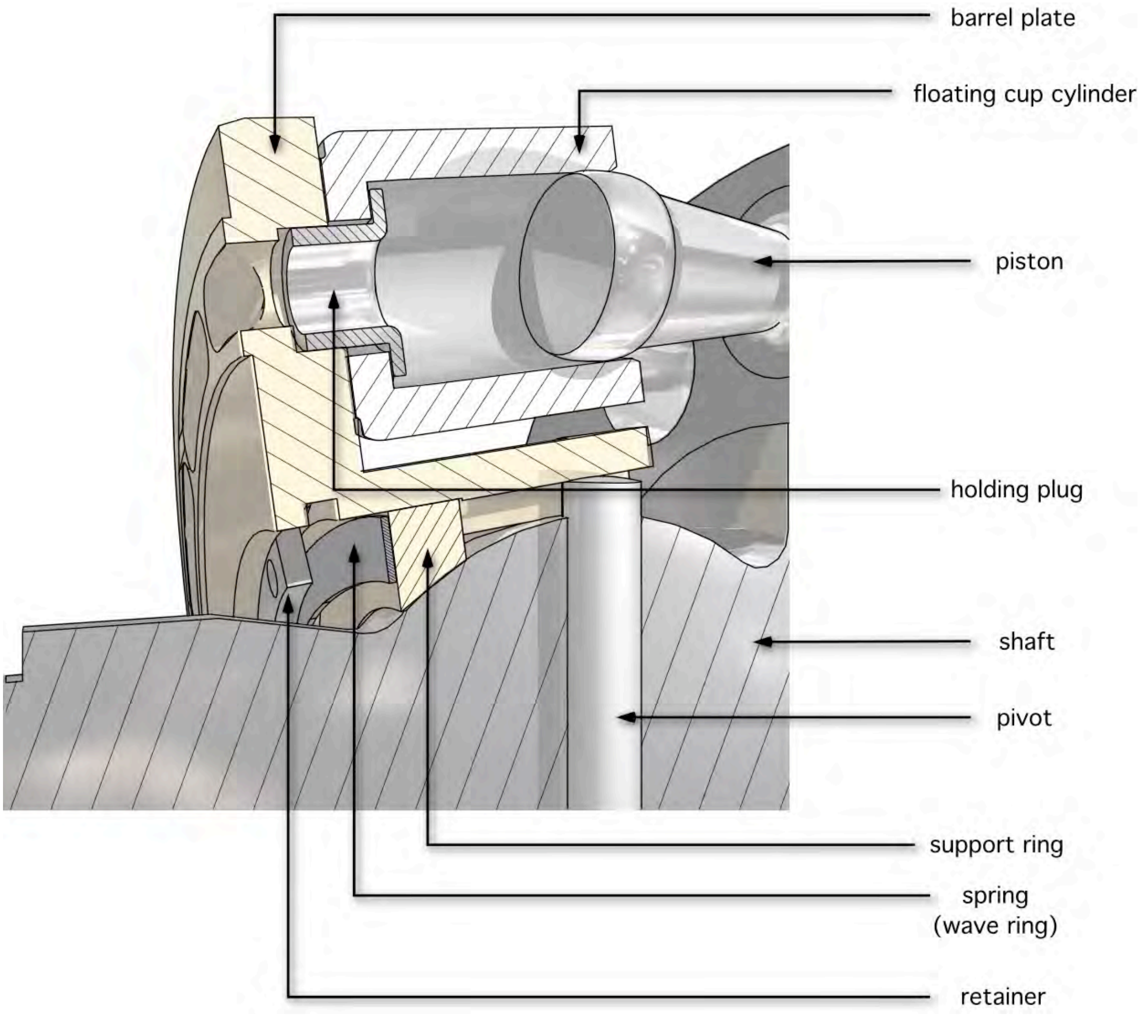


Fig. 5: Detailed cross section of the barrel assembly

The displacement of the pump is realized by positioning the port plates at an inclined angle relative to the rotary plane of the pistons. Due to the angle between the barrel and the rotor, the projected circular movement of the pistons and cylinders will be slightly elliptic on the inclined plane of the barrel plate. The deviation between the circle and the ellipse is taken by the cuplike cylinders, which are floating on the barrel plane. These ‘floating cups’ are balanced in a similar way as the barrel. The dimensions are chosen as such that the cups be slightly pushed against the barrel plane when being pressurized. During the suction stroke the hydraulic pressure will in most cases be insufficient to hold the cups against the barrel plate. In that case the holding plugs will keep the cups in position.

Each of the barrels is sliding on a ball shaped extension of the shaft. The synchronization of the barrel and the shaft rotation is realized by means of a pivot (see figure 5). The load on this universal joint is very low: only the friction forces between the barrel and the port plate and some inertia forces have to be taken by the joint. Unlike in-line pumps and motors there is no hydraulic power transmitted through the barrel or the joint between the barrel and the shaft.

3. PISTON SEALING

Although current axial piston machines suffer from a rather poor hydro-mechanical efficiency their volumetric efficiency is under most conditions higher than 97%. This is especially important for axial piston motors since they often have to hold a load at zero speed. Any leakage within the motor will cause slippage and although the leakage will not be entirely zero it has to be kept at a minimum.

For the new floating cup design with its many leakage gaps, this is special challenge. First of all, the new principle has two barrels and two port plates, and therefore two high-pressure ports. Moreover the ports are relatively long due to the large diameter of the piston pitch circle in the barrel. But if the barrel is well balanced the leakage between the barrel and the port plate is quite small.

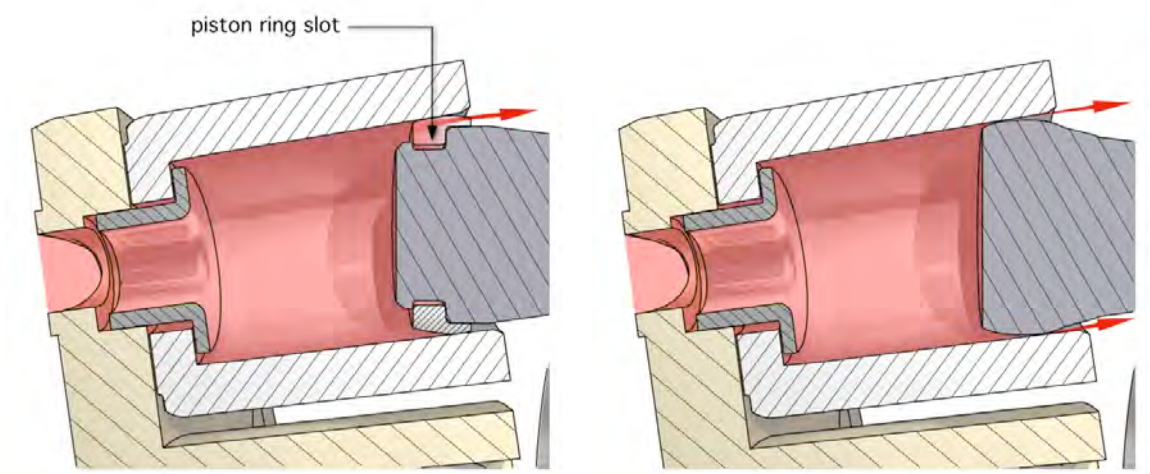
The same can be said about the cups. Similar to the barrel, the floating cups are pushed onto the surface of the barrel plane by means of the hydraulic pressure. However, since the cups and the barrel plate are moving at about the same speed, the cups can be pushed rather strong against the barrel plate and the friction losses (i.e. the product of force and speed) will remain low. In that case the remaining leakage gap between the cup and the barrel plate will be very small and the leakage can be neglected.

The last category of leakage gaps concerns the sealing between the pistons and the cylinders. Since the pistons are directly locked onto the rotor there are no further leakage gaps. This is an advantage compared to bent axis and slipper type machines, which have several leakage gaps in the moving links. On the other hand the floating cup principle features a high number of pistons. It is also quite difficult to apply more than one piston ring per piston. In general it can be said that the second piston ring will half the leakage between piston and cylinder. If we furthermore take into account the higher number of pistons and compare a 24 piston floating cup pump to a 7 piston bent axis unit than the floating cup pump could easily have a leakage which is

$$\tfrac{24}{7} \cdot 2 = 6{,}9$$

times as high (the factor of 2 takes into account the number of piston rings per piston). Furthermore the piston diameter could have an influence on the leakage. But the floating cup principle is characterized by a rather short piston stroke: despite the large number of pistons, the piston diameter is about equal to the piston diameter of a comparable sized bent axis unit. Therefore the length of the sealing gap for each piston does not change.

Finally the kind of sealing between the piston and the cylinder will have an important influence on the leakage. Figure 6 shows the two basic options, with or without a piston ring. In case of the piston ring the leakage will predominantly be influenced by the gap of the piston ring slot. Otherwise the leakage will only be in the circumferential gap between the piston and the cup.



a. sealing by means of a piston ring

b. sealing without a piston ring

Fig. 6: Piston sealing concepts

For the floating cup concept a new piston ring has been designed, which has a stepped cross section. Instead of supporting the piston ring in the axial direction at the bottom, the new piston ring is supported at about the middle of the ring height. In combination with the floating cup this concept allows a good compromise between friction forces and leakage losses [3, 4]. For both sealing options the tolerances are critical. In case of a direct sealing on the piston the leakage is directly dependent to the gap height *h* between the circumference of the ball shaped sealing and the cylinder, and the eccentricity ε (defined as the ratio between the physical eccentricity and the gap height). The equations for calculating the leakage flow for a ball sealing in a cylinder are formulated by Müller [5].

In case a piston ring is used, a larger difference between the outer diameter of the piston ring and the inner diameter of the cup will result in an increased gap of the piston ring slot (assuming the piston ring is expanded sufficiently against the cylinder wall, thereby eliminating the circumferential leakage). In case of equal tolerances both leakage concepts have about the same leakage behaviour. However, if a second or even third piston ring could be used, the piston ring concept is clearly better. But for the floating cup concept, with only one piston ring, a direct sealing on the piston could be beneficial especially if the eccentricity of the ring gap between the piston and the cup could be closer to zero.

Before going deeper into this, it is important to notice that the two sealing concepts have different contact forces between the cup and the sealing element i.e. the piston ring or the piston itself. There are four forces acting on the cup:

- The friction force between the cup and the barrel plate;
- The friction force between piston ring and cup or between piston and cup;
- The centrifugal force;
- Forces generated by the hydraulic pressure.

Assuming that –in the radial direction– the floating cup is a free moving body, all radial forces acting on the cup will have to be taken by the piston. Furthermore the line of contact between the piston and the cup stands perpendicular to the axis of the cup. As a result the cup itself is completely balanced in the radial direction. This is in theory true for both sealing concepts. However the piston ring is not balanced in the radial direction (see figure 7). And since the piston ring is a rather flexible construction element it will transfer the radial forces for a part to the cup ($F_{\text{hyd 1}}$) and for the rest directly to the piston ($F_{\text{hyd 2}}$). As a reaction to the first force, the cup will move upward until the cup is stopped by the combination of piston and piston ring. In this contact the piston ring and the piston will take the radial cup force ($F_{\text{cup}} = F_{\text{hyd 1}}$).

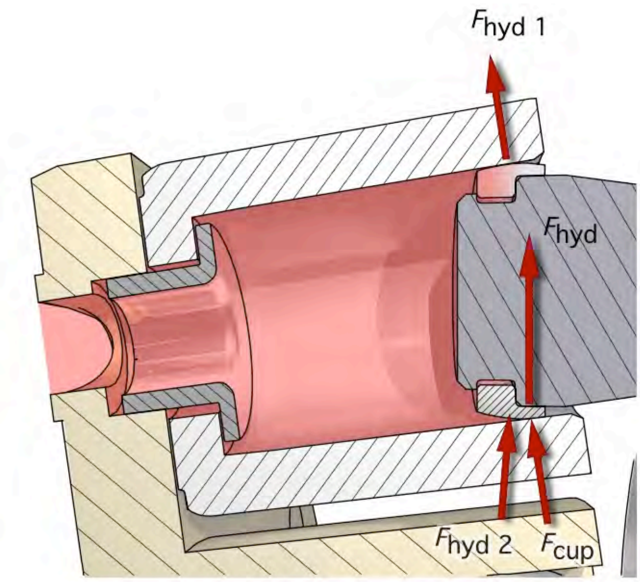


Fig. 7: Radial forces between cup and piston ring caused by the hydraulic pressure inside the cup.

The contact forces $F_{hyd\ 1}$ and F_{cup} will not only increase the wear of the cup and the piston ring but will also create a friction force acting in the direction of the axial movement of the piston. Compared to the other friction force at the contact between cup and barrel plate, and compared to the centrifugal force, the hydraulic forces mentioned above are dominant. At for instance a rotational speed of 3000 rpm the centrifugal force of the cup is about 40 N whereas at a pressure of 350 bar the forces $F_{hyd\ 1}$ and F_{cup} are each around 300 N.

The elemental advantage of a direct sealing on the piston itself, without any piston ring, is that the force generated by the hydraulic pressure is entirely taken by piston itself; the hydraulic contact forces between piston and cup are completely eliminated. Figure 8 shows the ringless concept. The dotted line represents the line of sealing between the piston and the cup. On the bottom part of the piston the unbalanced piston area is indicated.

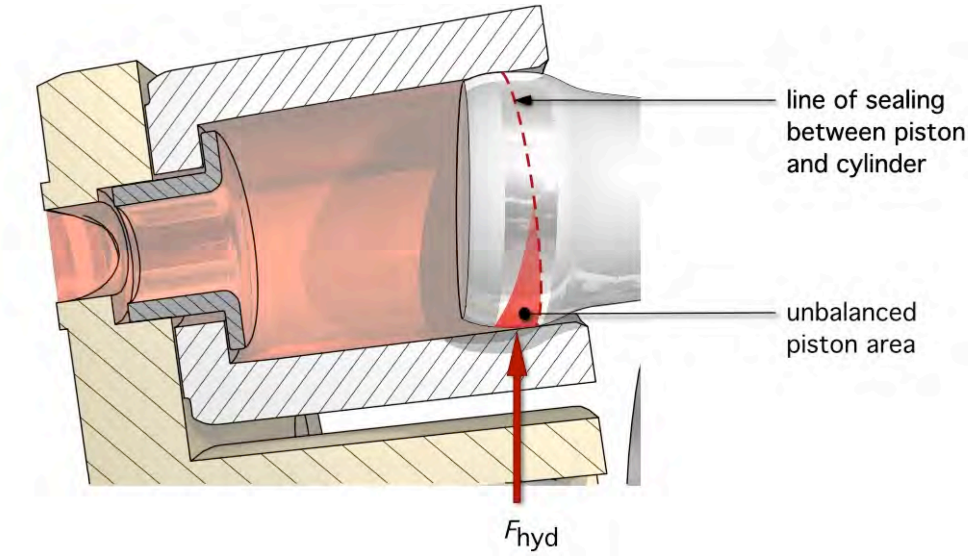


Fig. 8: Sealing concept without a piston ring in which the hydraulic pressure is the direct driving force on the piston i.e the rotor of the pump.

The low contact force will have a beneficial effect on the sealing. Firstly the wear between piston and cup will be strongly reduced and the piston-cup-sealing will remain about its original gap height during the lifetime of the pump. Secondly it is apparent that the piston will no longer be in contact with the cup. The wedge shaped gap between the moving piston and the cup will create an oil layer all around the circumference of the sealing line between piston and cup. As a result the gap will not be fully eccentric ($\varepsilon < 1$) which will reduce the leakage of the piston sealing.

It has to be noted that this will only work if the rotor can take the resulting hydraulic force acting on the piston, i.e. if there is no linkage between the piston and the rotor. In a conventional bent axis pump the pistons are connected to the drive shaft by means of a ball joint. All radial forces have to be taken by the cylinder barrel respectively the bearing structure of the barrel. Moreover the differences in thermal expansion between the pistons and the relatively bulky cylinder block of these machines could cause another problem if the gap between the piston and the cylinder is very small. For these machines a sealing by means of one or –preferably– more piston rings per piston offers the best solution.

For the floating cup concept it can however be concluded that a direct sealing on the piston only has advantages compared to the (single) piston ring. At equal conditions a direct sealing on the piston has less leakage. Furthermore the contact forces between the piston and the cup are under most conditions more than 90% lower if the piston ring is taken away. The ringless sealing will consequently have lower friction losses. Aside from a higher hydro-mechanical efficiency the reduced friction will especially be important for increasing the breakaway torque of the floating cup principle when applied in a motor. Finally a direct sealing on the piston takes away the costs of the piston rings: the production costs, the costs for breaking the rings and the costs for mounting the rings.

4. TOLERANCES

The gap between the piston and the cup is determined by:

- production tolerances
- expansion of the cup as a result of the internal oil pressure

Creating a hole in the piston can compensate the latter to a large extend. In that case both the piston and the cup will expand under the load of the internal oil pressure. Calculations have shown that the cylinder diameter will expand about 2 µm at a pressure of 200 bar. This is the cup expansion at the sealing line between cup and piston. If the piston would be solid instead of hollow the gap height between the piston and the cup would be increased by 1 µm. Although this might seem a small increase it has to be reminded that the leakage will increase to the third power of the gap height.

By far the most important parameters however are the initial tolerances of the internal cup diameter and the ball diameter of the piston head. If the diameter difference between cup and piston could be reduced to 2,5 µm, it can be calculated that the total leakage around all 12 pressurized pistons can be reduced to less than 0,16 L/min (at 200 bar and an oil temperature (HLP46) of 40°C). Measurements at these conditions have resulted in a total leakage flow coming from the cups of 0,21 L/min. Although the measured flow includes the leakage from the gap between the cups and the barrel plates the measurements clearly confirm the calculations. The leakage is now about equal to the total piston leakage of a 28 cc/rev bent axis unit with 7 pistons and two piston rings per piston. The disadvantage of the floating cup pump of having only one sealing line per piston and the higher number of pistons is in this way completely compensated by reducing the gap clearance between piston and cup.

But what are the cost consequences of these tight tolerances? In general precise tolerances result in increased production costs. On the other hand there are many examples of high quality, low cost components that need very precise tolerances.

Perhaps the best-known example is a roller bearing. In order to assure that all rolling elements share the bearing load all parts of the bearing must have an accuracy of around 1 µm. Another example, which resembles the floating cup concept closer, is a hydraulic valve lash adjuster (figure 9).

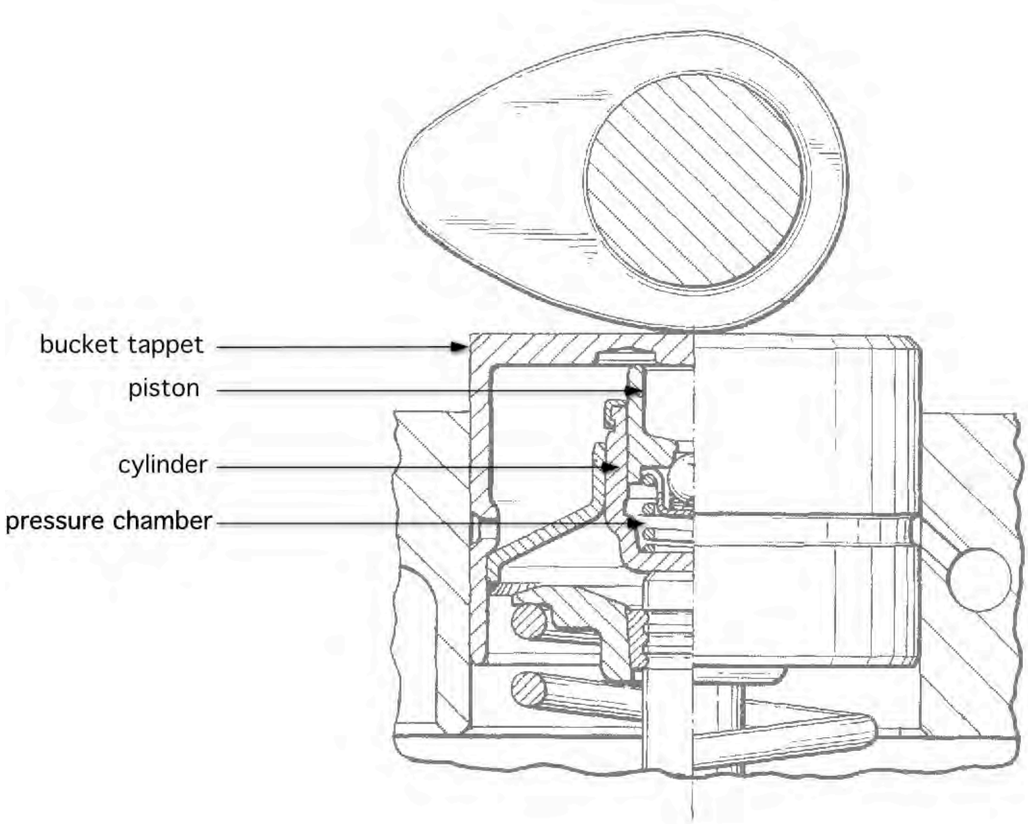


Fig. 9: Bucket tappet for hydraulic valve lash adjustment of intake and exhaust valves of internal combustion engines.

The demands for these tappets are very conflicting. They have to be inexpensive, yet very reliable and durable. The pressure in the pressure chamber varies strongly per revolution of the camshaft and reaches values up to 200 bar [6]. Because of the high acceleration of the engine valve (up to 15.000 m/s^2) the weight of the tappet has to be reduced as much as possible.

For the tappets the required gap height between the piston and the cylinder is 1 µm [7]. This is much more precise than is required for the cylinder and the piston of the floating cup pump. The low cost, high precision fabrication is obtained by combining mass production technologies, like extrusion and deep drawing, with classification of the components in the different tolerance classes before assembly. The classification process requires that the critical dimension of each component has to be measured and that the components have to be stored separately in the different classes. Although the number of classes for the floating cup components will be rather small, the classification process will increase the costs of the components to some extend. But this is easily compensated by the elimination of the costs of the piston rings (which cost more than €1.20 each). The classification is furthermore only possible if the components are produced in rather large quantities. Here the large number of pistons and cups of the floating cup concept are of great importance: for 10.000 pumps with 24 cups and 12 double-sided pistons per pump a production volume of 240.000 cups and 120.000 pistons is required. Finally the design has to be as such that the components can be made by means of low cost, high volume production technologies, like deep drawing and fine blanking. This is possible with the floating cup concept but it is hard to imagine that the cylinder block could also be made by means of these production technologies. Furthermore classification is only possible if you can take the components apart and group them in the different classes. For the cylinder block the question would be how to classify the 7 or 9 cylinders, considering you cannot take them out.

5. MEASUREMENTS

The floating cup concept is in the very early stages of its development. The first prototypes have been designed and built by Innas in 2002 and testing started in September 2002. The experiments, which are presented in this paper, are all performed by the Institut für Fluidtechnische Antriebe und Steuerungen (IFAS) of the University of Aachen. Figure 10 shows a diagram of the setup of the test bench. All measurements are performed at an oil temperature of around 40°C with HLP46 oil.

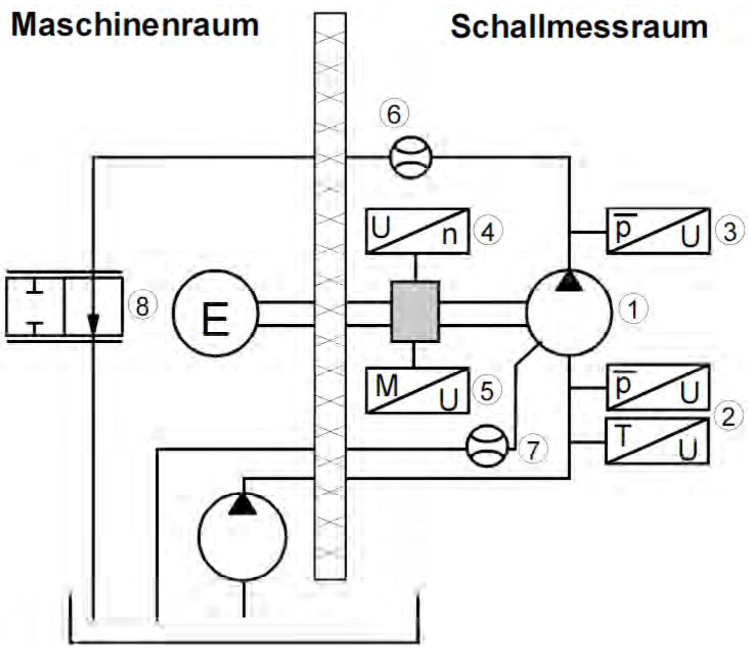


Fig. 10: Diagram of the test bench at IFAS.

1. pump to be tested
2. pressure and temperature sensor in the suction line
3. pressure sensor in the high pressure line
4. rotational speed sensor
5. torque sensor
6. flow sensor supply line
7. flow sensor leakage line
8. load valve

The test pumps are driven by means of an electric motor. The rotational speed can be controlled between 500 and 3000 rpm. The pump pressure is varied between 50 and 350 bar. The suction line is pressurized to a pressure of around 2 bar.

The three diagrams of figure 12 show the volumetric, hydro-mechanical and total efficiency for the floating cup pump. The volumetric efficiency is higher than 96% for the largest part of the measured field of operation. The floating cup pump was tested in a configuration in which the pistons were directly sealing on the cups (see figure 8) i.e. without applying piston rings. The difference in diameter between the pistons and the cups is around 5 µm when not pressurized. The pistons are solid and don't have a pressure compensating cavity. As a result the difference in diameter between the cup and the piston will be larger at higher pressures and the leakage increases more than proportional. Furthermore there are strong indications that most of the leakage is caused by an incorrect balancing of the delicate force and torque balance of the barrels. It is expected that the volumetric losses can be strongly reduced by:

- improvement of the balance of the barrel
- creating a pressure compensating cavity in the piston
- reducing the gap height between the pistons and the cups

But even without these further optimizations it is already clear that a direct sealing of the pistons on the cylinder is feasible. Furthermore the measurements have proven that, despite the high number of sealing gaps, the floating cup concept can have a volumetric efficiency, which is equivalent to that of current bent axis and slipper type pumps.

The second diagram of figure 12 shows the hydro-mechanical efficiency of the floating cup pump prototype in its current status. For the largest part of the field of operation the hydro-mechanical efficiency is higher than 94%. As can be seen in figure 11 the torque losses are governed predominantly by the rotational speed and are almost independent of the pump pressure.

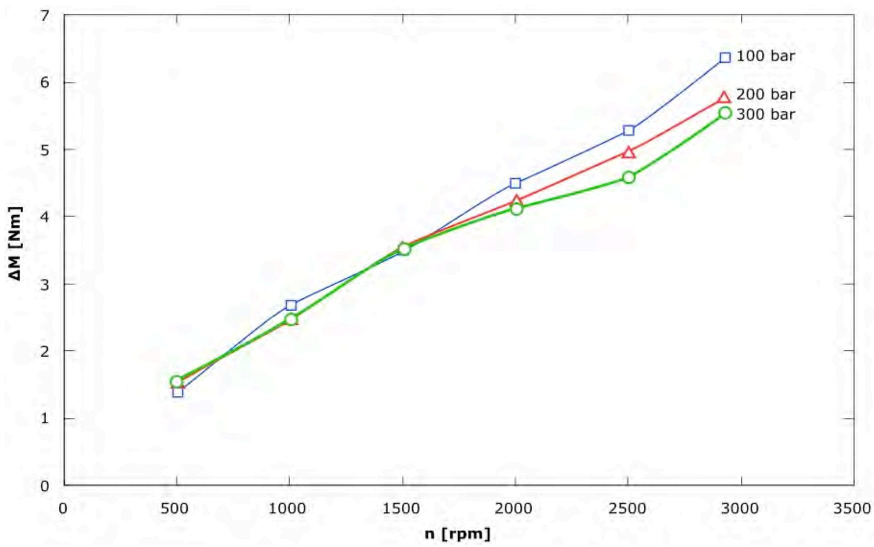


Fig. 11: Measured torque losses of the floating cup pump (40°C HLP46)

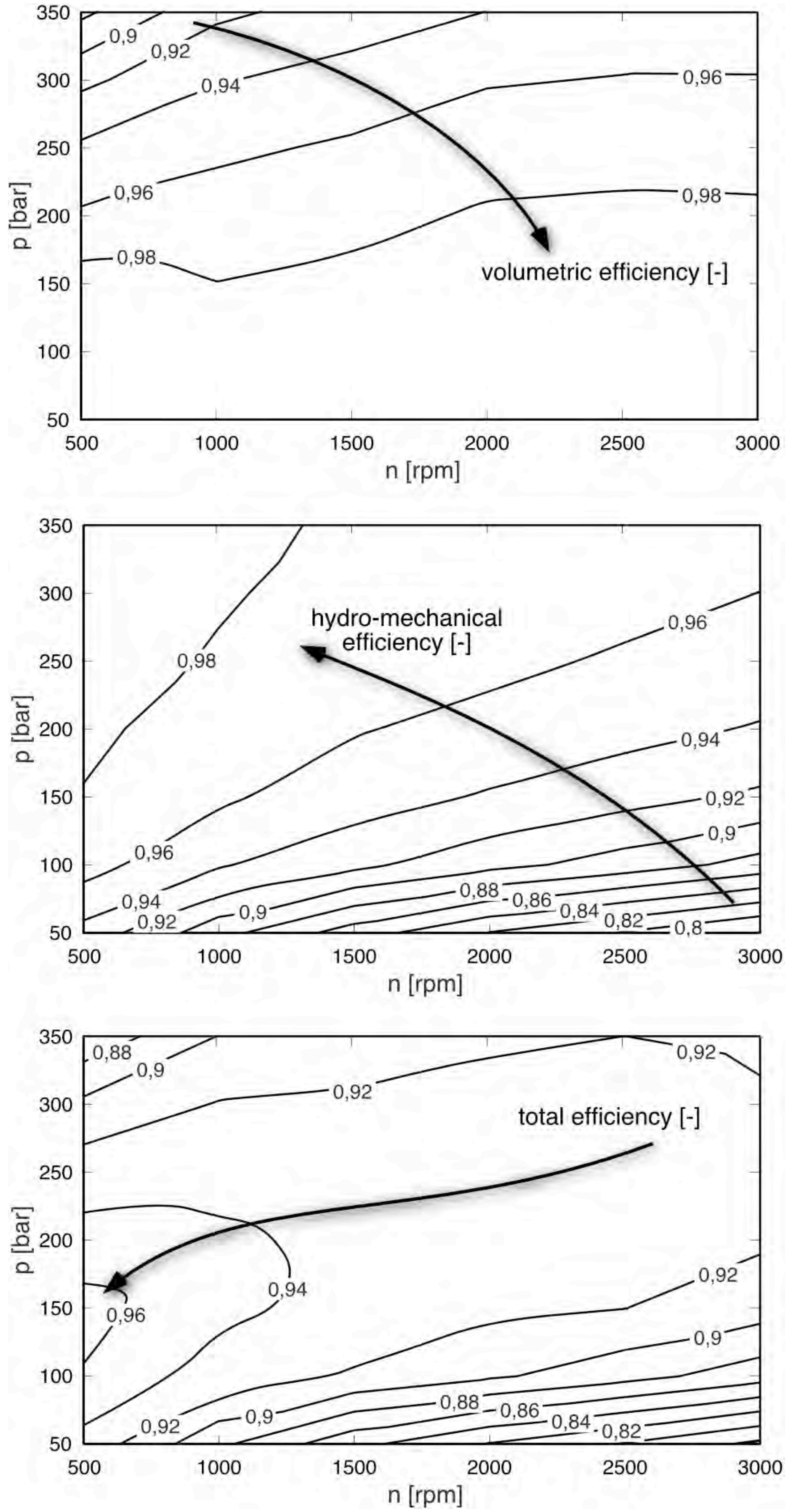


Fig. 12: Efficiency measurements of the floating cup pump (40°C, HLP46)

Compared to conventional axial piston pumps the torque losses are rather small, especially at low speeds and high pump pressures. This advantage is especially favourable if the floating cup principle is applied in hydrostatic motors. Combined with the high number of pistons, the low torque loss at maximum pressure will result in a high breakaway torque and a good start-up behaviour. The measurements also indicate that the double barrel configuration does not result in a reduction of the hydro-mechanical efficiency. The higher friction losses in the double interface between barrels and port plates are apparently more than compensated by the reduced friction between pistons and cups and by the fact that there no other linkages with high friction losses in the floating cup pump. Furthermore the bearing load is reduced to a large extend by means of the mirrored configuration.

The third diagram of figure 12 shows the total efficiency of the three pumps. It is obvious that the combination of an increased hydro-mechanical efficiency and an equal volumetric efficiency results in a high total efficiency of the floating cup pump. In a large area of the tested field of operation the total efficiency is higher than 92%. This condition is also achieved at maximum pump power i.e. at high pump speed and pressure.

6 CONCLUSIONS AND OUTLOOK

A first pump prototype of the new floating cup principle has been designed, built and tested. The tests have shown that the floating cup principle allows a stable operation in a wide range of pressures and rotational speeds. The higher number of leakage gaps of the floating cup principle have not resulted in a reduction of the volumetric efficiency. Even without applying piston rings the volumetric efficiency is as high as that of conventional axial piston pumps. The elimination of the piston rings offers a considerable cost reduction. It can be expected that the required tolerances for a direct sealing between piston and cylinders can be realized cost effectively by means of modern mass production technologies.

The elimination of the piston rings in the floating cup machine also reduces the friction between pistons and cylinders. Furthermore the double ring back-to-back piston configuration reduces the bearing load. This improves the hydro-mechanical efficiency of the pump as has been experimentally verified. In the pressure range between 50 and 350 bar, and the speed range between 500 and 3500 rpm, the average total efficiency of the floating cup is 91% with a maximum efficiency of almost 97%.

It is to be expected that the volumetric efficiency of the pump can still be improved. Furthermore it is expected that the pressure pulsations in the high-pressure output line will be strongly reduced because of the higher number of pistons. The high number of pistons and the back-to-back mirrored configuration will probably also result in a reduction of fluid and structure borne noise.

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