SAE 2004-01-2653

Efficiency and Low Speed Behavior of the Floating Cup Pump

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ABSTRACT

The floating cup principle is a new axial piston concept for hydrostatic machines. It features a high number of pistons, arranged in a double ring, back-to-back configuration. Furthermore the pistons are locked onto a central rotor and each piston has its own cuplike cylinder. These 'cups' are floating on and supported by a barrel plate. The pistons have a ball shaped crown, which is sealing directly on the cylinder without a piston ring.

A first prototype of the new pump has been built and tested. For comparison a state-of-the-art slipper type pump and a bent axis pump (both constant displacement, 28 cc/rev) have been tested as well. The steady-state performance tests have proven the high efficiency of the floating cup principle. The low speed tests, during which the pumps are tested as a motor, have confirmed the low friction losses and high starting torque of the floating cup principle. Furthermore the high number of pistons strongly reduces the torque variations. This paper describes and analyzes the outcome of the measurements.

INTRODUCTION

In a time in which 'better is just not good enough', the fluid power industry has concluded that it has failed, because it has failed to innovate [1]:

- "During the last few decades the fluid power market did not see any revolutionary product, which improves the efficiency of the end product. (…) The fluid power industry is facing a self-created competition from electric technology."
- "Fluid power manufacturers must address three primary problem areas that still plague the industry: leakage, noise and reliability. These problems can be solved, but fluid power designers must make a serious effort in this regard to stop the erosion of market share."
- "The fluid power industry has not seen investment for technology advancement since the sixties. All major manufacturers, such as, Vickers-Eaton, Bosch-Rexroth, Denison, Parker have not put in enough for research and development, to come out with futuristic revolutionary pumps, motors and controls. The industry in 2003, is using mid sixties technology."

Yet, hydraulic pumps and motors are not by definition too expensive, too noisy, or too inefficient. External gear pumps for instance are inexpensive. Internal gear pumps on the other hand feature a low noise level and screw type pumps offer a low pulsation level. Hydrostatic machines can also have a high efficiency. Bent axis pumps and motors for instance have an energetic efficiency of up to 96%. Several hydrostatic machines also provide the possibility of a through drive and some can even be made variable.

The problem however is that the hydraulic industry does not offer pumps or motors in which these characteristics are combined: high efficiency, low cost, low noise and low pulsations, high start-up efficiency and good low-speed behavior. Optionally the pump or motor should also be variable and offer a through drive. As long as the hydraulic industry does not present such a machine on the market, hydraulic systems are confined to applications for which the detrimental effects are not outweighing the other, remaining advantages.

The new 'floating cup' principle is designed to change this situation. Earlier design studies and theoretical analysis have already indicated the potential with respect to efficiency, pulsations and power density [2-12]. Meanwhile a first pump prototype has been built, and tested by the Institute of Fluid Power Drives and Controls (IFAS) at Aachen University. In this paper the results of the efficiency and the low-speed tests are presented. Aside from the floating cup pump, two state-of-the-art axial piston pumps –a bent axis and a slipper type pump– have been tested as well.

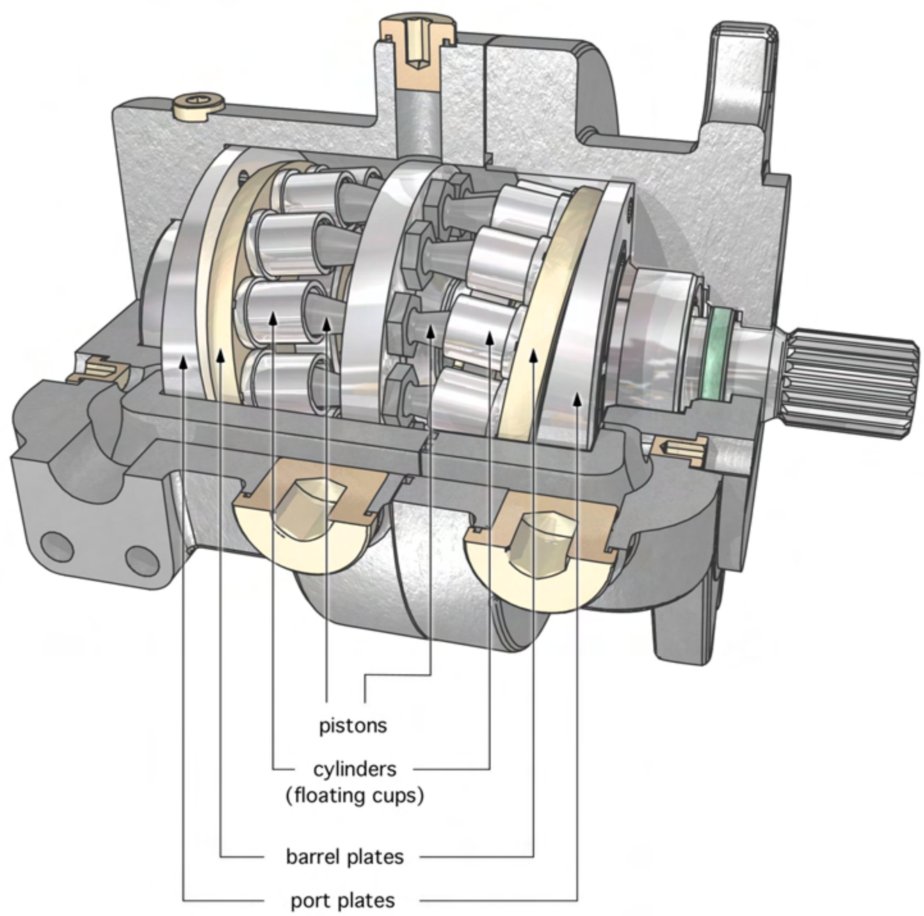


Fig. 1: Cut-away view of the first prototype of a floating cup pump

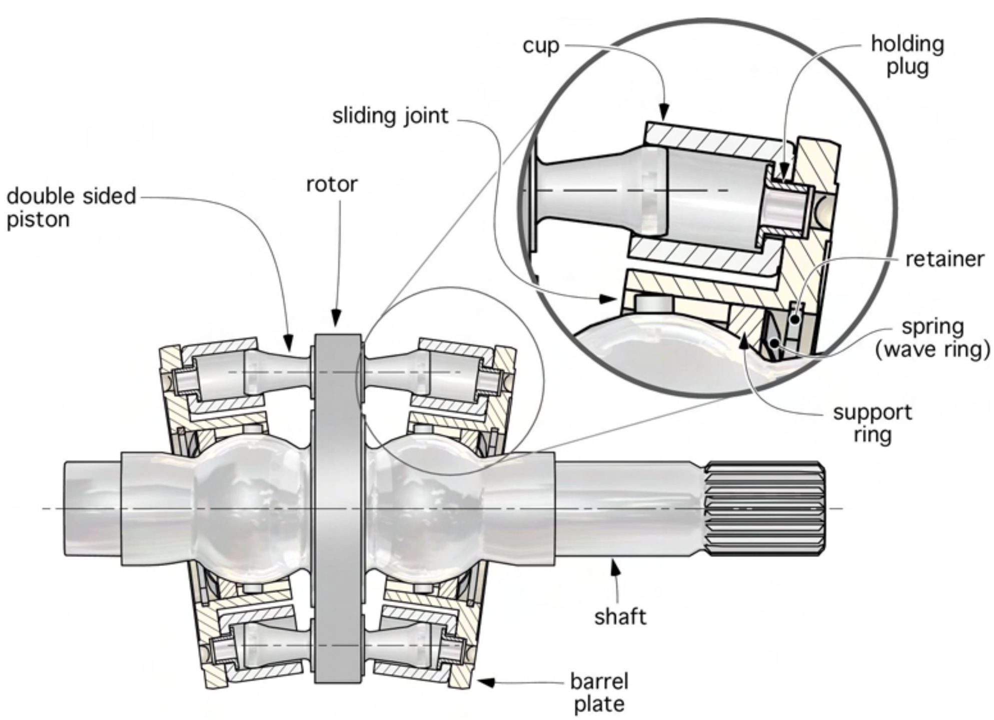


Fig. 2: Cross section of the rotating parts of the floating cup machine

THE FLOATING CUP PRINCIPLE

Most axial piston pumps and motors have either 7 or 9 pistons. Yet, a higher number of pistons could strongly reduce the pulsations and the fluid borne noise created by the hydrostatic machine [13-15]. Also the torque variations at the shaft could roughly be halved if the number of pistons could be doubled. But a doubling of the number of pistons in current bent axis and slipper type pumps is hardly possible. If the tilt angle of the barrel or the swash plate would remain the same, an increase of the number of pistons would reduce the diameter of the pistons to a point were they no longer could take the load. To overcome this problem the tilt angle of the barrel or the swash plate could be reduced, which would improve the bore to stroke ratio. But then, the smaller tilt angle and the increased number of pistons would both result in a strong reduction of the power density [12].

In the new floating cup principle (see Fig. 1 and 2) this dilemma has been solved in five ways:

1. Instead of a single ring of pistons, the floating cup principle features two rings of pistons. The pistons are arranged in a mirrored, back-to-back assembly. This way the pressure forces acting on the pistons are hydrostatically balanced, and the bearing load is strongly reduced.
2. The tilt angle of the barrels is reduced to about 10°. This allows for shorter and thicker pistons. Having a small tilt angle and a high number of pistons, there is enough space in the centre of the rotary group for a shaft, creating the possibility of a through drive.
3. The pistons are locked onto the rotor, without any linkage between the piston and the rotor. This creates the possibility to have a direct conversion of the hydraulic forces acting on the piston to shaft torque, and vice versa. Furthermore the rotor will also take the centrifugal forces from the pistons.
4. The cylinders have been separated from the barrel: each piston has its own cuplike cylinder, which is floating on and supported by a barrel plate. The use of piston rings can be avoided with this design, thereby reducing friction losses and extra costs for the production and mounting of the piston rings.
5. The rotational position of the barrels is synchronized with the shaft-rotor-combination by means of a simple sliding joint. There is no hydraulic power transmitted via the barrels. The joint only has to take the friction between the barrels and the port plates and the mass inertia of the barrels.

Despite the reduced barrel angle, the floating cup machine has the same power density as modern slipper type and bent axis pumps and motors [12]. The highest power density is achieved with 8 to 12 pistons on each site of the rotor i.e. with 16 to 24 pistons per machine. Although the number of parts is increased this doesn't necessarily imply that the production costs are increased as well. Many automotive parts, like roller clutches or hydraulic bucket tappets, are subassemblies, each of them having a large number of parts. Yet these components can be produced at very low cost, most and for all because the design has been optimized for applying low cost production technologies, like deep drawing, extrusion, sintering and fine blanking. The floating cup principle has been designed as such that these technologies can also be applied for the production of hydrostatic machines.

TESTED PUMPS AND TEST CONDITIONS

The first prototype of the floating cup pump (shown in figures 1 and 2) has 12 double-sided pistons. In this machine the port plate on the left side has its timing 15° earlier than the port plate on the right side [2, 11]. By means of this phase shift a true 24-piston operation is achieved. The two barrels have a tilt angle of 8° each. The pump is designed for a maximum continuous operating pressure of 40 MPa. The pump is self-priming up to a rotational speed of at least 3000 rpm.

The pump is tested in detail at the Institute of Fluid Power Drives and Controls (IFAS) at Aachen University. In parallel two standard axial piston pumps have been tested as well:

- In-line slipper type pump (Bosch Rexroth A4FO28, [16]), having 9 pistons and a tilt angle of the swash plate of 20°
- Bent axis type pump (Bosch Rexroth A2FO28 [17]), having 7 pistons and a tilt angle of the barrel of 40°

The floating cup pump and the two reference pumps have a fixed displacement of around 28 cc/rev. The displacement of each machine has been measured at IFAS. Table 1 shows the test results.

Table 1: Geometric displacement (in cc/rev) as measured by IFAS

Pump type	geometric displacement
Floating cup	28.14 cc
Bent axis	28.21 cc
Slipper type	28.20 cc

As with the floating cup pump, the two reference pumps can also be operated up to a nominal pressure level of 40 MPa. The maximum rotational speed at self-priming conditions is for the slipper type machine around 3000 rpm and for the bent axis pump 2500 rpm. All tests have been carried out with HLP46 oil at a temperature of 40°C (ν = 46 cSt).

EFFICIENCY

The efficiency has been measured in 42 stationary points, in a speed range between 500 and 3000 rpm and a pressure range from 5 to 35 MPa, with 500 rpm and 5 MPa increments. To avoid the risk of cavitation at high rotational speeds, all tests are performed at the supply pressure of around 0.3 MPa. Appendix A describes the test bed. Figures 9-11 summarize the outcome of the tests.

The diagram below (Fig. 3) shows the overall efficiency, this time only for a pressure difference of 30 MPa. As expected the bent axis pump has a higher efficiency than the in-line, slipper type pump. Over the entire test field, the difference in total efficiency between the two pumps is about 2 to 3%. In the speed range above 2000 rpm the floating cup has about the same high efficiency as of the bent axis machine. For lower speeds the efficiency of the floating cup pump is however significantly higher.

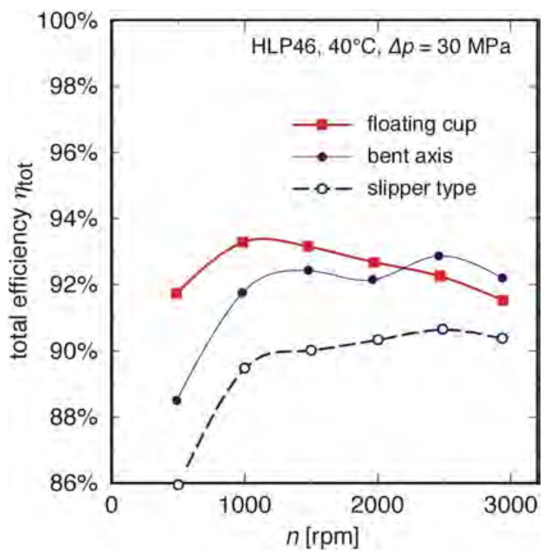


Fig. 3: Overall efficiency (ISO 4409 definition) for the three pumps at an operating pressure of 30 MPa.

Having measured the geometric displacement V (see table 1) it is also possible to define theoretical maximum values for the flow output and torque of the pumps, and to calculate the volumetric (Eq. (1)) and hydraulic-mechanical efficiency (Eq. (2)):

$$\eta_{vol} = \frac{Q'}{Q_{th}} = \frac{Q'}{\frac{n}{60} \cdot V} \tag{1}$$

$$\eta_{hm} = \frac{M_{th}}{M} = \frac{\Delta p \cdot V}{2\pi \cdot M} \tag{2}$$

In Eq. (1) Q' is the flow in the high-pressure line, which is calculated from the measured flow Q after the load valve (see Appendix A). Figures 4 and 5 show the volumetric and hydraulic-mechanical efficiency, as calculated from the measurements at an operating pressure of 30 MPa.

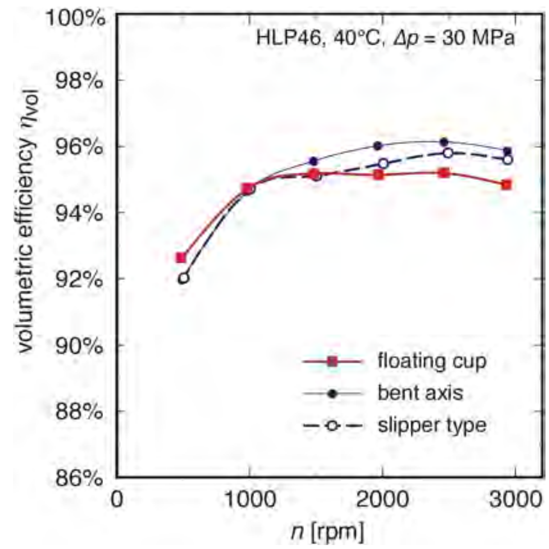


Fig. 4: Volumetric efficiency as a function of the rotational speed at an operating pressure of 30 MPa.

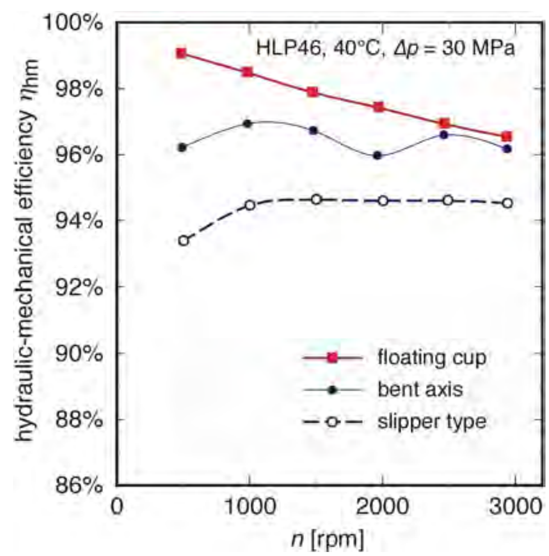


Fig. 5: Hydraulic-mechanical efficiency as a function of the rotational speed at an operating pressure of 30 MPa.

As can be seen in Fig. 4 the volumetric efficiency of the three pumps is almost the same. The higher overall efficiency of the floating cup pump appears to be completely due to the strong reduction of the friction losses (see Fig. 5). The difference in friction behavior between the floating cup pump and the slipper type pump is also shown in figure 6, which shows the torque losses ΔM for both pump principles:

$$\Delta M = M_{th} - M = \frac{\Delta p \cdot Q'}{2\pi} - M \tag{3}$$

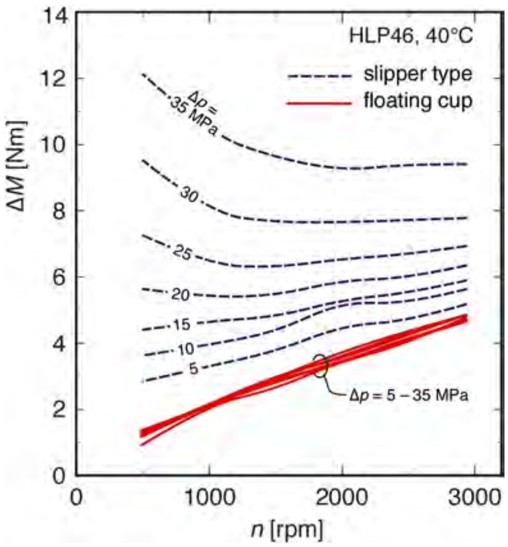


Fig. 6: Torque losses ΔM for the floating cup pump and a comparable slipper type pump.

The torque losses of the in-line slipper type pump are strongly related to the pump pressure, whereas the losses in the floating cup pump are almost completely independent of the pressure. The difference can be explained by looking at the differences in loading conditions for both principles.

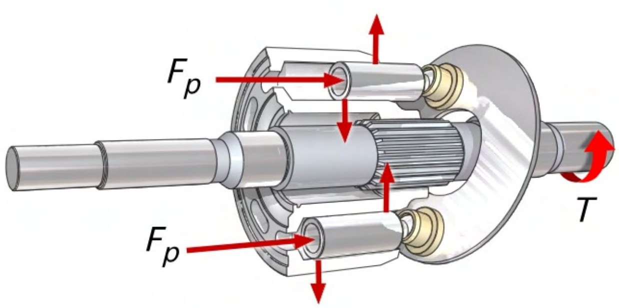


Fig. 7: Piston loads in a slipper type machine.

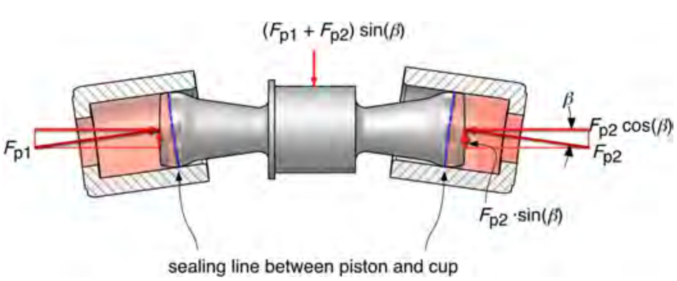


Fig. 8: Piston loads on a single piston pair of a floating cup machine.

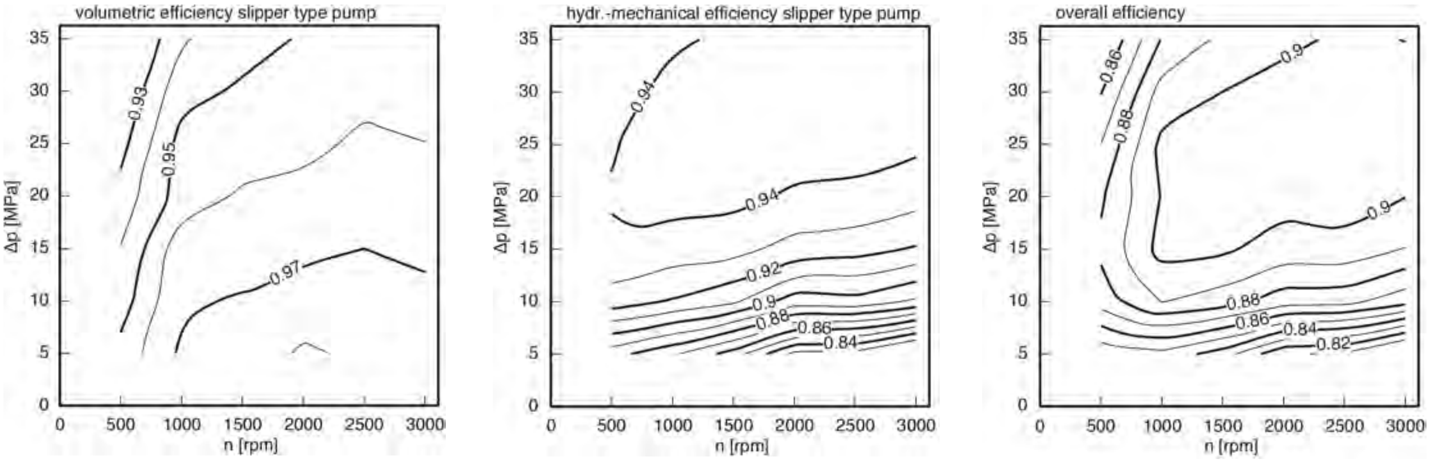
In a slipper type machine (Fig. 7), the pistons are supported on the tilted swash plate by means of slippers. Due to the slipper construction, the pistons will tend to be pushed away from the swash plate. This is prohibited by the contact between the cylindrical piston and the cylinder wall. The complete hydraulic power is transferred through this sliding interface between the piston and the cylinder wall. In addition the centrifugal forces acting on the piston are counteracted in this sliding interface as well. The resulting friction in the interface between piston and cylinder is pressure dependent [18], and contributes to a large extend to the total loss of a slipper type machine.

In the floating cup pump the load situation is completely different:

- The cuplike cylinders are not structurally attached to another part of the machine, but they are free to move on the rotating barrel plate [10]. Therefore the cups cannot transmit any forces. In the radial direction the piston governs location of the cup. In the axial direction the cup is pushed to the barrel plate by means of the internal pressure forces. The pressure forces in the gap between the cup and the barrel plate largely balance these forces.
- The sealing line between the cylinder and the ball shaped piston head is always standing at a 90° angle to the cylinder axis. As a result the hydraulic pressure creates a radial load on the cylinder that is equal in all directions. Thus the cups are completely balanced and unable to create a high load on the piston.
- The pistons are (for the most part) balanced in the axial direction but not in the radial direction (see Fig. 8). The radial component of the pressure forces creates a combined force and torque load at the point were the rotor supports the piston.
- Unlike the construction in slipper type and bent axis machines, the pistons of the floating cup machine are locked to the rotor. This way the radial forces acting on the pistons can be transferred directly to the rotor and the shaft.

In conventional pump and motor principles the complete hydraulic power is transmitted via one or more sliding interfaces. The combination of a high load and a high velocity is the main reason for the relatively high friction losses in these machines. Since it cannot be avoided to have relative movements in a positive displacement machine, the best way to avoid large friction losses is to avoid the high loads and poor lubrication situations. This is what is achieved in the design of the floating cup principle.

The remaining torque losses are probably due to the viscous friction in the gap between the rotating barrel plates and the stationary port plates. This friction is predominantly dependent on the oil characteristics, the gap height and the relative velocity between the sliding surfaces. Because the floating cup machine has two barrels instead of only one, these torque losses are higher than in comparably sized slipper type machines.

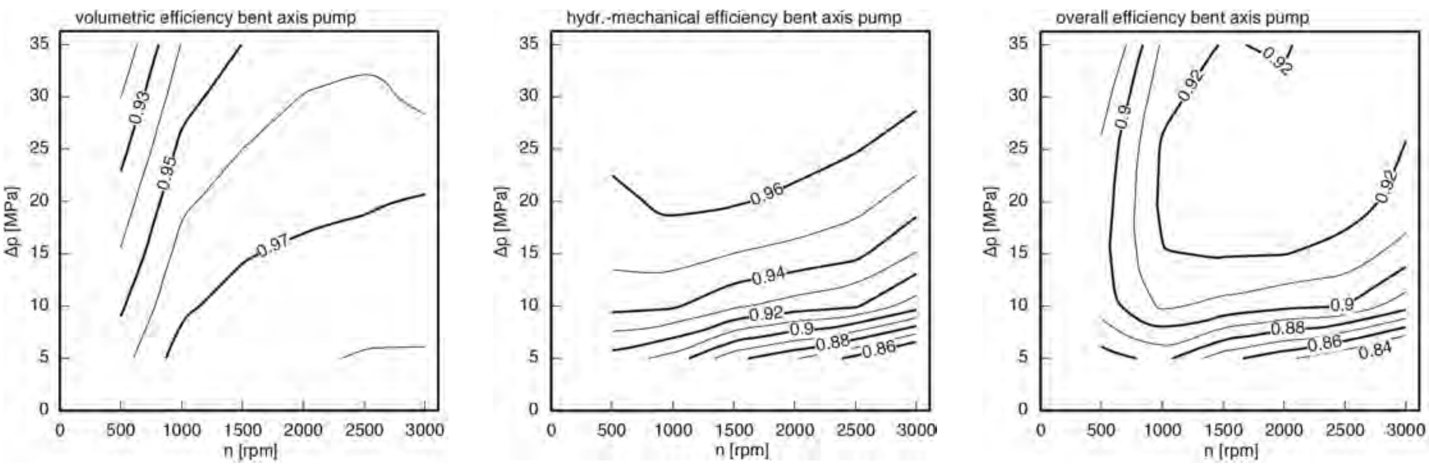


9a: volumetric efficiency

9b: hydraulic-mechanical efficiency

9c: total efficiency (ISO 4409)

Figure 9: Measured efficiencies of a constant displacement slipper type pump (28 cc/rev)

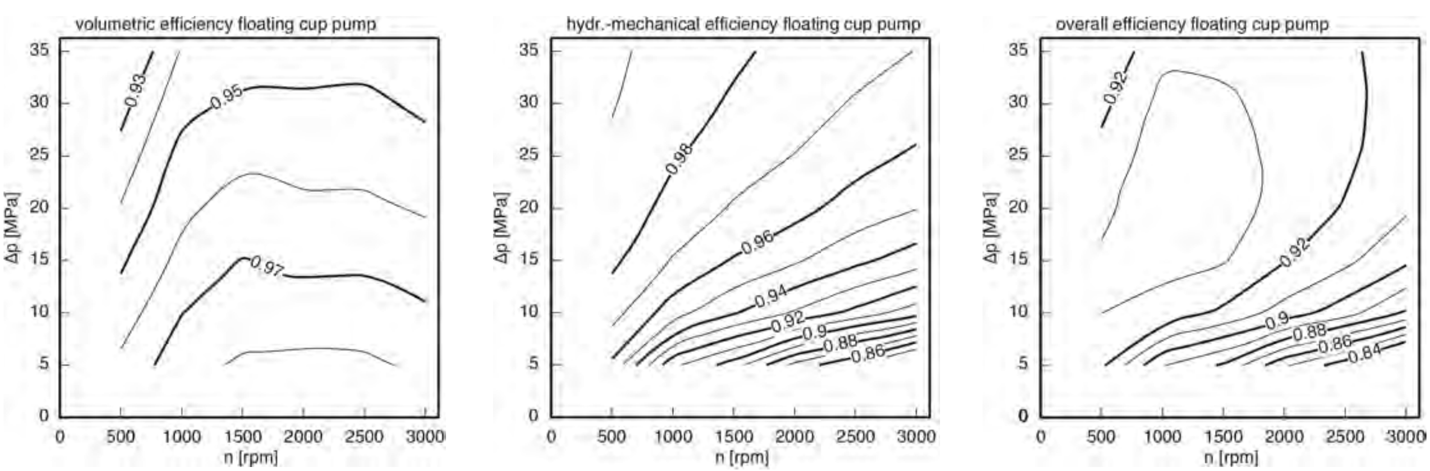


10a: volumetric efficiency

10b: hydraulic-mechanical efficiency

10c: total efficiency (ISO 4409)

Figure 10: Measured efficiencies of a constant displacement bent axis pump (28 cc/rev)



11a: volumetric efficiency

11b: hydraulic-mechanical efficiency

11c: total efficiency (ISO 4409)

Figure 11: Measured efficiencies of a constant displacement floating cup pump (28 cc/rev)

LOW-SPEED BEHAVIOR

The excellent low friction behavior of the floating cup principle has also been confirmed in the low-speed test. The test has been performed at a rotational speed of 0.1 rpm. At this speed, the low-speed test conditions are very close those of a real start, which makes the test also suitable for examining starting behavior [19]. The test has been performed in a pressure range from 5 to 35 MPa, with 5 MPa increments.

Appendix B describes the test bed, which has been used by IFAS to measure the low-speed behavior. Aside from the floating cup pump, the same bent axis pump as used for the steady-state performance has been examined as well. Both pumps are tested as a motor.

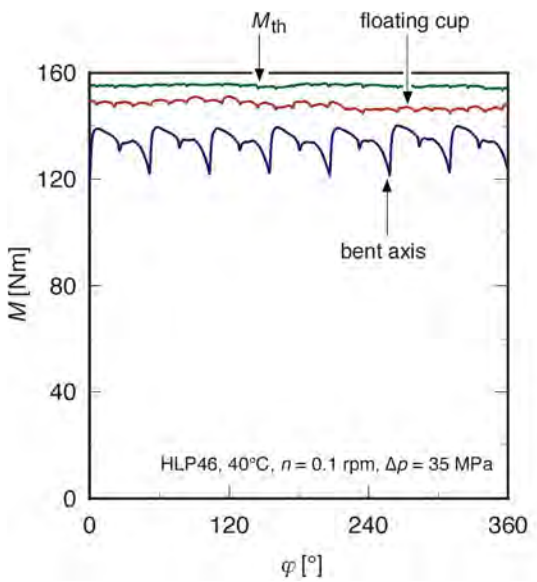


Fig. 12: Low-speed torque M as a function of the angular position φ measured during one revolution.

Figure 12 shows the measured torque during a single revolution for the floating cup machine and the reference bent axis machine. The torque of the bent axis machine varies strongly during one revolution. Clearly visible are the 7 pistons passing the valve lands of the port plate in the top and bottom dead centers. The torque line for the floating cup pump is however much more flat; due to the larger number of displacement volumes per revolution the torque variations are almost disappeared.

As before the maximum theoretical torque M_{th} can be calculated. This torque is also shown in the diagram of Fig. 12. Since the geometric displacement of both units is almost equal, the theoretical torque is valid for both pumps. Also now the torque losses of the floating cup machine are very low and certainly much less than of the bent axis machine.

For both machines a minimum torque (as defined in [14, 20]) can be found and compared to the theoretical maximum. The result is plotted in the figure below. On average the torque delivered by the floating cup pump is 19% higher than of the bent axis pump at equal conditions.

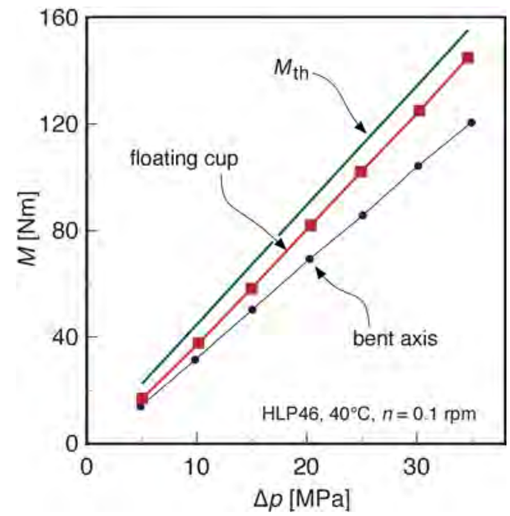


Fig. 13: Minimum torque of the floating cup pump and a reference bent axis pump, both compared to a theoretical maximum torque.

In the bent axis machine the torque losses increase linearly with the pressure, whereas in the floating cup machine the torque loss is almost constant. As a result the low-speed hydraulic mechanical efficiency of the floating cup pump is much higher than of the bent axis machine (see Fig. 14). For the calculation of this efficiency not the minimum torque but the average torque value is taken as a reference [20, 21].

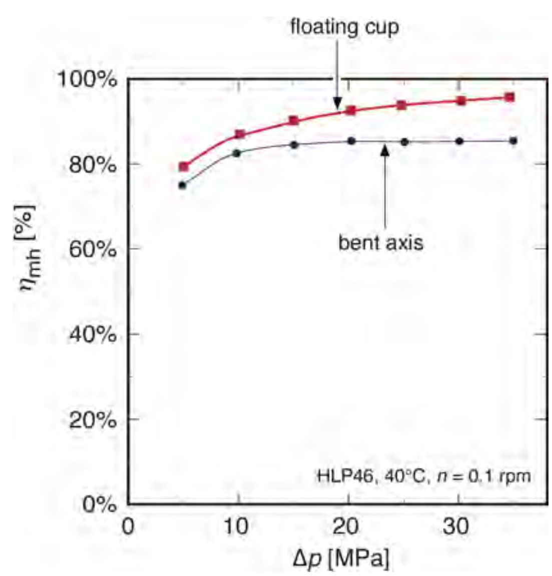


Fig. 14: Hydraulic mechanical efficiency at low-speed conditions.

Although the low-speed behavior of the slipper type machine has not been tested it is generally accepted that these machines have a poor start-up and low-speed behavior [14, 22]. Also compared to other types of hydrostatic motors the tested floating cup machine has the highest low-speed efficiency [19].

Table 2: Comparison of the low-speed hydraulic-mechanical efficiency for various hydrostatic positive displacement principles (n=0.1-0.16 rpm, T =40°C, HLP46-oil, p =21 MPa)

Hydrostatic principle	η at low speed
Floating cup	92.2%
Bent axis	84.8%
Radial piston I [19]	85.7%
Radial piston II [19]	79.1%
In-line slipper type [19]	78.3%

Aside from the torque losses at low speed the volumetric losses have been measured as well. The flow that needs to be supplied during stand still is an important characteristic for hydraulic motors since it determines the ability of a motor to hold a hanging load. Figure 15 shows the flow that is measured during the low-speed test at an operating pressure of 35 MPa. At an operating speed of 0.1 rpm the displacement flow of the machine is less than 0.003 L/min and can therefore be neglected. Consequently the flow measured can be completely attributed to the internal leakage of the tested machines.

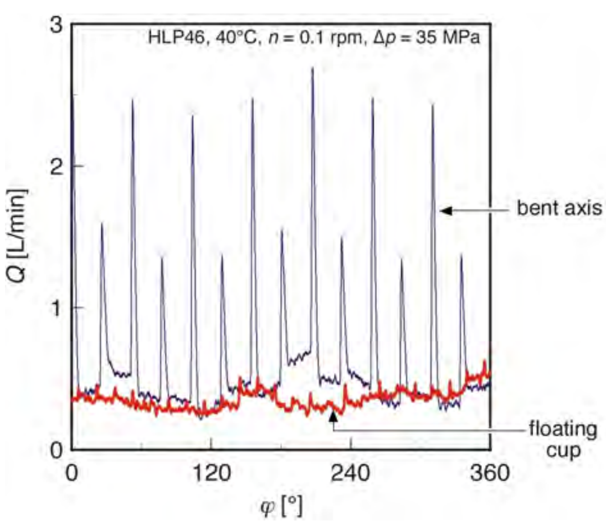


Fig. 15: Leakage flow as a function of the angular shaft position φ.

It should be reminded that the low-speed test has not been performed with motor configurations (having motor port plates), but with two pumps. The negative port overlap of the bent axis pump is clearly visible in the graph plotted in Fig. 15. But even a motor port plate still has to some extend a negative overlap between the barrel ports and the ports of the valve plate.

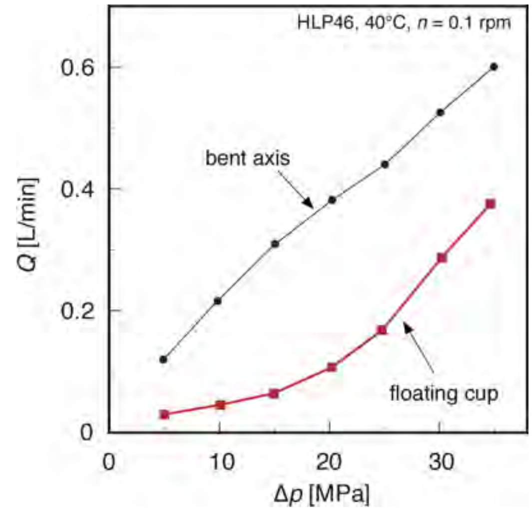


Fig. 16: Average leakage flow as a function of the operating pressure at a rotational speed of 0.1 rpm.

The tested floating cup pump on the other hand has a zero overlap, which strongly reduces the flow losses due to short-circuiting. But most important, the flow measurements prove that, despite the increased number of leakage gaps, the floating cup pump does not have high leakage. From separate leakage measurements it can be concluded that the leakage shown in Fig. 16 is almost completely coming from the gaps between the pistons and the cups.

CONCLUSIONS AND OUTLOOK

In the new floating cup principle the hydraulic pressure forces almost directly create a torque on the shaft. Sliding contacts with a high load are in principle avoided. This strongly reduces the friction losses in hydrostatic machines. The steady state and low speed experimental results, which are reported in this paper, have proven the low friction characteristics of the floating cup principle. The experiments have also shown that the volumetric losses are not increased because of the high number of leakage gaps, nor by the direct sealing of the piston crown on the cylinder wall.

Compared to the bent axis pump the floating cup principle has on average about the same efficiency, although at 500 rpm the overall efficiency is up to 4% higher. Compared to the slipper type pump the floating cup pump offers an efficiency improvement of around 3% on average and up to 7% at 500 rpm.

The floating cup principle has an excellent low-speed and start-up behavior. The hydraulic-mechanical efficiency of the floating cup machine at 35 MPa and 0.1 rpm is almost 10% higher than of the bent axis machine. Due to the higher number of pistons the torque variations of the floating cup machine are also much smaller than of the bent axis machine. Altogether the minimum torque is increased by almost 20%. Aside from a reduction of the torque losses, the floating cup pump also reduces the volumetric losses at low speed conditions, which improves the load holding behavior during stand still and inching.

It is also obvious that, unlike the bent axis principle, the new floating cup principle offers the possibility of a through drive. Furthermore, due to the high number of pistons, the floating cup principle strongly reduces the pulsations, thereby lowering the fluid borne noise in hydraulic systems. Experiments have already verified the pulsation reduction [9]. From a cost point of view the floating cup principle opens the door to the world of high volume, low cost production techniques which are already applied for many years in the production of automotive components.

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URL: http://www.rwth-aachen.de/ifas/

NOMENCLATURE

n	Rotational speed	[rpm]
M	Torque	[Nm]
M_{th}	Theoretical torque	[Nm]
ΔM	Torque loss	[Nm]
p	Pressure level after the load valve	[Mpa]
p'	Pressure level before the load valve	[Mpa]
Δp	Pressure difference across the pump	[Mpa]
Q	Flow after the load valve	[m^3/sec]
Q'	Flow before the load valve	[m^3/sec]
Q_{th}	Theoretical pump flow	[m^3/sec]
T	Oil temperature after the load valve	[°C]
T'	Oil temperature before the load valve	[°C]
V	Geometric pump displacemnet	[m^3]
α	Coefficient of cubic thermal expansion	[1/K]
β	Tilt angle of the barrel	[°]
η_{hm}	Hydraulic-mechanical efficiency	[-]
η_{vol}	Volumetric efficiency	[-]
η_{tot}	Overall efficiency	[-]
$\overline{K}_\tau$	Isothermal bulk modulus	[Pa]
υ	Kinematic oil viscosity	[cSt]

APPENDIX A: TEST CIRCUIT STEADY-STATE EFFICIENCY MEASUREMENTS

In Fig. 17 the hydraulic circuit of the test installation for the steady-state measurements at IFAS is shown. In order to avoid cavitation (especially for the bent axis pump) the pump inlet is pressurized to approximately 0.3 MPa. All tests are performed with HLP46 oil at a temperature of 40 ± 2°C.

In this paper the overall efficiency is defined as in ISO 4409-1986 [23]. According to this definition the compression energy of the output flow is regarded as a (flow) loss. This implies that the flow must be measured either directly after the pump (and before the load valve), or it is measured after the load valve and must be corrected for compressibility effects. At IFAS the latter method is applied. To calculate the flow at a compressed condition the pressure and temperature of the output flow is measured both before and after the load valve, and used to calculate the compressed oil flow:

$$Q' = Q \cdot \left(1 - \left[\frac{p' - p}{\overline{K}_\tau} \right] + \alpha \cdot \left[T' - T \right] \right) \qquad (4)$$

In this equation:

$\overline{K}_\tau$ = isothermal bulk modulus = 14850 [Pa]

α = coefficient of cubic thermal expansion = 0.0007 [1/K]

Because of this definition the overall efficiency and the volumetric efficiency are at high pressures up to 2.8% lower than in case of a pure energetic efficiency.

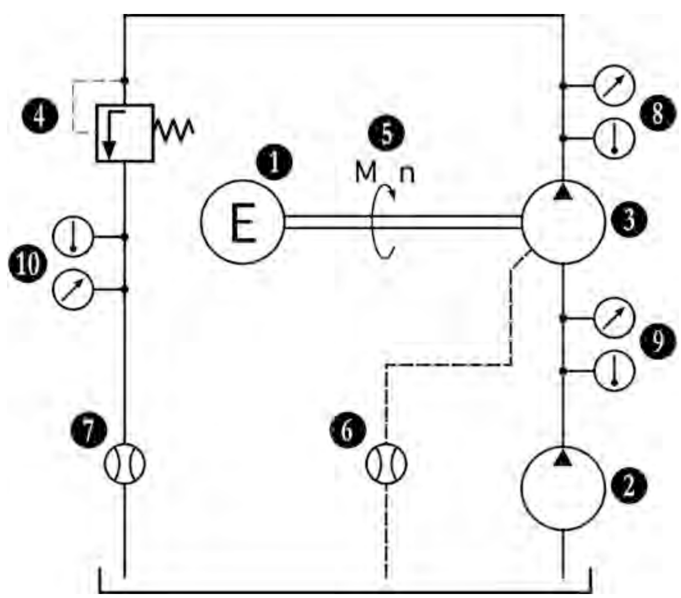

Fig. 17: Hydraulic circuit for steady state measurements at IFAS.
1. electric motor
2. low pressure supply pump
3. pump to be tested
4. load valve
5. rotational speed and torque sensor
6. flow sensor leakage line
7. flow sensor pump flow
8. pressure and temperature sensor in the high pressure line
9 pressure and temperature sensor in the low pressure line
10. pressure and temperature sensor after the load valve

Steady-state measurements are performed in a pressure range form 5 to 35 Mpa and a speed range form 500 to 3000 rpm, with 5 Mpa and 500 rpm increments. This results in total in 42 test points. After steady-state test conditions are achieved for each test point, 2000 readings are performed and averaged.

APPENDIX B: TEST CIRCUIT LOW SPEED MOTOR CHARACTERISTICS

For the low speed tests, IFAS has a separate test bed. The hydraulic circuit is shown in Fig. 18.

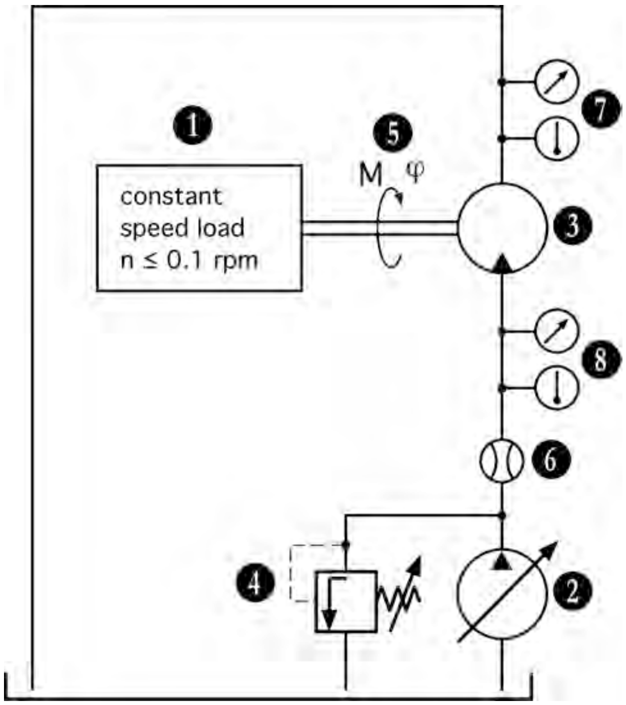

Fig. 18: Hydraulic circuit for low speed tests at IFAS.

1. constant speed load
2. high pressure supply pump
3. motor to be tested
4. pressure-relief valve
5. angular position and torque sensor
6. flow sensor high pressure suppy line
7. pressure and temperature sensor in the low pressure line
8. pressure and temperature sensor in the high pressure line

For the low-speed measurements the pumps are tested as motors in a pressure range from 5 to 35 Mpa with 5 MPa increments, which results in a total of 7 test points. The tests are performed at a constant speed of 0.1 rpm. The backpressure is around 7 bar. In each test point the motor is running for two full rotations. The data are recorded for each 0.08 seconds at a sample frequency of 12.5 Hz. For the tests HLP46-oil is used having a kinematic viscosity 46 cSt at 40°C.