Efficiency of a Variable Displacement Open Circuit Floating Cup Pump

Georges E.M. Vael, Peter A.J. Achten and Titus van den Brink
Innas BV
Nikkelstraat 15, 4823 AE Breda, Netherlands
Phone +31 76 5424080, Fax +31 76 5424090
E-mail: gvael@innas.com, pachten@innas.com, tvdbrink@innas.com

ABSTRACT

The Floating Cup Displacement principle is a relatively new axial piston displacement principle for hydrostatic pumps, motors and transformers. Since its origin in 2001, it has been mainly applied in fixed displacement pump prototypes.
At the SICFP'05, a design for a variable displacement open circuit Floating Cup Pump was presented. Now, the first prototype of a 28 cm^3 unit based on this design, has been built.
This papers reports the first efficiency measurements of this prototype. So far, the unit has been tested at 100% and 50% swash angle. The total efficiencies at these swash angles are already quite good, even if no design loops have been made yet.
The paper further describes a new method to separate the losses in this unit over the different loss mechanisms. This method is used to distill the mechanical torque losses from the measurements.
An inverse version of this method is then used to predict the total efficiencies of this unit at 25% and 75% swash angles.

KEYWORDS: Variable axial piston pump, Floating Cup, Efficiency. Loss separation.

1 INTRODUCTION

The Floating Cup (FC) displacement principle is a relatively new axial piston displacement principle. The idea for it originated in 2001, in the context of the on-going development of a new axial-piston, hydraulic transformer: the Innas Hydraulic Transformer (IHT). By that time, the functionality and potential of the IHT concept had been proven [1,2 and 3] but in order to advance it to an application stage, it was necessary to further improve its stability, its efficiency, and its noise- and pulsation behavior. A search for ways to achieve this, had resulted in a number of patented solutions [4]. During that search, however, it had also become clear that the IHT could be drastically improved on all these aspects, by significantly increasing the number of pistons of its rotary group. Doing this with a conventional axial piston displacement principle, is not an option, as that would increase the production costs considerably. With the FC displacement principle, a way has been found to increase the number of pistons from the typically 7 or 9 pistons of a conventional axial piston unit, to typically 24 for the FC design, without raising the costs of the rotary group [5 and 6].

It was soon realized that this new displacement principle could also be successfully applied in hydrostatic pumps and motors. FC pumps and motors represent a one-to-one replacement option to the market, an IHT also requires a change from the conventional hydraulic system lay-out to a Common Pressure Rail (CPR) type of system layout. Although the CPR system type offers considerable advantages (load controllability, energy recuperation and power peak shaving [2]), such a hydrostatic system change was - at that time - still quite a marketing hurdle to take. Therefore, the FC development has been initially and mainly aimed at hydrostatic pump an motors.

Over the years, several prototypes of fixed displacement FC units have been built and tested. Early tests have already shown the high efficiency, the low pulsation levels and the low noise level of the new principle [7, 8 and 9]. Later tests [10] illustrate that the development process of the FC principle has increased its efficiency even further. The efficiency of fixed displacement FC units is higher than that of modern bent axis machines. This is primarily due to the total absence of pressure dependent side forces between the pistons and the walls of the cups in which they move. This results in a strong reduction of the friction losses in the floating cup machine. Perhaps more important, in combination with the large number of pistons, the absence of these side force also results in an extremely high hydromechanical efficiency at startup and low speed conditions [7 and 9]. Despite the relatively large number of leakage gaps, the volumetric efficiency of floating cup units is about equal to conventional slipper type and bent axis machines [11].

The floating cup principle is designed such, that many of its components can be produced by means of deep drawing, extrusion, sintering and other modern, non-machining production technologies. Cost studies, performed together with the industry, have proven the strong cost reduction potential of the floating cup principle, if the units can be made in sufficient numers. The production cost break-even point of fixed displacement FC units, compared to conventional units, lies roughly at 3000 units per year.

Up until 2005, all FC pump and motor studies had been limited to constant displacement units. At the SICFP'05, a first design of a variable displacement floating cup pump for open circuits was presented [12]. The design - and especially the design of the swashing system [13] - was refined and in 2007 parts for a 28 cm^3 prototype were procured.
Because priority has been given to studies into 'hydrid' systems and to the development of a state-of-the-art Floating Cup hydraulic transformer for these systems [14 and 15 and 16], the actual testing of the variable displacement open circuit FC pump (FCVO) was delayed until the beginning of 2009.

This paper reports the results of the first efficiency measurements of the 28 cm^3 FCVO. Total efficiencies are presented and the division of the underlying losses over the several loss mechanisms. In order to make this division, a new approach to loss separation is adopted.

Efficiencies were measured at 100% swash angle and at 50% swash angle. Unfortunately, the planned measurements at 75% and 25% swash angles could not be performed in time for this paper. After the first two measurement series a design flaw became apparant: the swash plate bearings had be designed as a cast iron to cast iron contact and that material combination proved susceptible to 'Brinelling' when operating for a long time at a fixed swash angle. Another material combination was chosen but could not be implemented in time.

Therefore, the results of the 50% and 100% swash angle measurements have been used to predict the efficiencies at 25% and 75% swash angles. The prediction is based on the same new approach of loss separation, and gives very plausible results.

2 THE VARIABLE DISPLACEMENT OPEN CIRCUIT FLOATING CUP PUMP

Figure 1 shows the construction of the 28 cm^3 FCVO. The rotary group of this unit is identical to that of a constant displacement FC pump. The displacement volumes in this rotary group are 24 separate cylinders, the cups, which float under a light axial balance force on two barrel plates. In each cup, a piston with a spherical piston head moves up and down. No piston rings are used. As the sealing line between the piston head and the wall of the cup, is always perpendicular to its central axis, the internal pressures acting in radial direction on the cylinder walls are totally balanced. Because of this, and because of the absence of piston rings, there are no pressure dependent contact forces between cup and piston. In conventional axial piston units, pressure induced side forces between the pistons and the side walls are an important source of hydromechanical losses. This also explains the rather poor starting and low speed capabilities of conventional units.

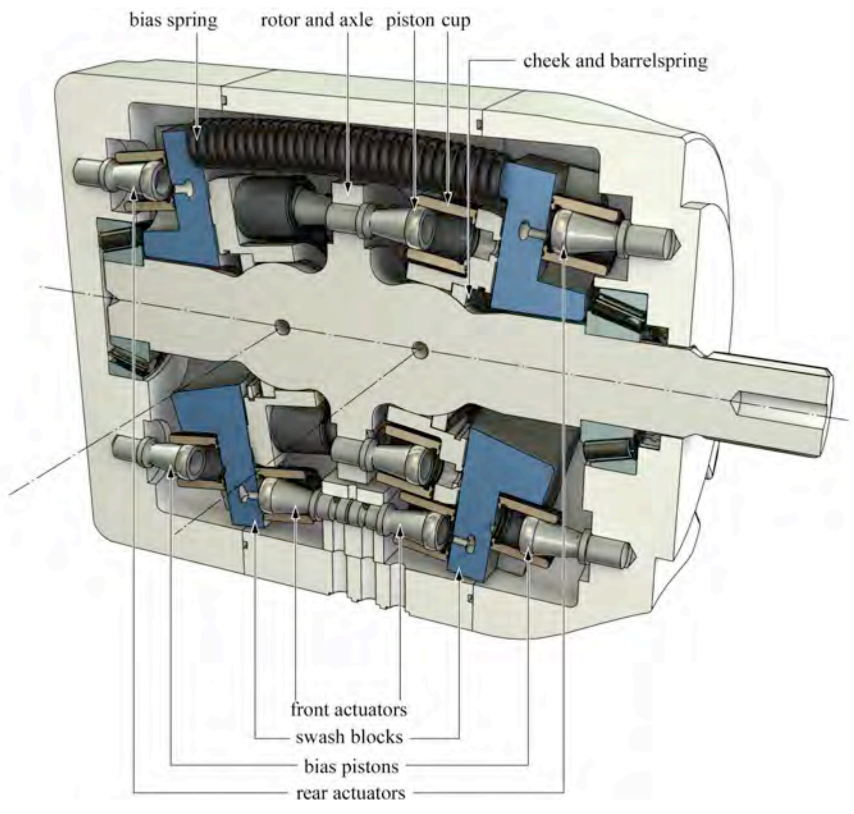

Figure 1: Construction of the 28 cm^3 variable displacement open circuit FC pump (FCVO).

The pistons in the FC design are rigidly mounted in two groups of twelve, one on each side of a rotor. The rotor is rigidly mounted on the axle. The oil flow in and out of the unit is split up over two port plates, one on each side of the unit. Always, the same number of pistons on each side of the rotor is connected to the high pressure. This results in a rotor that is totally balanced in axial direction. Consequently, the loads on the axle bearings are light and relatively small bearings can be chosen.
The barrel rotation is synchronized to the axle rotation by means of synchronization pins, localized in the spherical joints between axle and barrels. A sprung ring, (the 'cheek') ensures that there is always a minimum force pressing the barrel on the port plate. Under pressure, the resulting force between barrels and port plates is the sum of this spring force and the carefully selected hydraulic barrel balance. Setting the barrel balance is subject to optimization: setting it too light will result in low loss torques between barrels and port plates but may lead to high leakage in this interface, setting it too high would result in low leakage but at the cost of high loss torques in this interface. In the FC design, this is the interface which is responsible for the largest part of the loss torques, and thefore it is very important that the barrel balance is carefully set.

In the FCVO, the port plates do not lie directly on faces in the units case but on top of two swash blocks. The swash blocks determine the displacement of the machine by tilting the barrel plates with respect to the rotor. Each swash block can be rotated around its respective swash axis by a set of two actuators that work against the forces of a bias spring and a bias piston that is always connected to the units high pressure. The actuators and compensators are piston and cup combinations of the same size as those used in the rotary group. In this way the production costs for the swashing system are minimized. Extensive details on the construction of the swashing system can be found in [14 and 15].

3 TESTBED AND TEST METHOD

The efficiency tests of this unit were performed on a testbed in the hydraulic laboratory of the Control Systems Technology Group, Department of Mechanical Engineering, Eindhoven University of Technology (TU/e). The measurements were part of a larger research project into the losses in the barrel-port plate interface of FC units. The research project is sponsored by the The Netherlands Organisation for Scientific Research (NWO), under the 'Casimir' program.

In the testbed, the 28 cm^3 FCVO is driven by a constant displacement hydrostatic motor. The speed of the combination is determined by setting the flow which is sent to the motor by a Hydraulic Power Unit. The power unit has a large tank, with an extensive oil-temperature control system. The pump pressure is set using an array of variable orifices, downstream of the unit.

The measured signals and their designations as used in the rest of this paper, are given below.
For the derivation of equations in this paper, SI units were used for these measured signals.
For the presentation of measurement and calculation results in figures, the more intuitive units of bar and liter per minute are used.

pressure difference over the pump	Δp
pump inlet pressure	p_1
mass flow at the pump outlet	$q_{m, 2}$
leakage volume flow at the case drain	$q_{v, leak}$
temperature at the pump inlet	t_1
temperature at the pump outlet	t_1
temperature at the leakage flow measurement point	t_{leak}
Torque	T
speed	n

Typical for the measurement at the TU/e testbed is the measurement of the mass flow in stead of the more common volume flow measurement at the pump outlet.
To this end, the TU/e testbed contains a dedicated weighing vessel. As soon as the pump under measurement has reached a stationary operating point, the output flow is directed into the vessel and the weight of the vessel is recorded over a certain time.

Figure 2 shows the pump on the testbed at the TU/e. The weighing vessel is visible in the background.

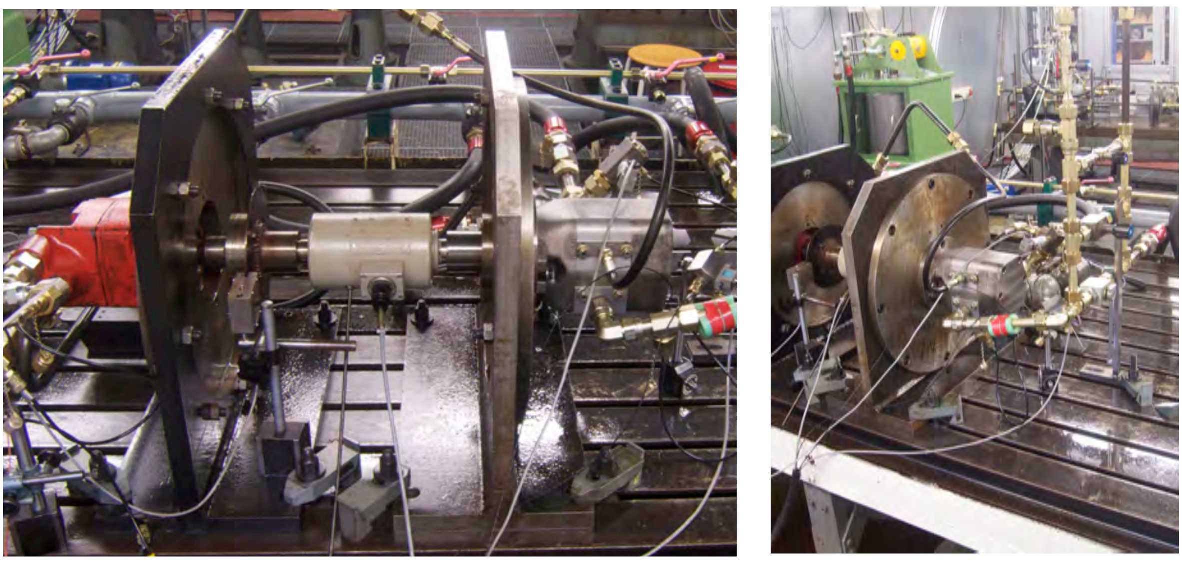

Figure 2: The measurement set-up at the TU/e.

With an appropriate equation of state - oil density ρ as a function of pressure p and temperature t - the measured downstream mass flow can be converted to the oil volume flow at any point in the pump outlet line for which the local oil pressure and temperature have been measured. At the TU/e, the equation of state as derived by Witt [17] is used:

$$\rho = \left[E_0\left(1-\left[\exp\left(-\frac{(p-1)}{E_1}\right)+E_2\,(p-1)\right]\right)+D\right]t+\frac{\rho_{0(0)}}{1-C\lg\frac{B+(p-1)}{B}} \tag{1}$$

In this so-called 'model function', the parameters $\rho_{0(0)}$, B, C, D, E_0, E_1 and E_2 have to be fitted for the fluid which is to be modeled. Fitting the full set of 7 parameters, would require an extensive and difficult set of measurements of the density at different temperatures and pressures. Luckily, in one of his many publications on this subject [18], Witt introduces the concept of a calculation fluid and proves that for mineral oils only two parameters have to be determined: $\rho_{0(0)}$ (the oil density at atmospheric circumstances and 0 ^{o}C) and D (the linear density-temperature relationship at atmospheric pressure. The rest of the parameters in the model function can be taken equal to those of a calculation fluid, which represents the 'average' mineral oil. Witt has determined these parameters but seems to have never published them. At the TU/e, where Witt did his research, they are available. The result of this simplified fitting process, is a model that is more than accurate enough for the normal pressure and temperature ranges in hydrostatics.

The parameters $\rho_{0(0)}$ and D of the Mobil DTE25 oil used in the testbed, were determined by a linear regression of a pycnometer measurement of the density of this oil at several temperatures and under atmospheric pressure.

Using the resulting derived equation of state for this oil, the mass flow $q_{m,\,2}$ can be converted to the volume flow directly at the pump outlet. Using the same equation of state and the measured t_{leak}, the measured leakage volume flow $q_{v,\,leak}$ can be converted to a leakage mass flow, $q_{m,\,leak}$. The the mass flow at the pump inlet $q_{m,\,1}$ can then be calculated as the sum of q_{m2} and $q_{m,\,leak}$. And using the model function again, $q_{m,\,1}$ can be converted to the inlet volume flow $q_{v,\,1}$. With that, all quantities required for the determination of the pumps efficiency according to ISO 4409 are available.

The pressure sensors, the torque measuring shaft and the weighting vessel, are all regularly calibrated against gravity, in the same laboratory at the TU/e. They have systematic errors well below the 0.5% required for an ISO class A measurement rating.

4 MEASURED EFFICIENCIES

As explained in the introduction, only two swash angles could be measured so far: both sides 100% swashed (8 degrees swash angles) and both sides 50% swashed (4 degrees swash angles). During the measurement, the swash blocks were secured in the measurement position by means of adjustable end-stops which were added to the prototype. The measured total efficiencies for both measured swash angles are shown in figure 3. The total efficiency has been calculated according to the ISO 4409 definition:

$$\eta_t = \frac{q_{v,2}\cdot p_2 - q_{v,1}\cdot p_1}{\omega\cdot T} \tag{2}$$

The figure shows that the overall efficiencies of this unit, at both swash angles, are already quite high, considering that these are the first measurements of the first FCVO prototype ever. Like the efficiency of the fixed displacement unit has increased during its development, the efficiency of the FCVO may also be expected to increase further when the inevitable ‘childhood’s diseases’ have been identified and cured.

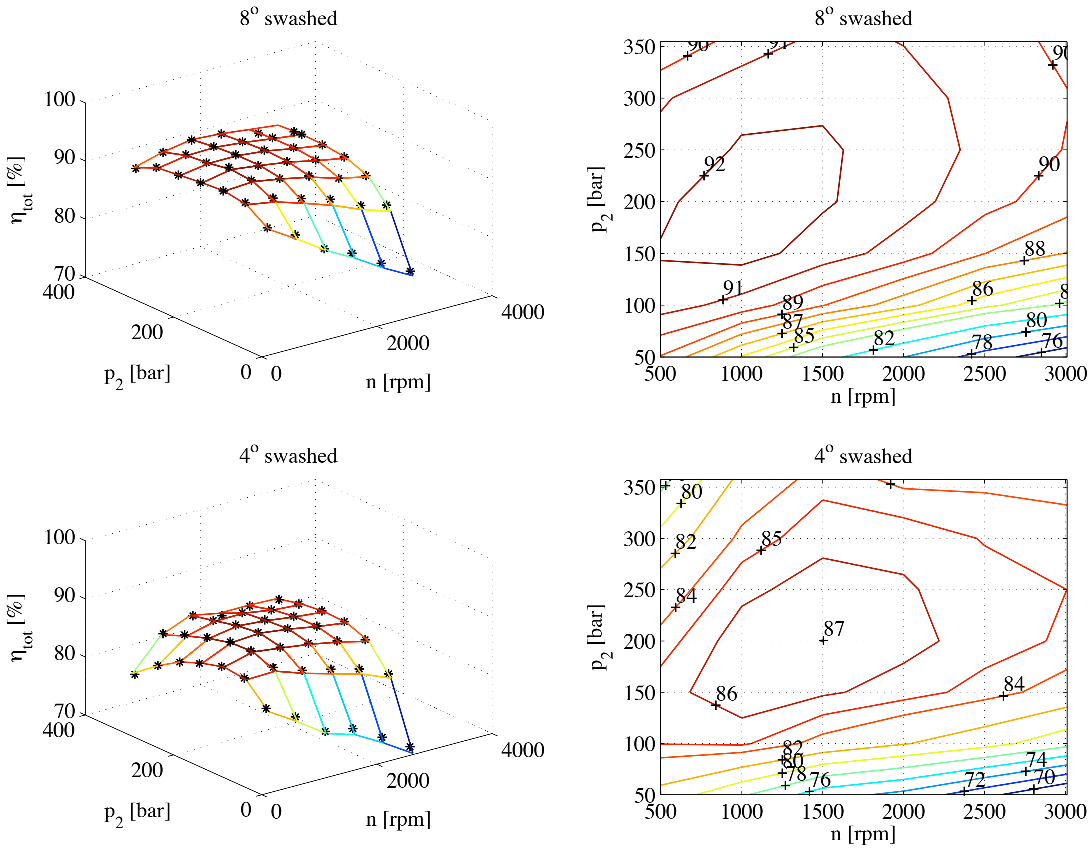

Figure 3: Measured efficiencies (∗) and fitted surfaces, 8° and 4° swash angles.

It is common practice to split up the measured total efficiency in a volumetric efficiency and a hydromechanical efficiency. From a pump-designers viewpoint, this split-up is hardly useful, as it does not reveal to what extend the different loss-mechanisms contribute to the overall efficiency loss. The pump designer wants to know the amount of leakage from the pump, in which interfaces that leakage is generated, how much torque is lost in the unit and where these torque losses occur.

In a hydrostatic pump with an external leakage line, the leakage flow from the unit, can be directly measured. In order to determine in which interfaces the measured leakage is generated, it is necessary to do extra experiments. In the FCVO, there are five interfaces through which oil can leak to the casing: the piston-cup interface, the cup barrel interface, the interface between barrel and port plate, the interface between port plate and swash block and the interface between swash block and case.

For all FC rotary group prototypes, the piston-cup interface and the cup-barrel interface are regularly measured on a separate dedicated test stand. These measurements have shown that

the leakage in this interface is very small and, in a well designed FC rotary group, remains equally small after prolonged periods of operation.
The port plate can be balanced such that it is heavily pressed on the swash block, so the leakage in this interface is negligible.

Consequently, the leakage flow that was measured at the TU/e testbed (visualized in figure 5), originates from the interface between barrel and port plate and from the interface between swash block and case. As the leakage flow from 28 cm^3 fixed displacement FC pumps is lower than that of this first measured FCVO prototype, it may be concluded that either the extra leakage between swash blocks and case is rather high or the dynamic loads on the swash block cause that block to oscillate around its pivot point, in a way that the barrel cannot totally follow. Future experiments will be aimed at determining which of these two possibilities is causing the extra leakage and how this can be reduced.

Determining the mechanical torque losses is less easy. The next section describes a new method to extract them without having to rely on a measured displacement volume. Determination of the displacement from measurements is prone to errors that can have a considerable influence on the resulting prediction of the mechanical torque losses.

5 DETERMINATION OF THE MECHANICAL TORQUE LOSSES

In order to determine the mechanical torque losses, a power based approach is adopted. Starting from the mechanical input power at the pump shaft, the loss powers in all loss mechanisms except the mechanical torque losses are calculated and subtracted. The remaining power term represents the power associated with the mechanical torque losses.

So first, for each measurement, the mechanical power at the pump shaft, is calculated from the measured speed and input torque:

$$P_{mech} = T \cdot \omega \tag{3}$$

The first term to subtract, is the hydraulic output power defined conform ISO 4409:

$$P_{hydr} = q_{v,2} \cdot p_2 - q_{v,1} \cdot p_1 \tag{4}$$

This definition of the output power only takes into account the direct displacement work that is performed by the pump. It does not take into account the potential energy that is present in the compressed oil. If the oil were to be expanded adiabatically, this energy could be also be delivered at the pump output. If the thermodynamical properties of the oil are known, this 'compression energy' can be expressed as a fraction of the direct displacement work. For mineral oils used in the normal pressure range of hydrostatic applications, this fraction is approximately linearly dependent of the pressure difference over the pump.

For Mobil DTE25, this yields the following relationship between the compression power and the hydraulic output power:

$$P_{compr} = \frac{\Delta p}{365e5} \cdot 0.01 \cdot P_{hydr} \tag{5}$$

The in- and outlet flows through the swash blocks and the channels in the pump, result in a certain pressure drop over the pump. In the FC design, this pressure drop can be measured fairly easily, as the unit can be assembled without cups. In that case, the pre sprung barrels still keep the port plates and the swash blocks pressed to the casing. With the unit without cups, the pressure drop can be measured as a function of the flow. A quadratic formula could be fitted to the measurements. Multiplying this formula for the channel pressure drop with the flow, results in the following formula for the Power loss associated with the channel pressure drop:

$$P_{chan} = 2.16 \cdot 10^7 \cdot q_{v,2} + 1.26 \cdot 10^{11} \cdot {q_{v,2}}^2 \tag{6}$$

The next term to subtract is the power loss associated with the leakage of oil. Making the plausible assumption that the leakage from the high pressure port, over the port plate to the low pressure port is negligible and also neglecting the very low leakage from the low pressure side of the pump, the leakage power can be calculated to:

$$P_{leak} = p_2 \cdot q_{v,leak} \tag{7}$$

The last power loss term to take into account are the power losses associated with the restricted flow through the flow areas, defined by the 12 ports in each barrel and the two ports in each port plate. Especially at the transition of a displacement volume from one kidney to the next, the barrel ports are not in full contact with the kidneys in the port plate and the flow has to pass through the grooves in the pre-compression and the pre-expansion areas of the port plates. Then the flow path is very restricted and significant pressure drops can result.

In all axial piston pumps designed for a large field of operating points, compression and expansion of the oil in a displacement volume will be mainly done by oil flowing in from or out to the respective kidneys. Geometrical compression and expansion (those due to the piston movements) contribute for a very little part only. The FC principle is no exception to this rule

These 'port flow' losses have been calculated for the 28 cm^3 FCVO for 2^o, 4^o 6^o and 8^o swash angles. The calculations were done in the AMESIM simulation package, using a single piston model. The resulting power loss fields are shown in figure 4.

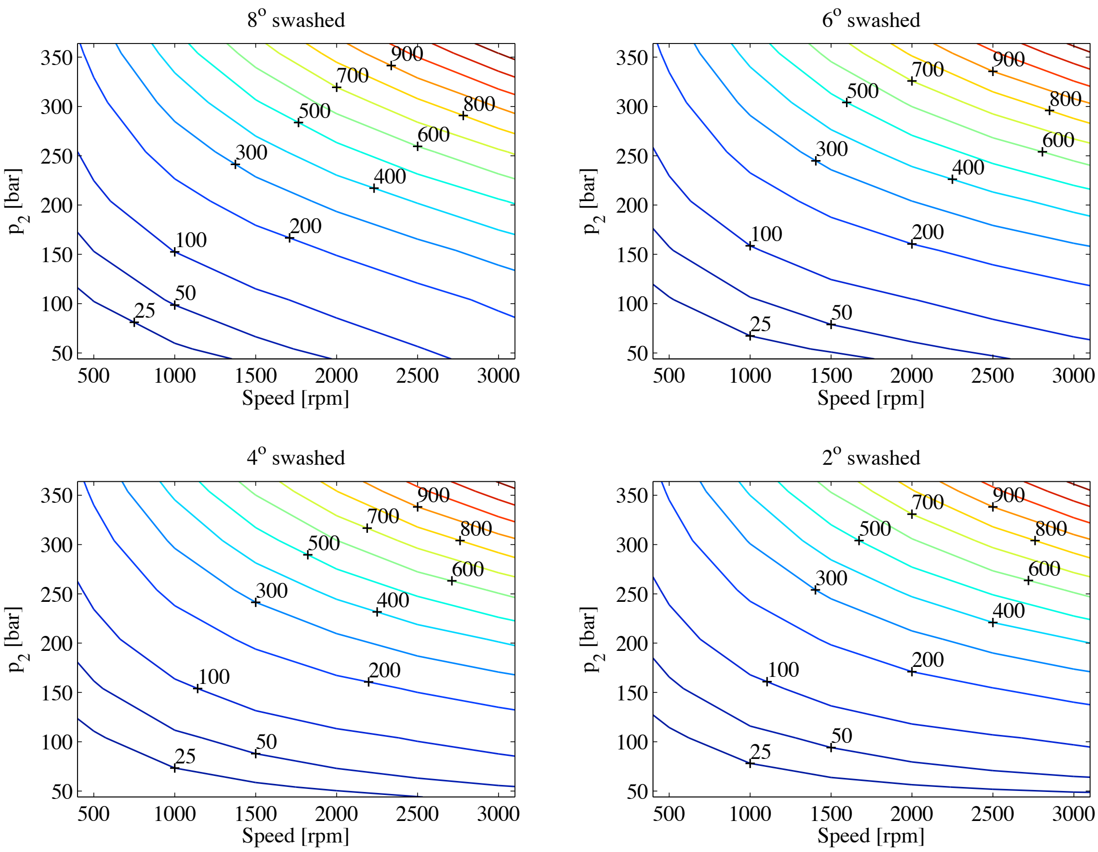

Figure 4: Calculated port flow power losses [W] at 2°, 4°, 6° and 8° swash angles.

Now all these power terms have been calculated, they can be subtracted from the measured mechanical input power in order to arrive at the mechanical torque loss power:

$$P_{torqueloss} = P_{mech} - P_{hydr} - P_{compr} - P_{chan} - P_{leak} - P_{portloss} \tag{8}$$

Division by the units angular speed, yields the mechanical loss torque:

$$T_{loss,mech} = \frac{P_{torqueloss}}{\omega} \tag{9}$$

Figure 5 shows the resulting reconstructed mechanical loss torques for the two measured swash angles. The measured leakage flows are also given. Measured and reconstructed points are indicated with an asterisk ($*$). Surfaces have been fitted to these points. Especially the reconstructed mechanical torque losses at 8° swash angle show some outlier points. It is not yet clear what has caused them. The fitted surfaces, however, seem very plausible.

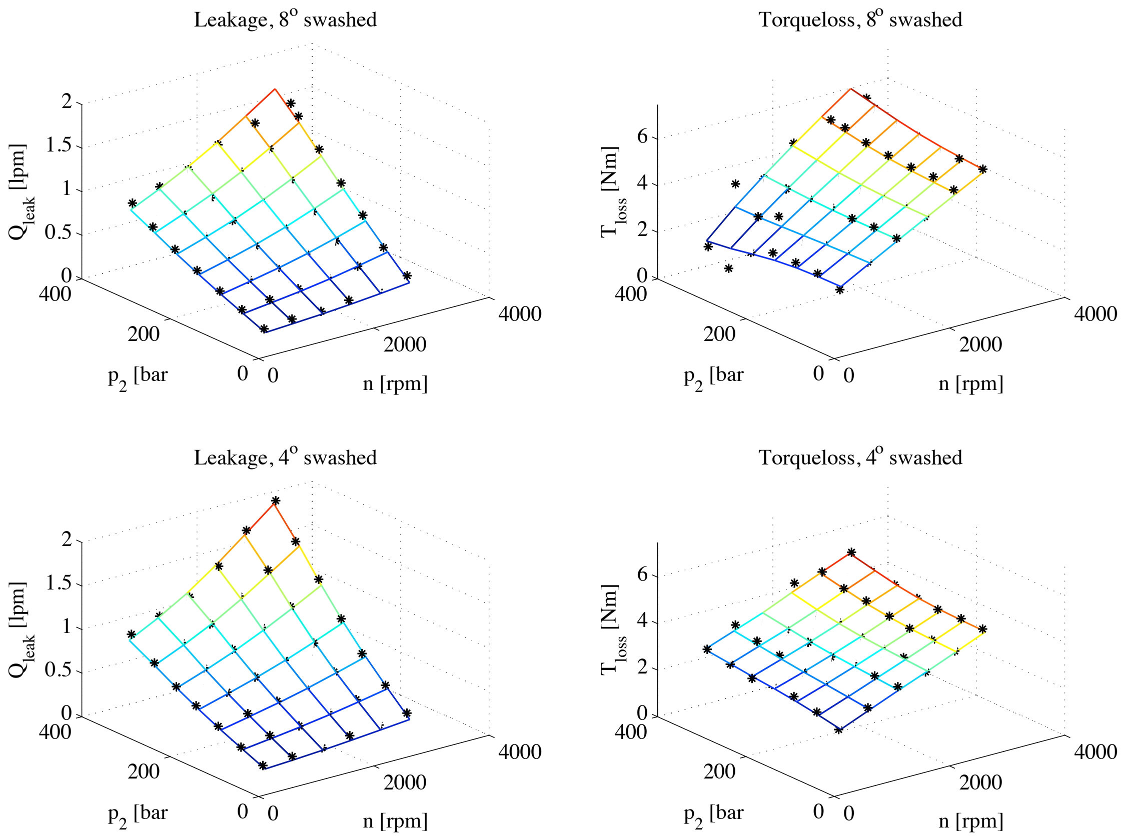

Figure 5: Measured leakages and reconstructed loss torques, 8° and 4° swash angles.

6 EXTRAPOLATION TO OTHER SWASH ANGLES

The fitted surfaces for the leakage and the torque losses at 4° and 8° swash angles can be used to predict leakages and torque losses for the 2° and 6° swash angles. To do that, the fitted surfaces for 4° and 8° are interpolated to generate the surfaces for 6° and extrapolated to generate the surfaces for 2°. The resulting surfaces are shown in figure 6. With these predicted leakage - and torque loss surfaces at 2° and 6° swash angles, the calculated port flow losses at 2° and 6° swash angles (figure 4) and the formulae given in section 5, the required mechanical power at each combination of rotational speed and output pressure, can be calculated using:

$$P_{mech} = P_{hydr} + P_{compr} + P_{chan} + P_{leak} + P_{portloss} + P_{torqueloss} \qquad (9)$$

The predicted overall efficiency can than be calculated:

$$\eta_t = \frac{P_{hydr}}{P_{mech}} \qquad (10)$$

The resulting predicted efficiency curves are shown in figure 7

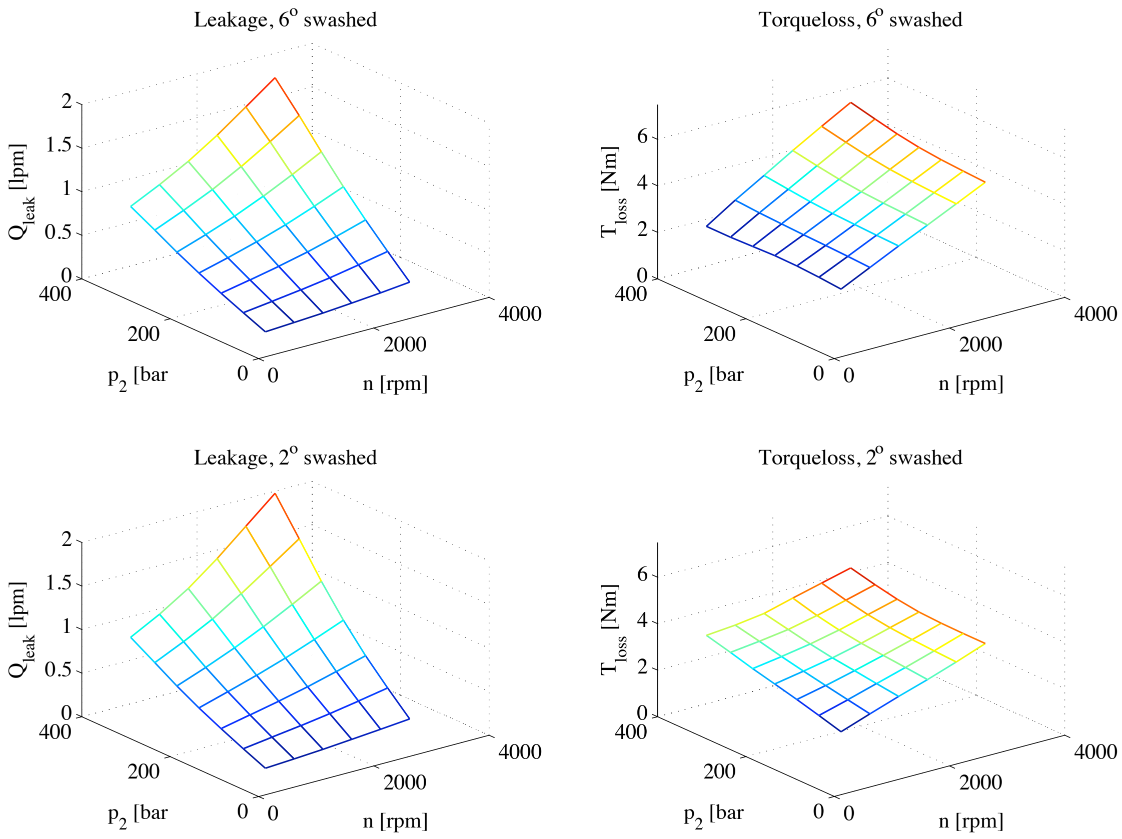

Figure 6: Predicted leakages and loss torques, 6° and 2° swash angles.

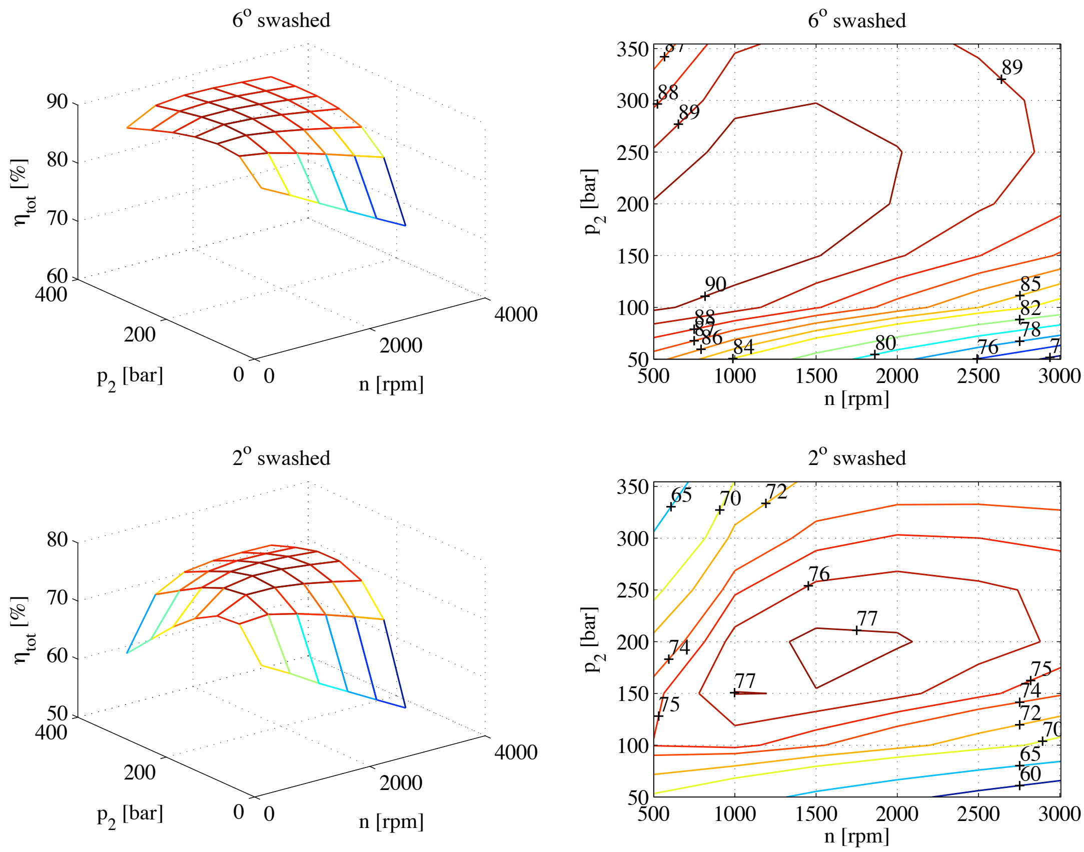

Figure 7: Predicted efficiencies, 8° and 4° swash angles.

7 CONCLUSIONS AND OUTLOOK

At the two measured swash angles, the total efficiency of this first prototype of a variable displacement, open circuit Floating Cup pump, is already quite high.

The measured leakage flows are somewhat higher than that of fixed FC pumps with an equal displacement. It has been shown that this leakage can only origin from two interfaces:

- The interface between the swash blocks and the case. Errors in the production of the two parts of this swash block bearings may have lead to gap heights which are too large . It is also possible that the swash block balance, which is deliberately set at the light side in order to ensure easy swashing, may be too light.

- The interface between barrel and port plate. The swash block may be oscillating around its pivot axis, because of the dynamic loads it experiences. The barrel may not be able to follow that movement.

Although the leakage is already very acceptable, decreasing it further will increase the units efficiency - especially at low swashing angles. Future experiments will be aimed at determining which of these two interfaces is causing the higher leakage and how this leakage can be reduced.

The new power-based method to isolate the mechanical torque losses from the measurements is very promising. The largest term in the mechanical torque losses is the torque loss in the interfaces between barrels and port plates. In order to balance this loss against the leakage losses from the same interface, it may be deduced from the total mechanical torque loss by subtracting estimates for the torque losses in the bearings, the friction losses between the cheeks and the spherical joints and the friction losses due to the centrifugal contact forces between pistons and cups.

The interpolation and extrapolation of the losses to other swash angles, and the subsequent prediction of the efficiencies at these swash angles, yields very plausible results. As soon as a full set of swash angles has been measured, the quality of the prediction will be checked.

REFERENCES

1. P. Achten, Z. Fu, G. Vael (1997) “Transforming future hydraulics: a new design of a hydraulic transformer”, Proc. of the 5th SICFP’97, Part 3, Linköping University

2. G. Vael, P. Achten, Z. Fu (2000) “The Innas Hydraulic Transformer - the key to the hydrostatic common pressure rail”, SAE paper 2000-01-2561

3. Werndin, R., Achten, P.A.J., Sannelius, M., Palmberg, J.-O. (1999) “Efficiency performance and control of a hydraulic transformer”, Proc. The 6th Scandinavian International Conference on Fluid Power SICFP '99, Tampere University.

4. Achten P.A.J., Vael, G.E.M., Oever, J. van den, Fu, Z. (2001) “‘Shuttle’ technology for noise reduction and efficiency improvement of hydrostatic machines”, Proc. The 7th Scandinavian International Conference on Fluid Power SICFP '01, Part 3, Linköping University

5. Achten, P.A.J., (2002). “Dedicated design of the Hydraulic Transformer”, Proc. IFK.3, Vol. 2, IFAS Aachen, pp. 233-248

6. Vael, G., van den Brink, T., Paardenkooper, T., and Achten, P. (2003). “Some Design Aspects of the Floating Cup Hydraulic Transformer”. In Bath Workshop on Power Transmission and Motion Control (PTMC2003).

7. Achten, P.A.J., M.P.A. Schellekens, H. Murrenhoff and M.Deeken, 2004, “Efficiency and low speed behavior of the floating cup pump”, SAE 2004-01-2553

8. Murrenhoff, H. and M. Deeken, 2004, “Wirkungsgradmessungen an einer Kolbenpumpe - Abschlussbericht”, RWTH-Aachen

9. Murrenhoff, H. and M. Deeken, 2004, “Langsamlaufverhalten einer Axialkolbenmaschine”, RWTH-Aachen

10. Palmen, A. and Murrenhoff, H., 2007, “Wirkungsgraduntersuchung an einer Floating Cup Axialkolbenpumpe, Abschlussbericht”, RWTH Aachen

11. Achten, P.A.J., 2005, “Volumetric losses of a multi piston floating cup pump”, Proc. Of the 50th National Conference on Fluid Power, NCFP-paper I05-10.2

12. Achten, P., Brink,T. van den, Schellekens, M., “Design of a variable displacement Floating Cup pump”, in The 9th scandinavian international conference on fluid power. 2005. Linköping, Sweden.

13. Platzer, T., Brink,T. van den, Vael, G.E.M., Achten, P.A.J., 2005, “The Design Of A Floating Cup Swash System”, Proc. Of the 50th National Conference on Fluid Power, NCFP-paper I05-10.2

14. P. Achten, G. Vael, T. Kohmäscher, M. Ibrahim Sokar (2008) “Energy efficiency of the Hydrid”, Proc. of the 6.IFK, Dresden, 1-2 April 2008, Volume II

15. P. Achten, G. Vael, H. Murrenhof, T. Kohmäscher, M. Inderelst (2009) “Low-emission Hydraulic Hybrid for Passenger Cars”, ATZ 5/2009 Volume 111

16. Achten, P., Brink,T.van den, Potma, J., Schellekens, M., Vael, G. (2009) “A four-quadrant hydraulic transformer for hybrid vehicles” in the 11th Scandinavian International Conference on Fluid Power, SICFP’09, Linköping Sweden.

17. Witt, K., 1974, ” Die Berechnung physikalischer und thermodyamischer Kennwerte von Druckflüssigkeiten, sowie die Bestimmung des Gesammtwirkungsgrades an Pumpen unter Berücksichtigung der Thermodynamik für die Druckflüssigkeit” PhD thesis, Technische Hogeschool Eindhoven.

18. Witt, K., 1976, “Thermodynamisches Messen in der Ölhydraulik: die Thermodynamik der Druckflüssigkeiten”. Ölhydraulik und Pneumatik, 20(9):603–614.