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A Robust Hydrostatic Thrust Bearing for Hydrostatic Machines

Peter Achten

Titus van den Brink

Georges Vael

Innas BV, Nikkelstraat 15, 4823 AE, Breda, Netherlands

ABSTRACT

In axial piston machines, the interface between the barrel and the port plate is one of the most complicated hydrostatic bearings. The gap height is governed by a complex combination of mechanical force and torque balances, which are in the end controlled by elastohydrodynamic and thermal effects. Even small deformations and wear can result in strong variations of the gap height, and consequently can have a significant effect on viscous and non-viscous friction. This paper describes a new design of a bearing geometry, which strongly reduces the effects of deformation and heat transfer and creates a more robust definition of the hydrostatic pressure forces in the sealing gap between the barrels and the port plates. The construction is also applicable for other thrust bearings in hydrostatic machines.

NOMENCLATURE

Δp	pressure differential	bar
K	isentropic bulk modulus	GPa
n	rotational speed	1/min
p	pressure	bar
P	power loss	kW
Q	flow	l/min
T	torque	Nm

1 THRUST BEARINGS IN HYDROSTATIC MACHINES

Hydrostatic machines often apply thrust bearings annex face seals. In slipper type pumps the interface between the barrel and the port plate, and between the slippers and the swash plate represent examples of these thrust bearings. There is abundant research in the field of thrust bearings and face seals, but most of it is outside the application of these bearings in hydrostatic machines. The high hydraulic pressure of these machines creates an extra challenge and dimension to the design of the bearing interfaces. The pressure load creates a deformation of the mechanical components, which influences the bearing capacity of the leakage path /Ach07/. Furthermore the high pressure differential across the leakage gap increases the heat generation in the gap. This results in stronger thermal expansion effects. The throttling and shearing of the oil flow also results in a temperature increase of the oil. Because of this the viscosity of the oil drops while passing the leakage gap, thereby creating a stronger pressure drop in the first half of the leakage path and a smaller pressure drop in the second half.

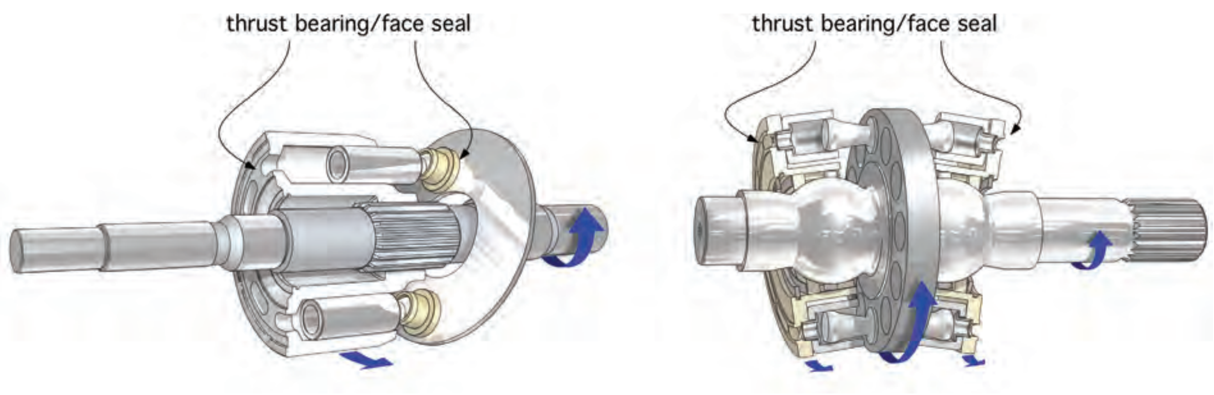


Figure 1: *Rotating parts of a slipper type (left) and a floating cup (right) principle*

The design of the axial thrust bearing is especially of interest for slipper type and floating cup designs. Both feature a large pitch circle of the barrel ports, which results in a long leakage path, as well as in a high relative velocity between the barrel and the port plate. The floating cup principle generally has two of these barrel port plate interfaces but lacks the piston slippers.

This paper describes a new design approach for pressurised thrust bearings in general, and the application in hydrostatic machines in particular. In the new design the high pressure of hydrostatic machines is used as a means to create a robust bearing, thereby counteracting other factors that affect the bearing capacity.

2 GAP HEIGHT BETWEEN THE BARREL AND THE PORT PLATE

A barrel of an axial piston machine has three degrees of freedom that are relevant for the behaviour of the axial bearing of the barrel (see ***Figure 2***):

- axial displacement in the z-direction
- rotation around the x-direction
- rotation around the y-direction

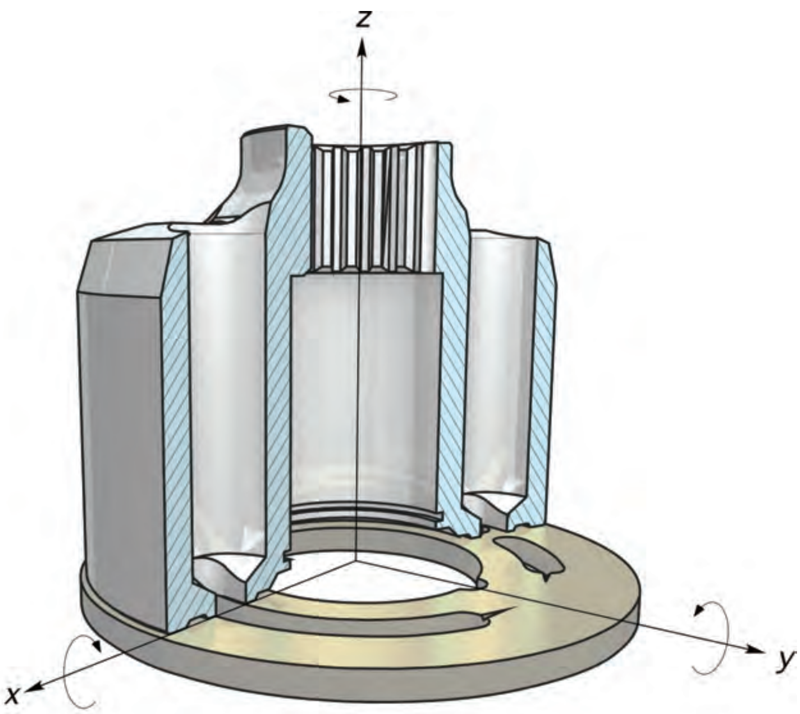


Figure 2: The barrel and port plate of a slipper type pump.

However, these three degrees of freedom are not the only parameters that define the gap between the barrel and the port plate. The barrel and the port plate can't be considered flat and completely rigid. Deformation of the housing, the port plate and the barrel due to thermal expansion and mechanical loading have a significant influence on the gap height /Ach07/. Also wear and production tolerances cannot be neglected but can have a significant and even dominating effect on the bearing capacity and the sealing performance.

The complicated interaction between machine design, thermal behaviour, operating parameters and the force and torque balance of the barrel seems yet to be too complicated to be modelled in an accurate way. Ivantysynova et al /Wie02, Hua03, Iva03, Bak09/ have published several papers in which the gap height between the barrel and the port pate is calculated to be in the order of 1 μm. These simulations are however not supported by direct measurements of the gap height. Instead, the leakage flow is measured and compared to the results of the simulated leakage flow, thereby finding a relatively good match /Hua03/. This is however by no means sufficient evidence that the simulations result in a correct calculation of the gap height and the tilt angle of the barrel, since there are many combinations of gap height, tilt angle, oil temperature, deformation and wear that can result in about the same leakage flow.

Another theoretical analysis of a slipper type pump was performed by Deeken /Dee03/. According to his simulations the height of the barrel gap varies between 1 and 5 μm. Unlike Ivantysynova, Deeken also performed direct measurements of the axial barrel position. The measurements were performed at three different points, evenly spread around the circumference of the barrel. According to these measurements, the gap height varies between 5 and 30 μm, much higher than the calculated gaps. Also Jang /Jan97/, Kim /Kim03, Kim05/, Van Ufford /Uff99/ and Wang /Wan09/ measured the gap height between the barrel and the port plate. All of these measurements resulted in large gap heights, some up to 100 μm. In all experiments the barrel showed a tilted position, having a minimum gap height at the high-pressure kidney. In slipper type pumps, the application of spherical port plates result in a reduction of the gap height variations /Kim05/. Also the application of bearing pads seems to have a stabilising effect on the gap height /Kim03/.

A tilted barrel position can result in a combination of high friction and relatively high leakage losses (See ***Figure 3***). The leakage will be high at those area's of the seal lands where a large gap is combined with a high pressure. High friction losses on the other hand will occur where the gap height is zero or close to zero. The local friction can be both coulomb and viscous friction. The viscous friction is strongly dependent on the gap height and the (local) oil temperature. The variation of the gap height across the circumference of the seal lands results in a strong variation of the local friction between the barrel and the port plate. This non-uniform distribution of the friction will create another torque load on the barrel, which can have a strong destabilising effect on the barrel bearing.

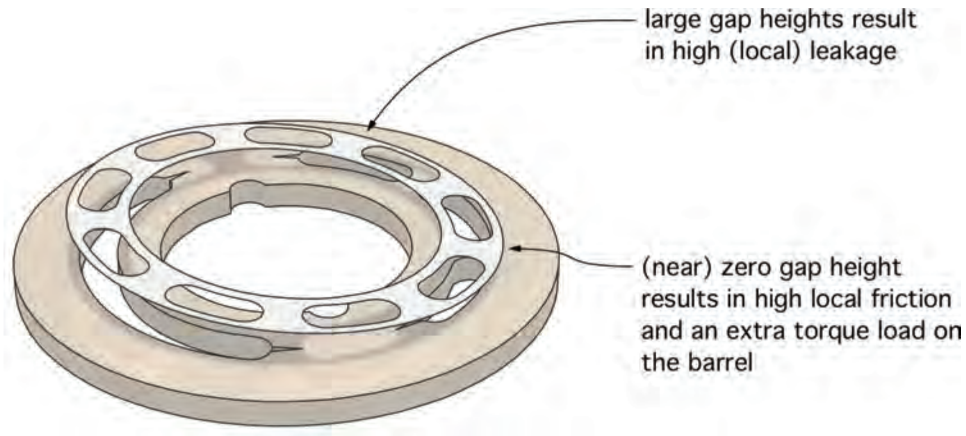


Figure 3: Tilted barrel position

There are a few mechanisms that can stabilise the barrel position and keep it afloat on the port plate:

- squeezing of the oil film results in buffer forces
- converging gap profiles (in the direction of the leakage flow) result in an extra lift force on the barrel

The effect of these mechanism is strongly dependent on the operating conditions (operating pressure, operating speed, oil temperature) as well as on wear and production tolerances. The deformation and the wear of the sealing area's can also create diverging gap profiles in which case the bearing capacity will be further reduced, or even become negative.

3 ANALYSIS OF THE LOSSES OF A FLOATING CUP PUMP

The floating cup, axial piston principle can be applied in pumps, motors and transformers. ***Figure 4*** shows a cross section of a 28 cc variable displacement floating cup pump. The floating cup principle is a multi piston design, generally having around 24 pistons, which is 3 to 4 times as much pistons as conventional slipper type and bent axis pumps and motors.

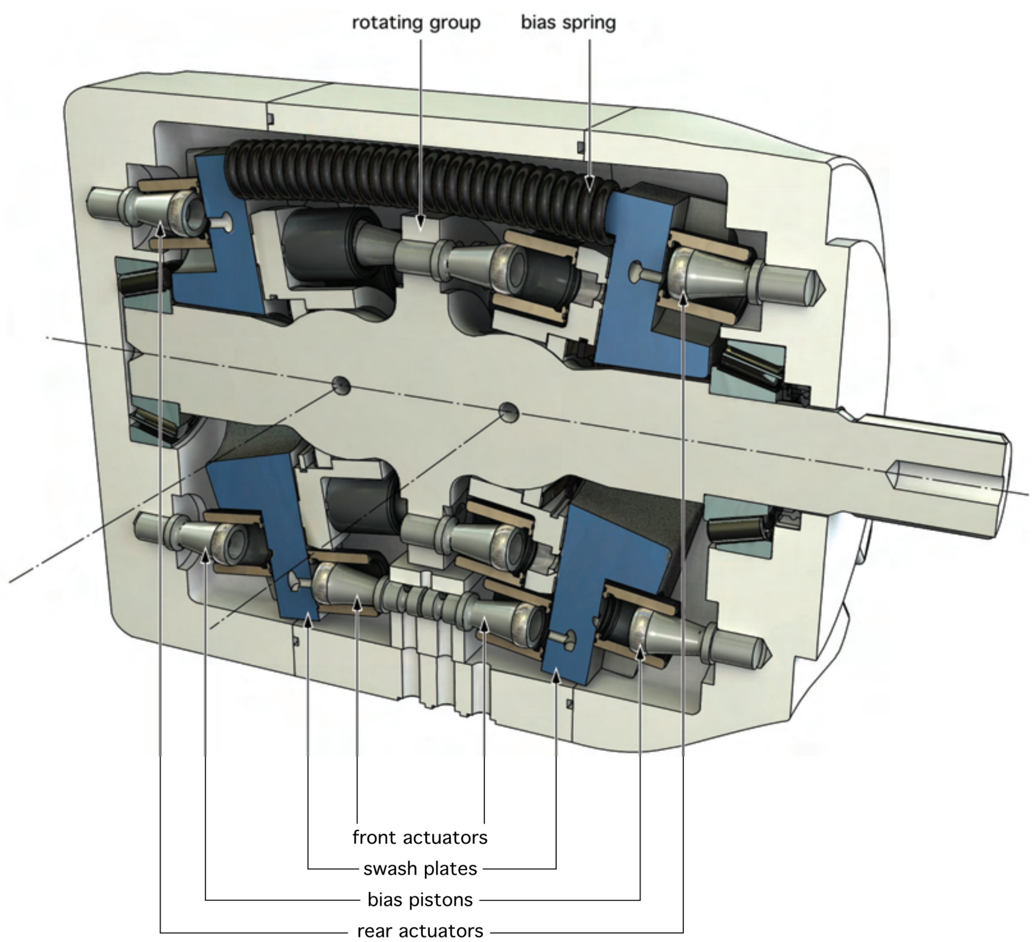


Figure 4: Cross section of a 28cc variable displacement floating cup pump

The efficiency of the 28 cc variable displacement floating cup pump (FCVP28) has been measured at the fluid power laboratory of Eindhoven University of Technology, in the context of a research cooperation sponsored by The Netherlands Organisation for Scientific Research (NWO), under the ‘Casimir’ program. ***Figure 5*** shows the test results at maximum displacement. The measurement has been performed according to ISO 4409. The oil temperature at the supply side of the pump was kept constant at a temperature of 40±2 °C. The average oil viscosity was 0.039 Pa·s. The supply pressure was kept constant at 3 bar. The performance of the pump has been tested for 6 different rotational speeds, varying between 500 and 3000 rpm, and 7 different pressure levels (Δp = 50…350 bar), resulting in a total of 42 stationary operating points.

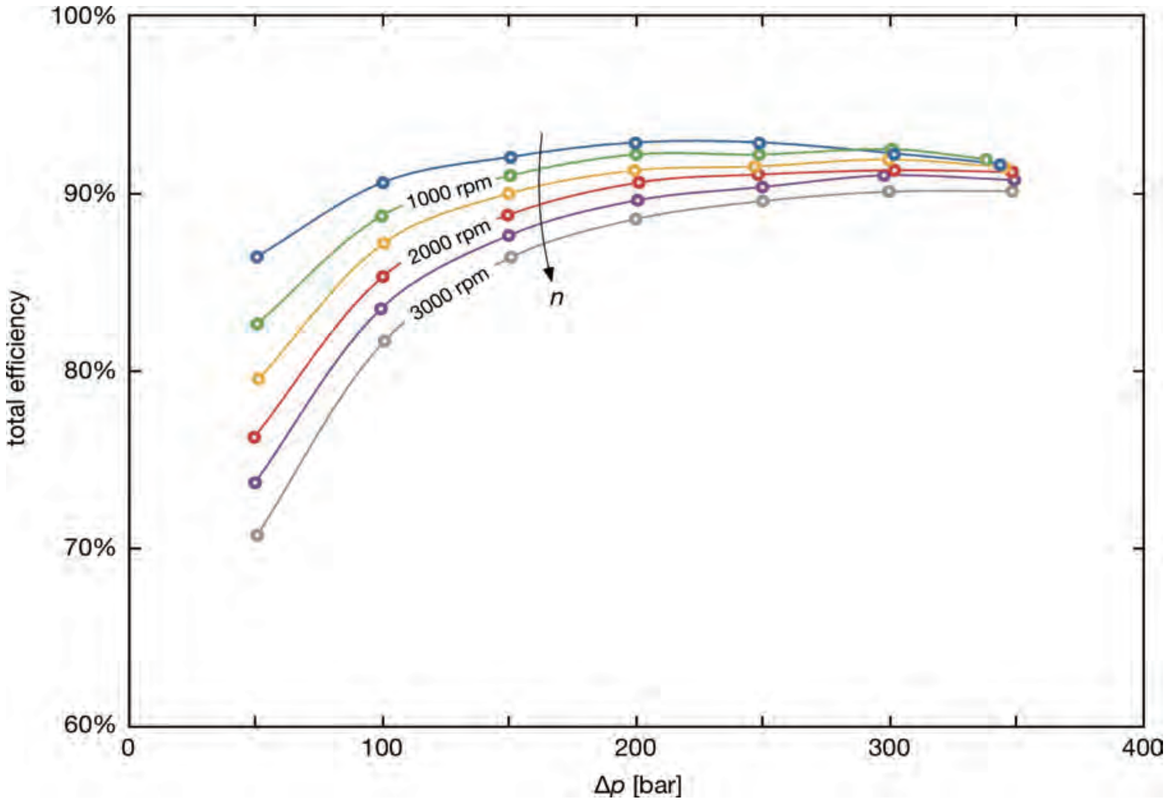


Figure 5: Measured total efficiency of the 28cc variable displacement floating cup pump (FCVP28) at full displacement (measurements TUe, ISO4409, 40°C, p_1 = 3 bar).

Taking the efficiency as defined in ISO 4409, the losses of the pump can be divided in nine different terms (see ***Table 1***). Some of these individual losses can be calculated, some can be measured, both with reasonable accuracy. Only the friction of the barrel running on the port plate is considered to be too difficult to be calculated or measured.

The churning losses, and the friction losses caused by the shaft seal, the bearings, the pistons and the cups have been measured in separate tests. These losses are small: they can almost be neglected in the considered speed domain. The impulse losses have been calculated. Even in the worst case, assuming there is no regeneration of the kinetic energy of the oil flow, the impulse losses are also negligible.

power loss term		determination
P_{leak}	volumetric losses due to leakage	are measured during the performance test
P_{comm}	losses due to commutation and flow resistance in the ports of the barrel and the port plate	can be calculated
P_{compr}	compression energy of the pressurized output flow	can be calculated
P_{shaft}	friction of the bearings and the shaft sealing	are measured in a separate test
P_{flow}	losses due to flow resistance of the housing and the swash blocks	are measured in a separate test
P_{piston}	piston and cup friction	are measured in a separate test
$P_{impulse}$	impulse losses due to the change of direction of the oil velocity wile entering and leaving the barrels	can be calculated
$P_{churning}$	churning losses	are measured in a separate test
P_{barrel}	friction of the barrels running on the port plates	not measured nor calculated

Table 1: Power loss terms and means of determination

The main factors to consider in the loss analysis therefore are:

- leakage
- losses due to flow resistance of the housing and the swash blocks
- commutation losses
- compression energy of the output flow
- friction between the barrels and the port plates

The losses due to leakage are measured during the performance tests. The flow losses of the housing and the swash blocks are measured separately. At an oil temperature of 40°C and a viscosity of 0,039 Pa·s the following equation can be derived for the power loss (in kW):

$$P_{flow} = 3{,}68 \cdot 10^{-4} \cdot Q^3 + 8{,}90 \cdot Q^2 \tag{1}$$

The commutation losses are calculated by means of a AMESim-model of the pump. ***Figure 6*** shows the calculated loss power for the FCVP28 at maximum displacement.

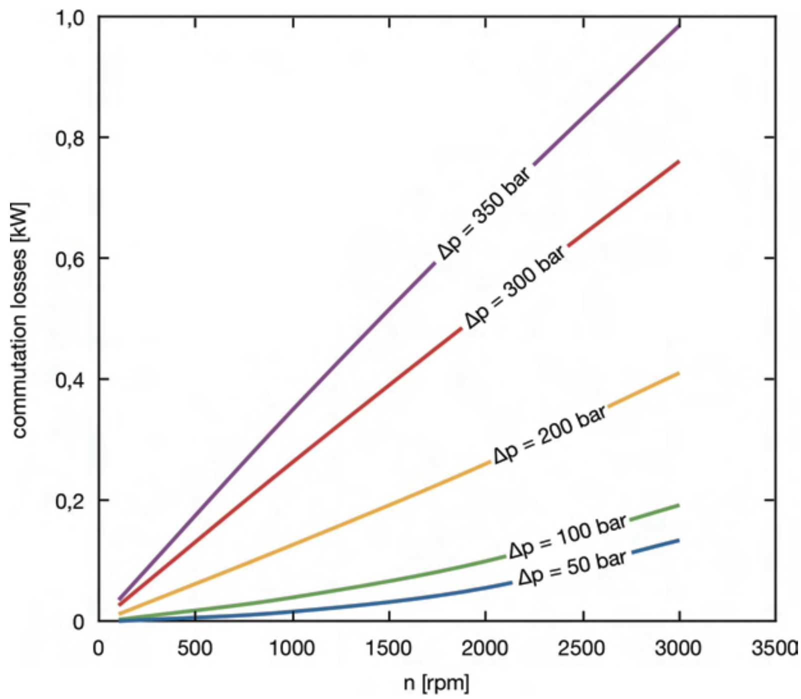


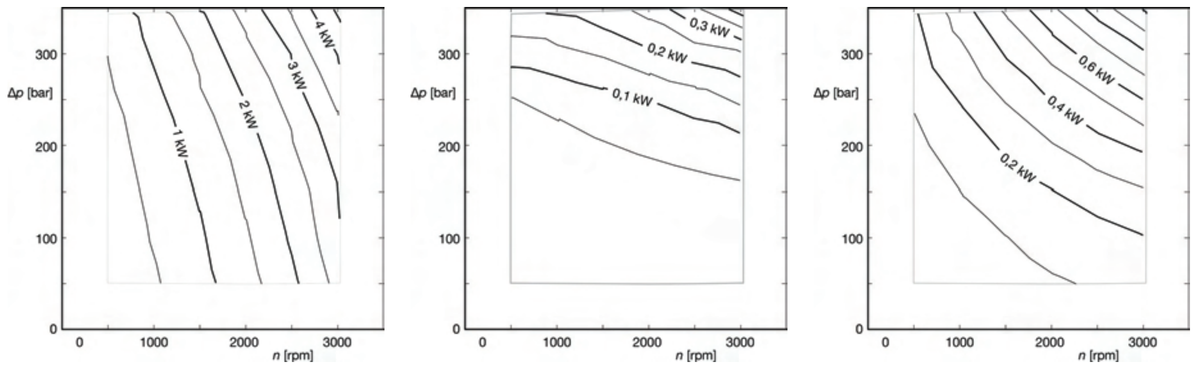
Figure 6: Commutation losses FCVP28 at full displacement

The compression energy taken by the output flow can also be calculated:

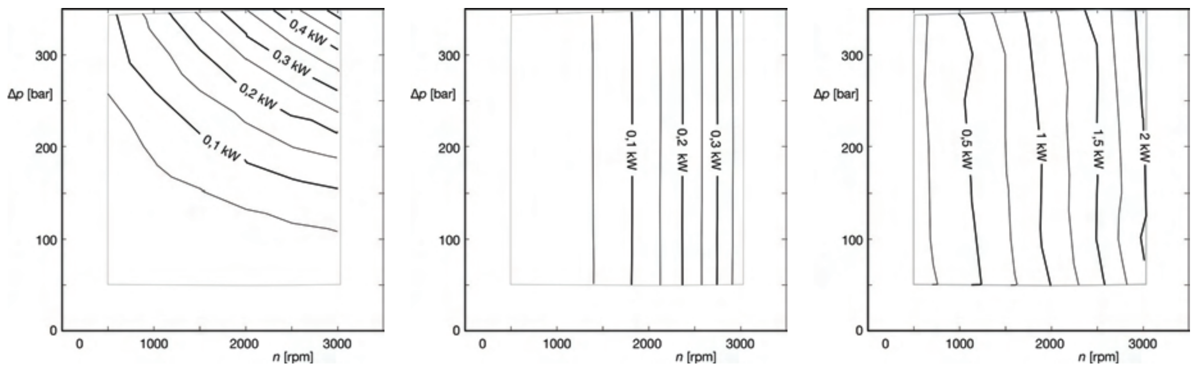
$$P_{compr} = \frac{1}{K} \cdot \frac{\left(p_2^2 - p_1^2\right)}{2} \cdot Q \tag{2}$$

The only remaining loss term is the unknown friction between the barrels and port plates. Subtracting all the loss terms above from the total loss that is measured during the performance measurements derives this loss:

$$P_{barrel} = P_{loss} - \left(P_{leak} + P_{comm} + P_{compr} + P_{shaft} + P_{flow} + P_{piston} + P_{impulse} + P_{churning}\right) \tag{3}$$



a) total loss P_{loss} b) leakage loss P_{leak} c) commutation loss P_{comm}



d) compression loss P_{compr} e) flow loss P_{flow} f) barrel friction loss P_{barrel}

Figure 7: Analysis of the losses of the FCVP28 at full displacement

The total loss and the main loss terms are displayed in the plots of ***Figure 7***. The friction between the barrels and the port plates is the dominating loss factor. On average, across all measured operating points, more than 50% of the total energy losses is caused by this friction. Excluding the compression energy of the output flow (which is not really a dissipative loss), about 63% of the losses can be contributed to the barrel friction.

The friction between the barrels and the port plates can be calculated in terms of the torque loss T_{barrel}. The diagram of ***Figure 8*** shows the barrel friction as a function of the rotational speed of the pump for different Δp-values.

The barrel friction is most and for all depending on the rotational speed of the pump. The relationship between the friction torque and the speed is almost linear. The offset could be explained by a coulomb friction, indicating that there is somewhere a contact between the barrel and the port plate. The proportional relationship also indicates a substantial viscous friction, as would have been expected.

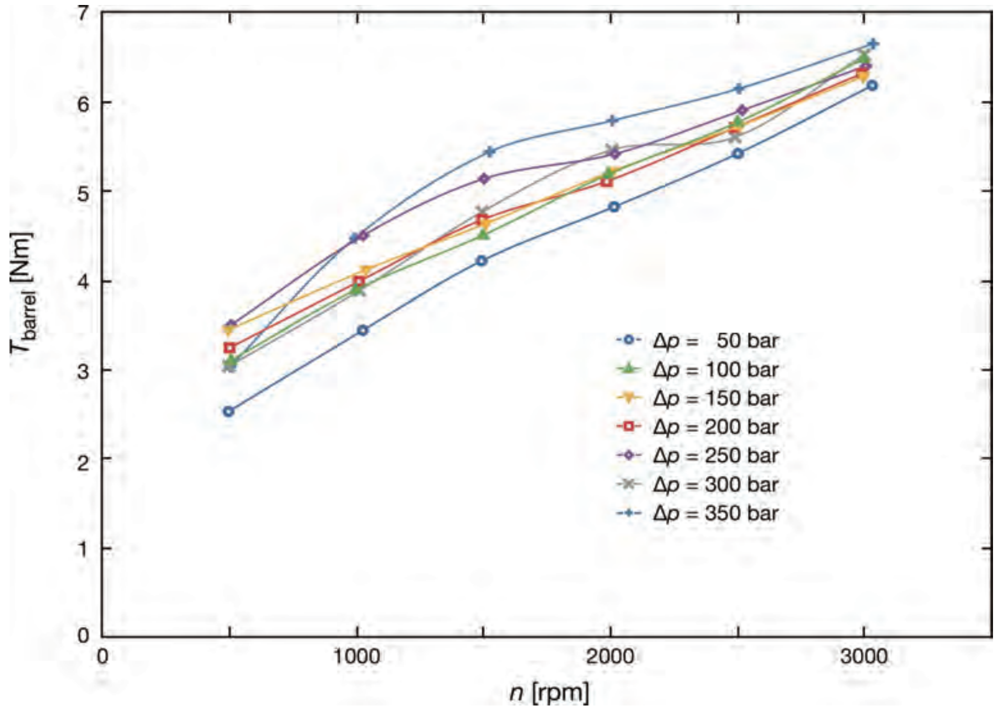


Figure 8: Torque loss due to friction between barrels and port plates in a 28 cc variable displacement floating cup pump

4 A NEW BEARING DESIGN

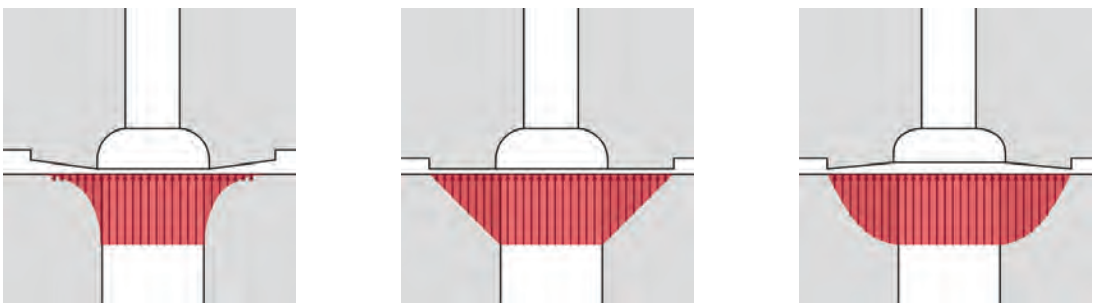
As in other axial piston machines, the force and torque balance of the barrel of a floating cup pump is quite delicate and extremely complicated. The forces acting on the barrel are:

- hydrostatic force generated by the oil columns in the cups
- hydrostatic force generated by the barrel ports
- hydrostatic force generated by the pressure field of the seal lands
- poiseuille-couette, viscous and (eventual) coulomb friction forces
- centrifugal forces
- impulse forces
- spring force

The centrifugal forces are generated by the cups /Bri06/. The impulse forces are due to the change of the direction of the oil flow, while entering and leaving the rotary group. The spring force is necessary to create a defined position of the barrel after assembly or at low pump pressures. In the floating cup pump the spring force is around 200 N, which is relatively low compared to other axial piston machines.

At most operating conditions the hydrostatic forces are dominant. Having 6 pressurised pistons per barrel with a diameter of around 12,2 mm, a pump pressure of 350 bar results in an axial force of 24,5 kN. This large axial force is counter balanced by the hydrostatic force generated by the barrel ports and seal lands. If the two hydrostatic forces not equally large or exactly in line there will be a remaining axial force and torque. The result is that in most cases the barrel will seek some kind of contact with the port plate. Due to commutation and other effects, the position and size of the axial forces changes while the barrel runs on top of the port plate. Consequently the 'point of contact' will change as well. In case of a tilted barrel position this means that oil will be squeezed constantly, thereby creating extra buffer forces.

The hydrostatic force and torque balance is strongly determined by the pressure field acting on the seal lands of the barrel. In its most simple form, a cross section of the seal land, in the direction of the leakage path, would show a triangular pressure profile (***Figure 9.b***). In reality, the pressure profile is not linear. Due to deformation, thermal expansion, oil temperature variation, wear, and barrel tilting the pressure field in the cross section of the gap can be much smaller (***Figure 9.a***) or larger (***Figure 9.c***). Even gap profile variations in the order of 1 μm can have dominating effects on the force and torque balance of the barrel, thereby strongly influencing the friction and leakage.



a) divergent gap b) parallel gap c) convergent gap

Figure 9: possible pressure profiles in the gap between the barrel and the port plate

Convergent gap profiles are favourable since they create an extra lifting force as soon as the barrel gets close to the port plate /Ram66, Lub99, Man02, Man04/. The convergence of the gap profile is however extremely difficult to create during manufacturing. Furthermore the convergence changes due to wear, and operating conditions as oil temperature, operating speed and operating pressure. To overcome these disadvantages a new bearing structure has been designed. In the new design (***Figure 10***) the seal land is split in the middle into two sealing areas having multiple shallow pockets. Each barrel port has at least one pocket.

The left side of figure 10 shows a cross section of the seal lands. Each pocket has one or more individual, restricted connections to the barrel port. These connections can be realized through the barrel or via the port plate. The extra restriction that supplies oil to the pockets can also be variable, thereby creating an active control over the bearing and sealing function. This paper however discusses only the most simple configuration, with a small groove in the first seal land between the barrel port and the pocket, as is illustrated in *Figure 10*.

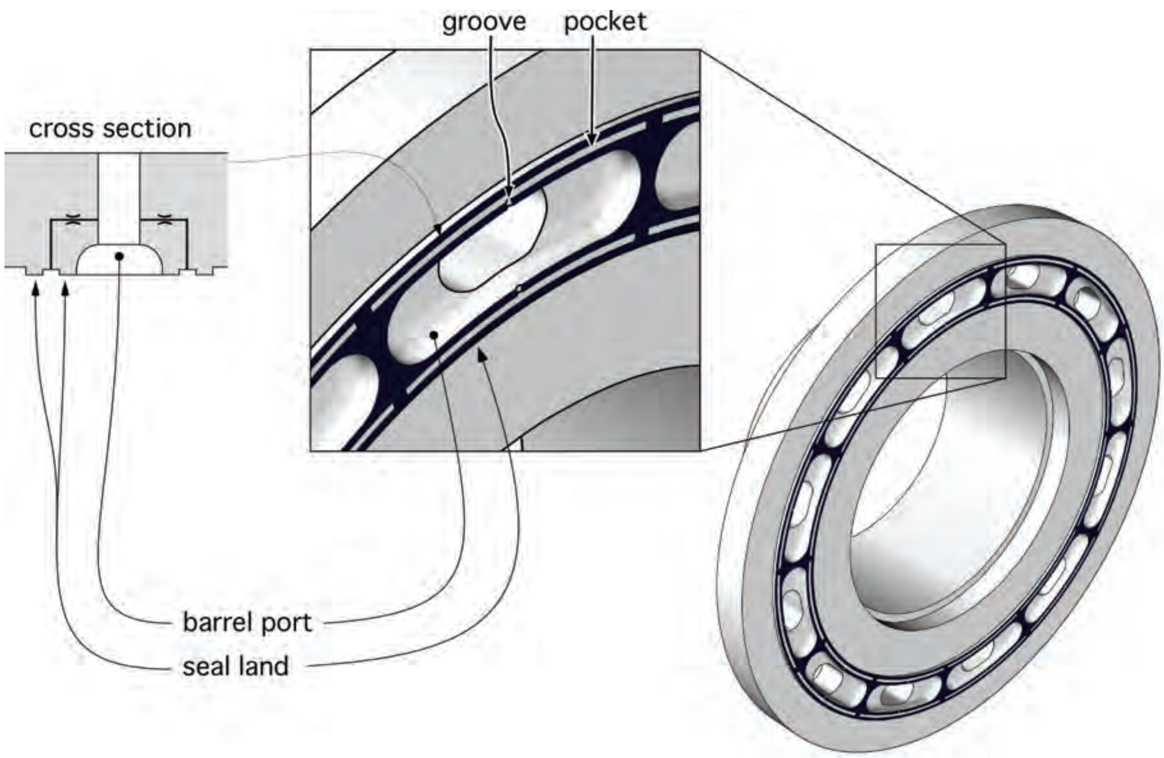
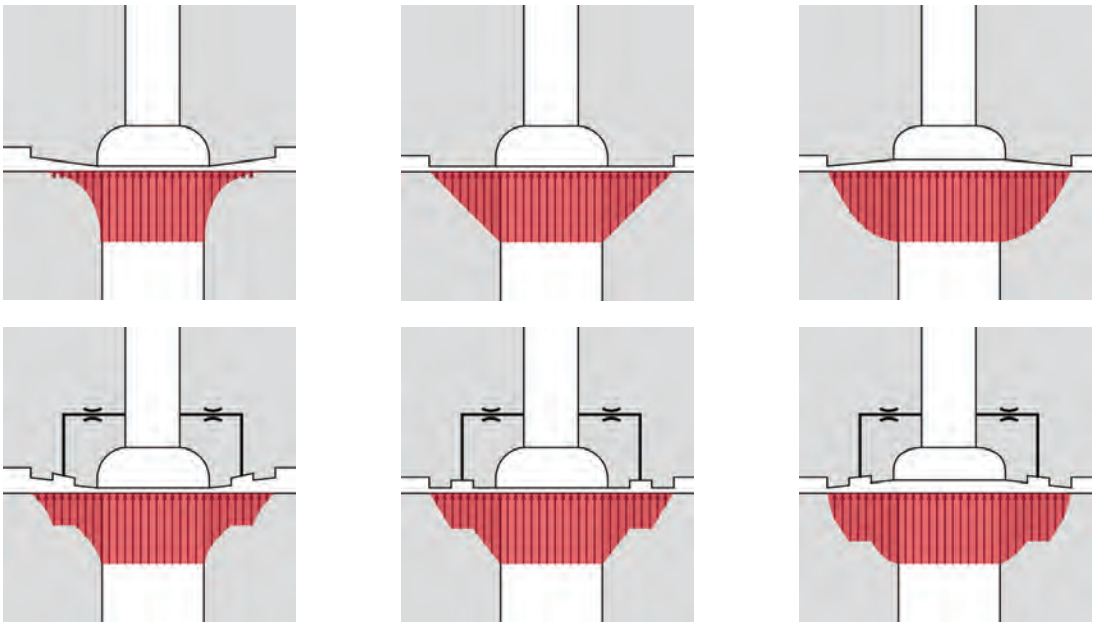


Figure 10: New bearing and sealing area design

Aside from the individual connection, each pocket is also connected to the barrel port and to the pump case by means of the leakage paths of the (split) seal lands: oil is supplied to the pockets by means of the individual restrictions (i.e. the grooves) and the seal land between the pocket and the barrel port. On the other side, oil is leaking away via the seal land between the pocket and the case. Essential for the new bearing structure is that the extra restriction via the grooves is in principle nearly independent of the gap height between the barrel and the port plates. Even when (local) gap height is reduced to zero there will still be oil leaking via the grooves, thereby creating a pressure in the pockets that will lift the barrel again.

Figure 11 shows the resulting pressure profiles with and without the pockets for various gap profiles. The pockets create a much more constant pressure profile, especially when the gap profile is divergent. On the other hand the pockets still maintain and even enhance the extra lifting force created by a convergent gap profile.



a) divergent gap   b) parallel gap   c) convergent gap

Figure 11: Pressure profiles with and without pockets

5 TEST RESULTS

The pockets have been implemented in the barrels of the variable displacement floating cup pump. ***Figure 12*** shows a detail of the laser manufactured grooves and pockets. The grooves have a width and depth of 50 µm. With this configuration, the same performance test and loss analysis has been performed as in paragraph 3. ***Figure 13*** shows the efficiency increase, which results from the application of the pockets. The pockets cause an increase of the leakage losses, obviously especially at high pump pressures. The leakage losses are however more than compensated by a reduction of the friction between the barrels and the port plates.

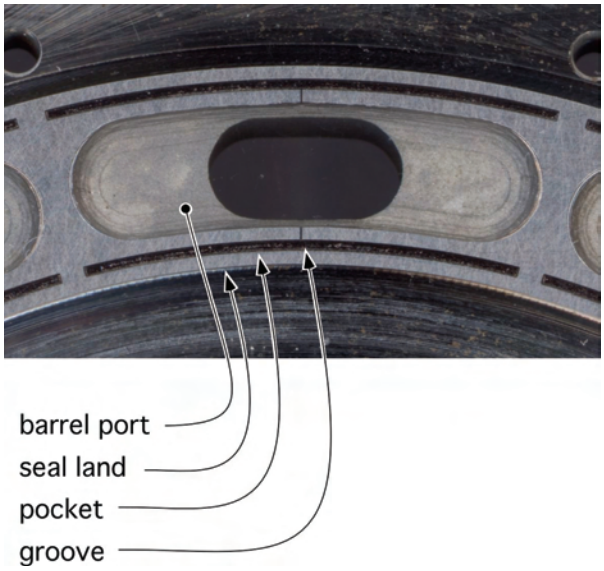


Figure 12: Seal lands with pockets and grooves

The reduction of the barrel friction becomes apparent in the comparison of torque losses due to barrel friction (***Figure 14***). For a pump pressure of 50 bar the reduction is limited to 1 Nm. At other pump pressures the reduction is 2 to 3 Nm. The reduction is larger for higher pump pressures, indicating the stronger effect of the pockets at higher pressures. The torque losses still seem to be dominated by the viscous friction, which, judging from the unaltered gradient of the torque loss lines, seems to be unaffected by the pockets. It should be noted that this is just a first test result of the pocket system, having just one configuration of pocket, barrel and groove dimensions. The dimensions of the barrel ports, the seal lands, the pockets and the grooves all have a large influence on the strength and stiffness of the new bearing.

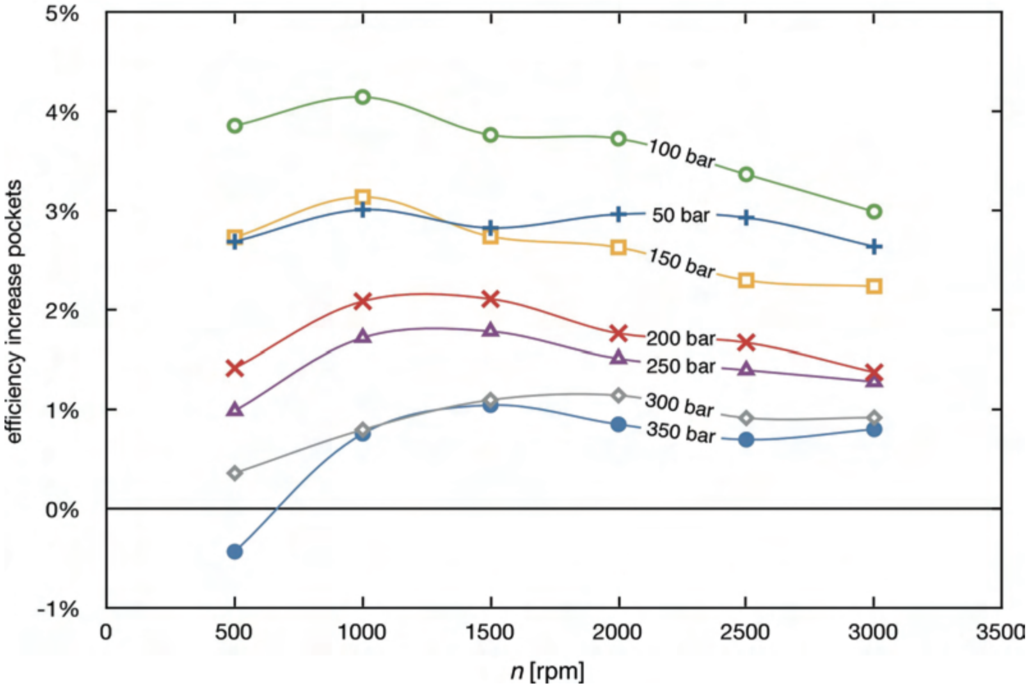


Figure 13: Effect of the application of pockets and grooves on the total efficiency of a 28 cc variable displacement floating cup pump.

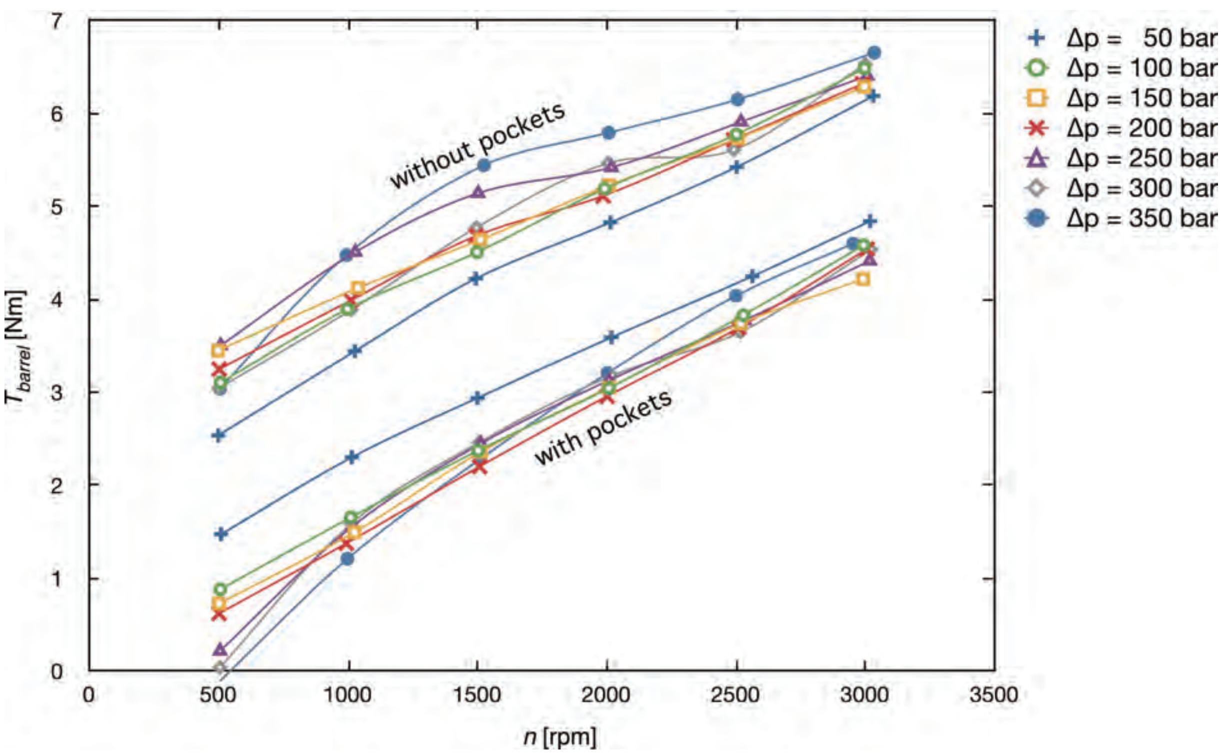


Figure 14: Effect of the application of pockets and grooves on the total efficiency of a 28 cc variable displacement floating cup pump.

CONCLUSIONS AND OUTLOOK

The conventional split between hydraulic-mechanical losses and volumetric losses is inadequate for understanding and analysing the energy dissipation in hydrostatic machines. A new and more elaborate analysis is developed to get a better understanding of the losses /Vae09/. The new method is applied on a variable displacement floating cup pump, revealing that most of the losses are due to the friction between the barrels and the port plates. A new bearing design has been implemented and tested. The pockets and grooves have resulted in a significant improvement of the overall efficiency. With the new bearing design, the peak efficiency has been increased from 93% to 95%. The efficiency improvement is entirely due to a strong reduction of the friction between the barrels and the port plates.

The results presented are just a first indication of the potential of the new bearing structure. More research needs to be done, both theoretically and empirically to get a better understanding of the friction losses and gap height of the interface between barrels and port plates in axial piston machines in general and floating cup machines in particular.

ACKNOWLEDGEMENT

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 Netherlands Organisation for Scientific Research (NWO), under the 'Casimir' program.

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