Some Design Aspects of the Floating Cup Hydraulic Transformer

G.E.M. Vael, T.L. van den Brink, T. Paardenkooper, P.A.J. Achten
Innas BV, Breda, The Netherlands

SYNOPSIS

This paper focusses on some important design aspects of the first Floating Cup (FC) prototype of an Innas Hydraulic Transformer (IHT). The IHT and the FC displacement principle are briefly introduced and equations are derived for the scaling of the FC concept to larger displacements. It is shown that the balancing of the port plate largely determines the control torque, a very important aspect of IHT design.
In the IHT design, pistons generally change their port connection while they are still pumping fluid. Port plate imbalance, excessive noise production and decreased efficiency may result. The paper shows that these phenomena are positively influenced by the large number of pistons which is possible in the FC concept.

1 INTRODUCTION

Since the introduction of the Innas Hydraulic Transformer (IHT) [1], prototypes have been built by changing the port plate and the end cap of standard axial piston displacement units. With these prototypes the functional viability of this hydraulic transformer type has been demonstrated and a number of initial shortcomings have been identified and improved [2, 3]. Toward the end of 2001 it became clear that if the full potential of the IHT concept had to be demonstrated, a dedicated design would have to be made. The subsequent design process led to the Floating Cup (FC) displacement principle [4]. Very soon it was realised that the FC displacement principle held a lot of promise for pumps and motors too. As a successful applications in these would lead to an earlier adoption of the principle, further design and testing activities were concentrated on pumps and motors. As soon as the potential of the FC principle had been confirmed [5], the design of the first FC IHT prototype was continued. After a brief introduction of the IHT and the FC displacement principle, this paper centres on some important aspects of the design of this prototype.

2 OPERATION OF THE IHT

The operating principle of the IHT has been described in several earlier publications [1 , 6]. Here, only a brief description will be given, refering to the Bosch Rexroth A2FM based IHT prototype shown in figure 1. When the barrel of this IHT is rotating, the ports in the surface of the barrel

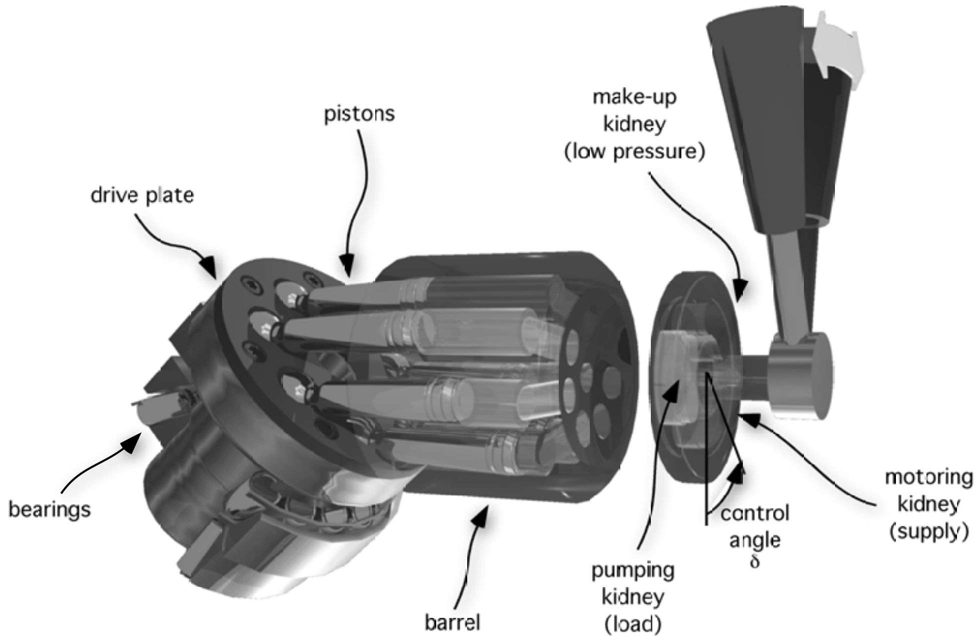

Figure 1: An A2FM based IHT prototype

will subsequently pass the three kidneys in the port plate. The pistons to which these ports connect are subjected to the pressure in those kidneys and will exert a corresponding torque to the unit's axle. The barrel is synchronized to the axle, in this case by the pistons themselves. One of the kidneys in the port plate (the 'motoring' kidney) is connected to a pressure rail which is kept at a semi constant pressure (typically 30 MPa). Another kidney (the 'make up' kidney) is connected to a rail at a low pressure (typically 1 MPa). The two rails together form a hydraulic mains system, from which the IHT can take power. The power is transfered to the third kidney (the 'pumping' kidney), which is connected to a load.

By summing the momentary torques generated by the pistons connected to one of the kidneys and by averaging this sum over time, the average torque generated by that kidney can be calculated. As the axle is not mechanically connected to its environment, the axle and barrel assembly is only moving at a stationary speed if the sum of the three average kidney torques is zero. If this is not the case, the remaining torque will accelerate (or decelerate) the assembly. As a consequence, the IHT will pump more (or less) oil to the load, thereby compressing or expanding the oil column between the IHT and the load. The acceleration or deceleration ceases if the load pressure has reached its equilibrium value: the value at which the torque generated by the load kidney balances the torque generated by the other two kidneys.

At a given port plate geometry and given values for the rail pressures, the equilibrium load pressure is a function of the port plate control angle δ only.

3 THE FLOATING CUP PRINCIPLE

Figure 2 shows the rotary group of the 50 cm^3 Floating Cup IHT currently under development. In this Floating Cup design, 12 pistons are arranged on each side of the central rotor. This number of pistons was chosen after a thorough analysis of the flow pulsations and the start up and low speed behavior as a function of the number of pistons. A description of these phenomena and how they can be expected to scale with the swept volume are beyond the scope of this paper.

The rotor is fixed rigidly on to the axle and the pistons are fixed rigidly in the rotor. Each piston has its own cup-like cylinder and all the ‘cups’ of one rotor half rest on a common ‘barrel plate’. The opposite side of each barrel plate runs on the inclined face of a port plate. The inclination ensures that the pistons move relative to the cups and pump oil. In order to keep the loads on the pistons within limits and in order to keep the friction within limits if the balanced piston ring design described in [4] is used, the angle of inclination should not become too large. For the first 10 cm^3 FC design a maximum angle of ca. 9^o was calculated, which can be shown to be constant when the unit is scaled. A hole in the bottom of each cup connects to a corresponding

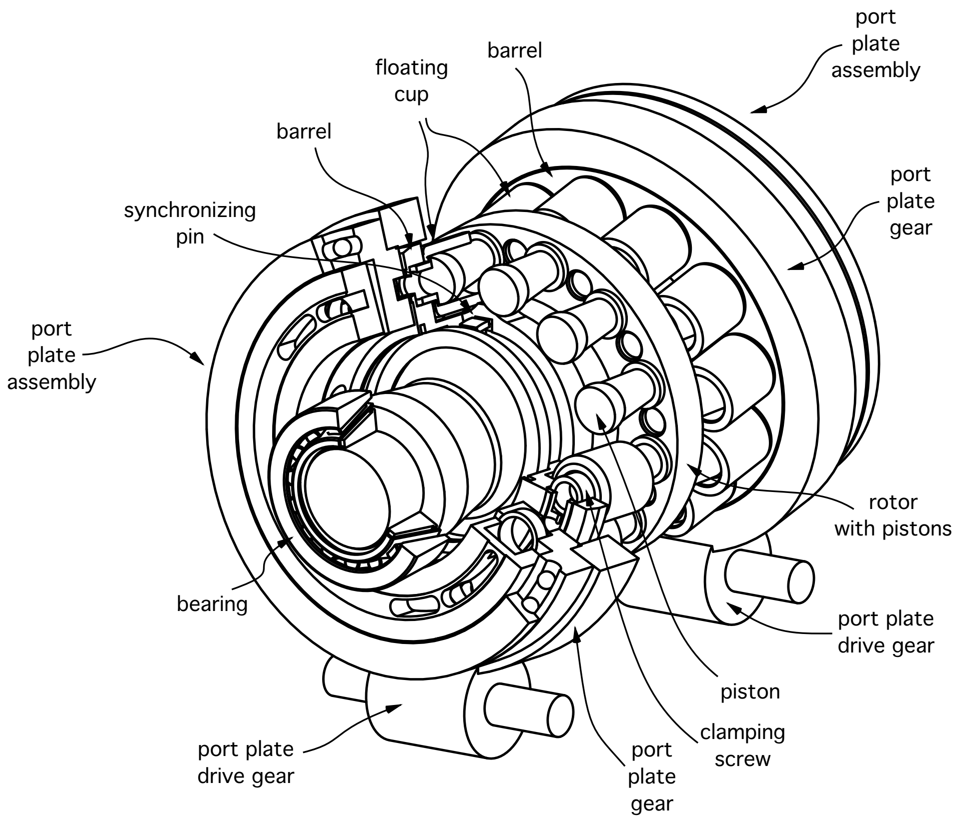

Figure 2: The Floating Cup displacement principle

hole in the barrel plate. The hole in the cup and the seal land around it are carefully tuned to give a resulting hydrostatic force that is large enough to press the cup down on the barrel plate. The cups are thus free to move over the barrel plate and this ‘floating’ motion provides the degree of freedom which allows the cups to follow the ellipsoidal track which is the projection of the piston circle on the barrel. The relative motion between cups and barrel is so small that even at somewhat higher contact forces between the cups and the barrel, the energy dissipation in this

interface is low. Thus, the contact forces are not minimized stringently but in order to avoid wear of the interface they are not set too high.
In each cup a clamping screw ensures that the cups do not break out or lift off at high rotational speeds and low internal pressures.

The barrel plates are connected to the axle by means of a spherical joint. In each joint, two simple synchronizing pins force the barrel plate to rotate synchronously with the axle, thereby reducing the three rotational degrees of freedom of the sperical joints, to two.

4 SCALING THE FLOATING CUP PRINCIPLE

The first FC design studies have been performed for a rotary group with a nominal displacement volume of 10 cm^3. The design was meant for an IHT and had 18 pistons. Although its design was already advanced, only a kinematic prototype of this rotary group was built. As described in section 1, the FC development was soon redirected to pumps and motors and the first functional FC prototypes were 28 cm^3 pumps with 24 pistons.

In this section, it will be shown how the main dimensions of any FC rotary group, can be scaled from the 10 cm^3 and 28 cm^3 FC prototypes that have already been worked out in detail. The scaling is used in the design process, in order to quickly arrive at the dimensions of the new design. Apart from that, scaling laws also provide insight in the basics of the concept.

The geometry of an FC rotary group and the parameter definitions are given in figure 3.
The pistons are arranged on the rotor on a circle with radius R_r. In order to maximize the unit's

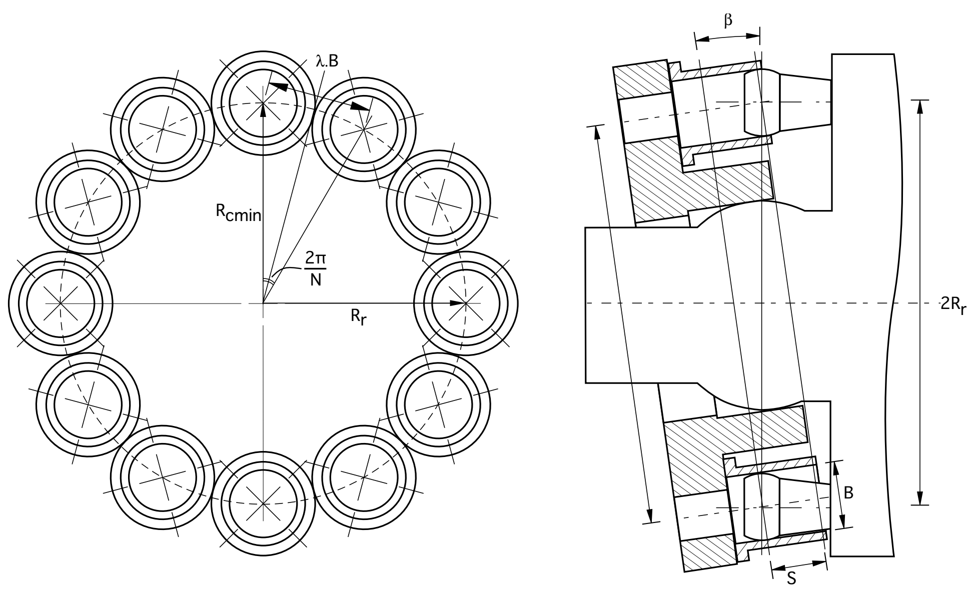

Figure 3: FC geometry and definitions

power density, R_r is chosen as small as possible. The minimum value of R_r is determined by the number of pistons, the dimensions of the cups and the trajectory the cups follow on the barrel plate.

The dimensions of the cups are largely determined by the requirement that the walls of the cups have to be thick enough to prevent them from expanding too much under pressure, as this could lead to an excessive leakage flow between plunger and cup. This leakage flow can be estimated using the well known equation for the laminar flow through a cylindrical gap with an average diameter of B, length l and height h_0 :

$$Q_c = \frac{\pi \cdot B \cdot {h_0}^3 \cdot p}{12 \cdot \eta \cdot l}, \tag{1}$$

in which p is the pressure difference over the gap and η is the kinematic viscosity of the fluid. In [7], it has been shown that the same equation can be used for the laminar flow past a sphere in a cylindrical hole, if l is replaced by an equivalent length:

$$l_{eq} = \frac{3}{8} \cdot \pi \cdot \sqrt{B \cdot h_0} \tag{2}$$

For the scaling of the leakage flow around the pistons, h_0 is equated to the radial expansion of the cup under pressure. This expansion can be estimated using the equation for a thin-walled tube (wall thickness t) of infinite length, subjected to a uniform internal pressure p:

$$h_0 = \frac{B^2 \cdot p}{4 \cdot t} \tag{3}$$

With equations 1, 2 and 3 it can be seen that the laminar leakage from the cups Q_{c_l} scales with:

$$Q_{c_l} \sim \frac{B^{5.5}}{t^{2.5}} \tag{4}$$

When the floating cup principle is scaled, the ratio of this leakage to the delivered flow is an important parameter, as this largely determines the volumetric efficiency of the unit. In order to calculate how the delivered flow scales with B, the relationship between B and t has to be chosen. If the floating cup concept is scaled in such a way that t varies linearly with B equation 4 reduces to:

$$Q_{c_l} \sim B^3 \tag{5}$$

As each barrel plate is inclined over an angle β relative to the rotor, the projection of the piston circle on the barrel plate is an ellipsoid with the dimension of the short axis equal to $R_r \cdot \cos\beta$ and that of the long axis equal to R_r. Two adjacent cups are closest to each other when they are in the vicinity of the short axis. Their distance in this position can be approximated by assuming that they are both on a circle with radius $R_r \cdot \cos\beta$.

The minimum distance between two adjacent cups is $2 \cdot t + B$. Scaling t linearly with B, implies that this minimum distance also scales linearly with B. A factor λ can be introduced which represents the constant ratio between this minimum distance and the piston diameter B. With this factor and a given the piston diameter B, the minimum radius R_r which gives the tightest packed rotor with N pistons in each rotor half can be calculated to:

$$R_r = \frac{\lambda \cdot B}{2 \cdot \sin\frac{\phi_{pi}}{2} \cdot \cos\beta} = \frac{\lambda \cdot B}{2 \cdot \sin\frac{\pi}{N} \cdot \cos\beta} \tag{6}$$

From figure 3 the relationship between the R_r and the stroke S can be deduced:

$$S = 2 \cdot R_r \cdot \tan\beta = \frac{\lambda \cdot B}{\sin\frac{\pi}{N}} \cdot \frac{\tan\beta}{\cos\beta} \tag{7}$$

With equations 6 and 7, the geometric swept volume V_g of a pump based on this tightest packed rotor can be determined to be:

$$V_g = 2 \cdot N \cdot \frac{\pi}{4} \cdot B^2 \cdot S = N \cdot \lambda \cdot \frac{\pi}{2} \cdot B^3 \cdot \frac{\tan\beta}{\sin\frac{\pi}{N} \cdot \cos\beta} \tag{8}$$

In conventional axial piston units, the maximum speed of the unit decreases with its size. As a consequence, the maximum flow Q_{max} does not scale linearly with V_g. For the A2FM series by Bosch Rexroth for instance, a scaling law can be fitted to:

$$Q_{max} \sim V_g{}^{\frac{3}{4}} \tag{9}$$

For the floating cup unit, as not enough experimental data does yet exist, the same relationship between V_g and Q_{max} is postulated.

Equations 8 and 9 show that, for a given number of pistons and basic unit angle β, the maximum output flow scales with the piston diameter B according to:

$$Q_{max} \sim B^{\frac{9}{4}} \tag{10}$$

With equations 5 and 10 the ratio of this leakage flow to the maximum output flow can be shown to scale with:

$$\frac{Q_{C_l}}{Q_{max}} \sim B^{\frac{3}{4}} \tag{11}$$

Equation 8 shows that if the number of pistons N, the unit angle β and the factor λ are fixed, V_g scales with the third power of B only. Equation 11 can thus also be written as:

$$\frac{Q_{C_l}}{Q_{max}} \sim V_g{}^{\frac{1}{4}} \tag{12}$$

This seems to suggest that if t is scaled linearly with B, the ratio of these leakage losses to the maximum output flow increases with the size of the unit. However, the increase is far less than linear and it should also be considered that in the derivation above, only the expansion of the cups was included in the gap height h_0. Production tolerances will offset the expansion gap with an initial gap height. As this initial gap height can be kept fairly constant over the total size range, it will affect the volumetric efficiency of small units more than that of large units.

Considering this, the choice to maintain a linear relationship between t and B when scaling the FC principle, is a good one. This also implies that equations 6, 7 and 8 may be used to scale the main dimensions of a known rotary group to a new one with the desired displacement V_g. They have been used to scale the 50 cm^3 IHT prototype form the known dimensions of the 28 cm^3 pump prototype.

5 PORT PLATE DESIGN

The design of the interfaces of the port plate with the barrel and cover, is a matter of carefully balancing the hydrostatic forces in those interfaces, with the fundamental hydrostatic forces that originate from the pressurized oil in the floating cups. The goal is to minimize the resulting contact forces, as these determine the port plate control torque and a large part of the hydro mechanical efficiency of the unit.

5.1 Fundamental forces

The pressurized oil in the cups presses the assembly of barrel plate and cups to the port plate and presses both the assembly and the port plate to the cover (figure 4).

As the rotor turns and the cups change their kidney connection, the resultant of all cup forces

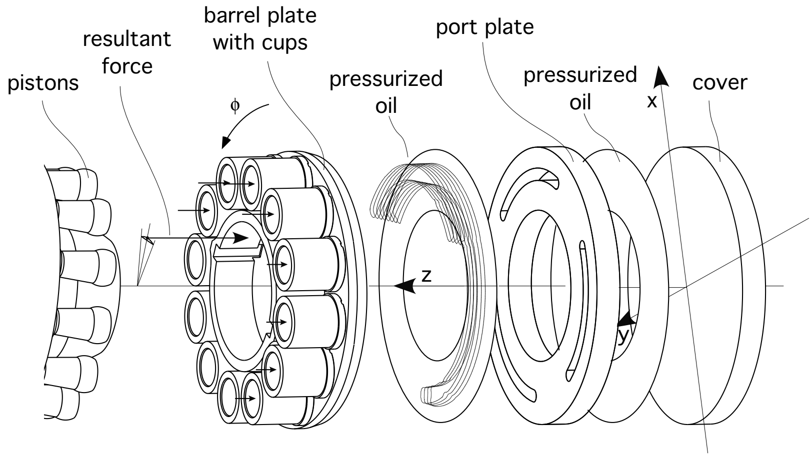

Figure 4: The interfaces between barrel, port plate and cover of the FC IHT

moves in the x-y plane. To determine which effect the commutation has on the magnitude of the resultant force, a distinction has to be made between 'even' and 'odd' machine types. Since this paper discusses the IHT, the definition of 'even' and 'odd' should be generalized.

If an even machine (or machine half in the FC design) is defined as a machine in which the number of pistons is divisible by the number of kidneys and an odd machine as a machine in which the number of pistons is not divisible by the number of kidneys, the concept of odd and even can still be applied to pumps and motors with two kidneys but also to the IHT, which has three kidneys per machine.

In an 'even' machine the magnitude of the resultant force is constant, only its position varies. In an odd machine both the magnitude and the position of the resultant piston force vary with the momentary position of the rotor (Figure 5). The cups and barrel plate assembly, the port plate and the cover form a chain through which the resultant force generated by the pressurized oil in the cups is transferred. The resultant force works in every interface between these links. In the interfaces between barrel plate and port plate and between port plate and cover, the resultant force is divided in a hydrostatic force in the oil film and a direct mechanical contact force. The aim of the port plate design is to minimize these contact forces by balancing the resultant cup force with the hydrostatic forces in the film. Large contact forces would cause friction in the interfaces which reduces the hydro mechanical efficiency and increases the port plate control torque.

5.2 The interface between barrel and port plate

Between the barrel plate and the port plate a pressurized oil field travels along with the rotating

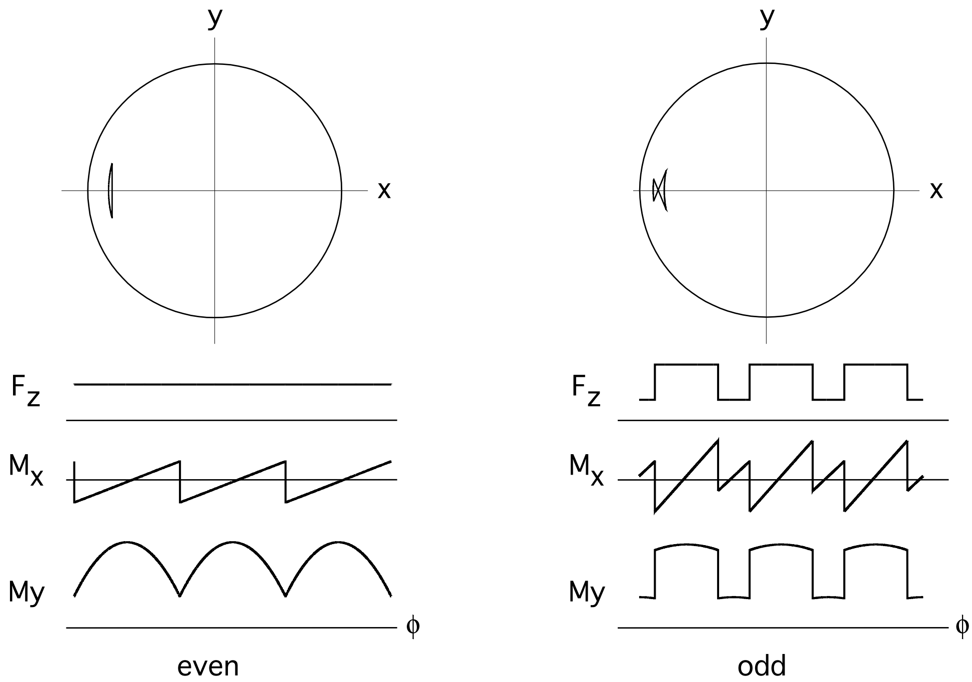

Figure 5: The resulting force on the barrel in 'even' and 'odd' machine types

barrel. This oil field mimics the moving resultant cup force (figure 4) almost perfectly. However, small differences in magnitude and position are created when a port in the barrel plate travels over a valve land between two kidneys. The pressure distribution can be calculated in good approximation, by interpolating linearly between the given pressures in the ports in the barrel, the kidneys in the port plate and the pressure in the IHT housing. In this section, only even machines are treated.

5.2.1 Valve land effects

To understand the valve land effects it is useful to divide the pressure field in the port plate barrel interface into segments, one for each piston. When all segments are in balance with their piston, the whole barrel plate is in balance. In figure 6 a balanced field is drawn for one of the pistons. Differences between this balanced field and the practical fields that occur at certain other piston positions, cause differences in magnitude and position of the resultant cup force and the hydrostatic force in this interface. When the rounded ends of the high-pressure port and kidney are tangent, there is a pressure dip in the field. In figure 6 this is the situation indicated by $-dF_1$ at $\phi = +0.1^o$. This is called the 'port shape effect'. The port shape effect causes a rise in the contact force F_z between port plate and barrel and influences the contact moments M_x and M_y.

Another pressure dip is created when the port land is completely covered by a low-pressure kidney. In figure 6 this is the situation indicated by $-dF_2$ at $\phi = 0.1^o$. A pressure rise occurs the port land is covered by the high-pressure kidney. (dF_2 at $\phi = -0.1^o$ in figure 6). This is called the 'port land effect'. Because the rise and dip of the port land effect are created simultaneously there is no effect on the magnitude of the contact force F_z but it does influence the contact moments M_x and M_y. The port land effect can be minimized by choosing a small port land length (pl). Concurrent ports (see section 6) can minimize the port shape effect.

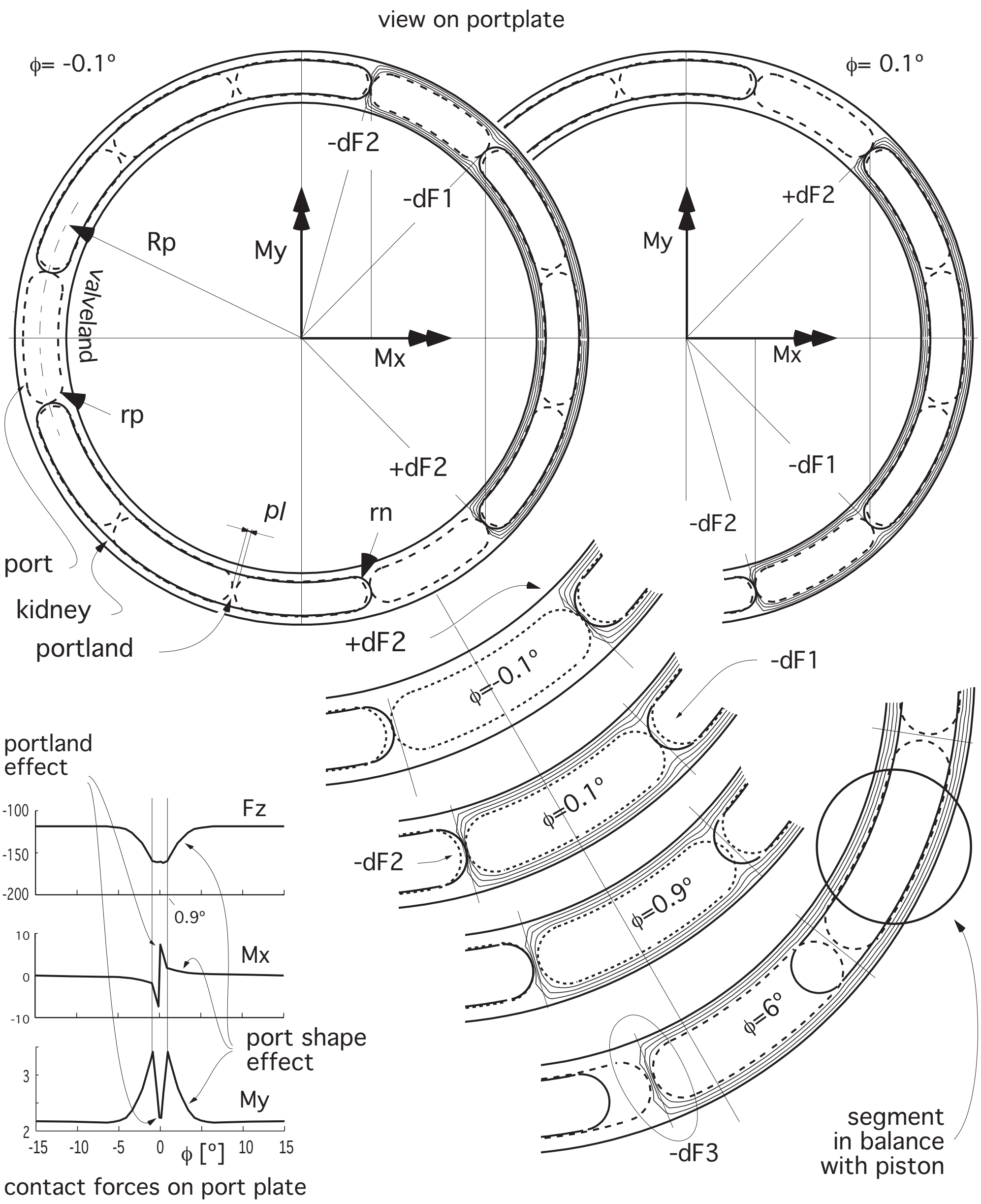

Figure 6: Valve land effects

5.2.2 Barrel equilibrium

As the barrel rotates, the hydrostatic force travels along almost exactly with the resultant cup force. The valve land effects cause small differences between the positions of both forces, which represent moments. The contact force travels over the interface area in response to these moments and counteracting them. By tuning the geometry of the hydrostatic interface between port plate and barrel, the contact force is set large enough to guarantee that it will stay within the outer radius of the interface when the maximum contact moment occurs (Figure 7). In this position and with the rotor in a standstill position, the contact force F_c causes a maximum friction torque

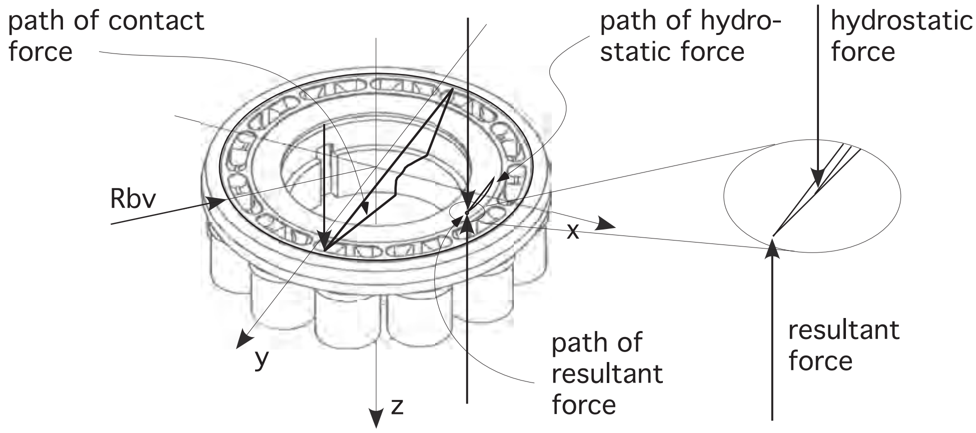

Figure 7: Barrel equilibrium

between barrel plate and port plate:

$$T_{friction} = \mu \cdot F_c \cdot Rbv \tag{13}$$

in which μ is the dry friction coefficient.
For a 50 cm^3 machine a maximum contact force of 160 N and a friction torque of 1.5 Nm per interface is calculated.

5.3 The interface between port plate and cover

In contrast to the barrel-port plate interface the pressurized oil field between the port plate and the cover is static and can only balance the moving resultant cup force at one rotational position of the rotor. As the rotor turns, large differences between the position of the resultant cup force and the position of the hydrostatic force occur. These large differences necessitate a large contact force to counteract them. Without extra measures, this large contact force may cause large port

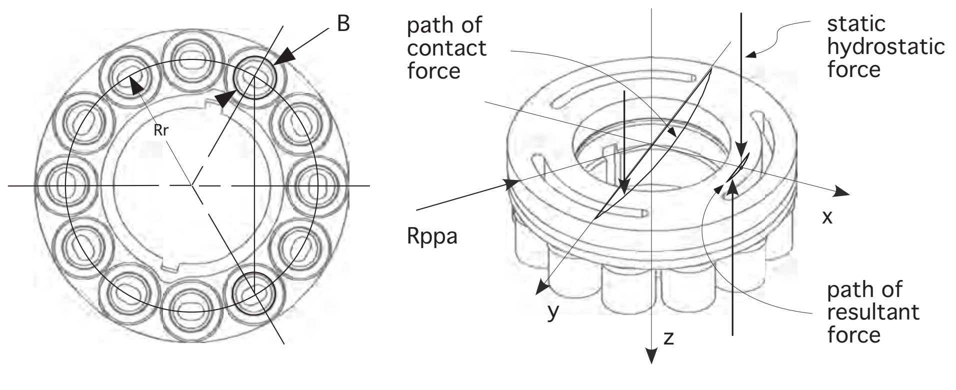

Figure 8: Port plate equilibrium

plate friction. This is unwanted, as in an IHT this port plate friction directly determines the required port plate control torques.

5.3.1 Port plate control torque

In an even transformer the amplitude of the fundamental moment variation caused by the movement of the resultant cup force is:

$$dM = 2 \cdot Rr \cdot \sin 60 \cdot \frac{\pi}{4} \cdot B^2 \cdot p \tag{14}$$

Barrel and port plate stability demands:

$$2 \cdot F_c \cdot Rppa \geq dM \tag{15}$$
$$F_c \geq \frac{dM}{2 \cdot Rppa} \tag{16}$$

This causes a maximum control friction torque between port plate and cover:

$$T_{friction} = \mu \cdot F_c \cdot Rppa = \mu \cdot \frac{dM}{2} = \mu \cdot Rr \cdot \sin 60 \cdot \frac{\pi}{4} \cdot B^2 \cdot p \tag{17}$$

This maximum friction in the port plate cover interface directly determines the maximum torque required to control the port plate. With equations 6 and 8 the scaling law for this part of the control torque can be determined to be:

$$T_{control} \sim \frac{V_g}{N \cdot \lambda . \beta^{\frac{1}{3}}} \tag{18}$$

For an odd transformer similar calculations can be made. In figure 9 the maximum control torque in an unmodified interface between port plate and cover is plotted ($\mu = 0.2, \beta = 9^o$). The left figure shows the torque for an even unit plotted as a function of the nominal displacement volume V_g. The right figure shows the torque in for the 50 cm^3 FC IHT design, as a function of the number of pistons $N$of one rotor half. The right plot shows that the control torque in odd

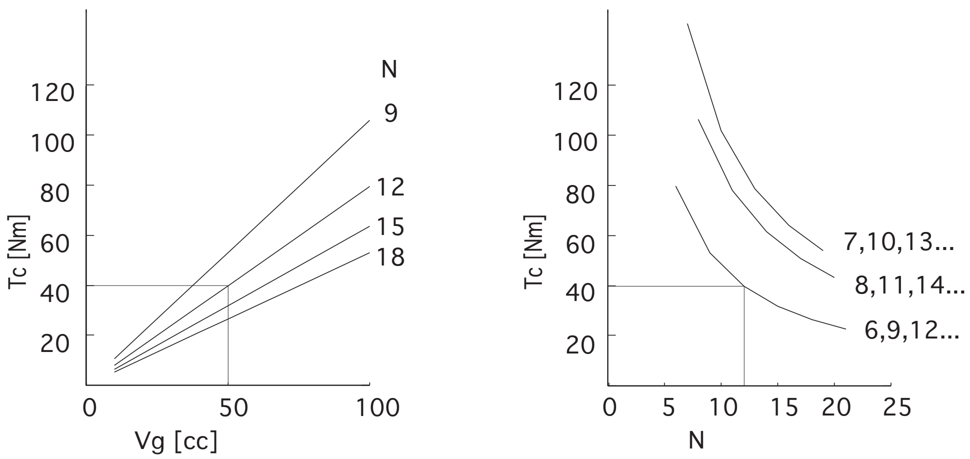

Figure 9: Maximum control torque in the interface between port plate and cover

machines doubles compared to even machines. This is the main argument for choosing an even number of pistons for an IHT. In the left plot lines for even machines with 9, 12,15 and 18 pistons are drawn. In a 2x12 piston 50 cm^3 transformer with an unmodified interface between port plate and cover, the control torque is dominated by the friction in this interface, which amounts up to 40 Nm per interface. For this reason, in the 50 cm^3 FC IHT design, a ball bearing is added between port plate and cover. With this bearing, the friction torque between port plate and cover can be decreased by a factor of 200. Then, the control torque is dominated by the friction in the interface between port plate and barrel, which was calculated before to be 1.5 Nm per interface.

5.3.2 Ball bearing design

Because of tolerances in the production of port plate, bearing and cover, adding a mere ball bearing would result in an rather large gap between port plate and cover, which would lead to excessive leakage at that interface. In order to prevent this, the port plate is divided in a back and front part. The motoring and pumping kidneys of both parts are connected via bushings with O-rings that press the back part to the cover. The oil in the make up kidney flows radially through the port plate to the machine housing. The varying moment on the front part is taken by the ball bearing. The interfaces between the front part and the bushings, between the bushings and the back part and between the back part and the cover are hydrostatically balanced with the constant part of the resultant cup force.

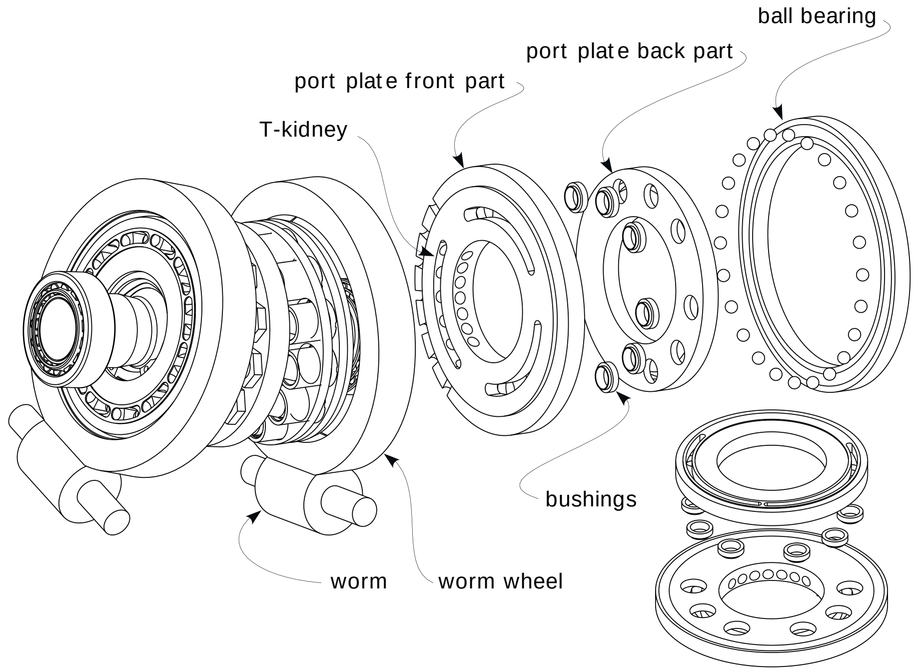

Figure 10: The split up port plate with bushings and ball bearing

6 VALVING LAND PHENOMENA

In a hydrostatic pump or motor, the pistons change their pressure connection at or near the dead centers of their movement. In contrast to that, in an IHT the pistons can change kidney while they are moving. During these kidney transitions in the IHT, the passages through which the pistons pump oil, decrease from their maximum area to zero and increase to the maximum again, while the flow from these pistons is not zero.

In IHT designs based on conventional axial piston units and without extra measures, this commutation at non zero flow can cause severe pressure spikes in the displacement chambers. These can lead to elevated noise levels, problems with the port plate balance and the risk of cavitation. In addition to this, because the flow is forced through small areas, the hydro-mechanical efficiency deteriorates.

These overlap phenomena have been described in [2], in which also the solution of 'concurrent' ports has been introduced. In this solution, the edges of the flow ports of the displacement chambers in the barrel, are designed to coincide with the edges of the kidneys. In this way, the flow area between chambers and kidneys, is opened as quickly as possible and the blocking of the flow is minized.

The concurrent ports brought an improvement but when further testing of several IHT designs showed that the gain was not enough, a different solution was conceived and patented. In this solution, small shuttle valves are arranged between the displacement volumes. When the flow passage from a volume to a kidney is being blocked and the pressure in that volume rises above the value of the adjacent volume, the shuttle valve opens and provides an unblocked flow path through that volume, to the next kidney. A detailed description of this solution and its effects can be found in [3]. In figure 11 three simulated traces of the pressure in a cylinder bore are

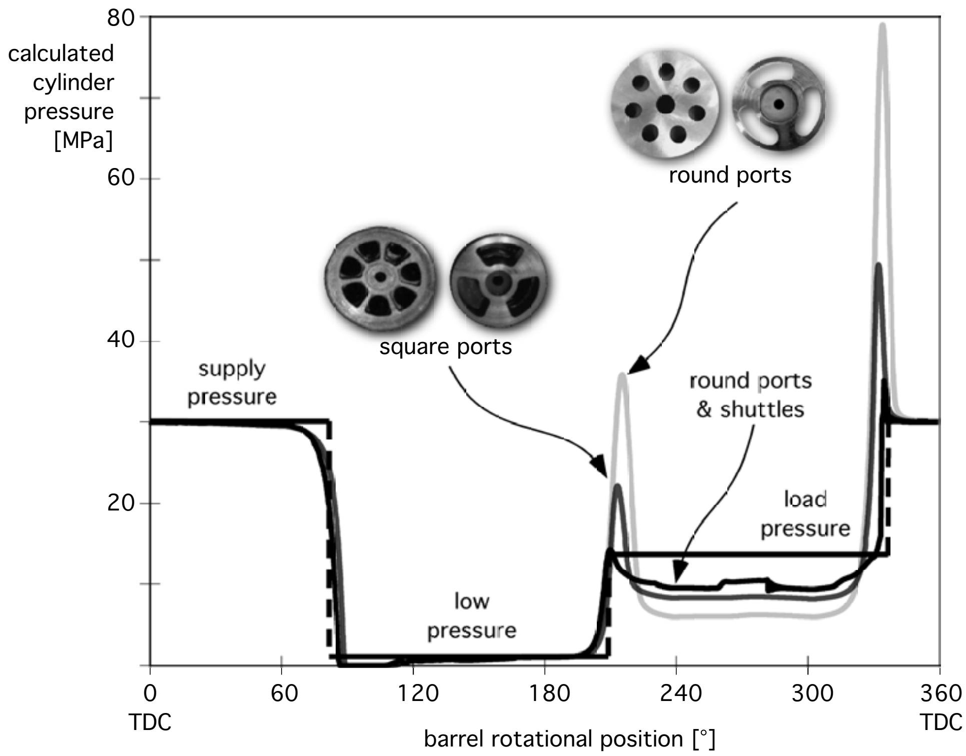

Figure 11: The influence of square ports and shuttles on the pressure spikes

presented: one for an IHT with normal round port edges, one for an IHT with square port edges and one for an IHT with round port edges and shuttles. In all three cases, the base unit is an A2FM10 with seven pistons, simulated at a speed of 7000 rpm. With the floating cup concept, IHT's with a far larger number of pistons can be build economically. As this reduces the flow per plunger considerably, it may be expected that overlap phenomena will be considerably less severe. Figure 12 shows that this is indeed the case. The simulated pV-diagrams for a 10 cm^3 FC IHT are compared to those of a 10 cm^3 A2FM based IHT. The comparison is given for three speeds (1000, 4000 and 8000 rpm) and for three different port plate angles δ (20, 40 and 60 degrees). In order to estimate the influence of the unit size on the magnitude of the pressure

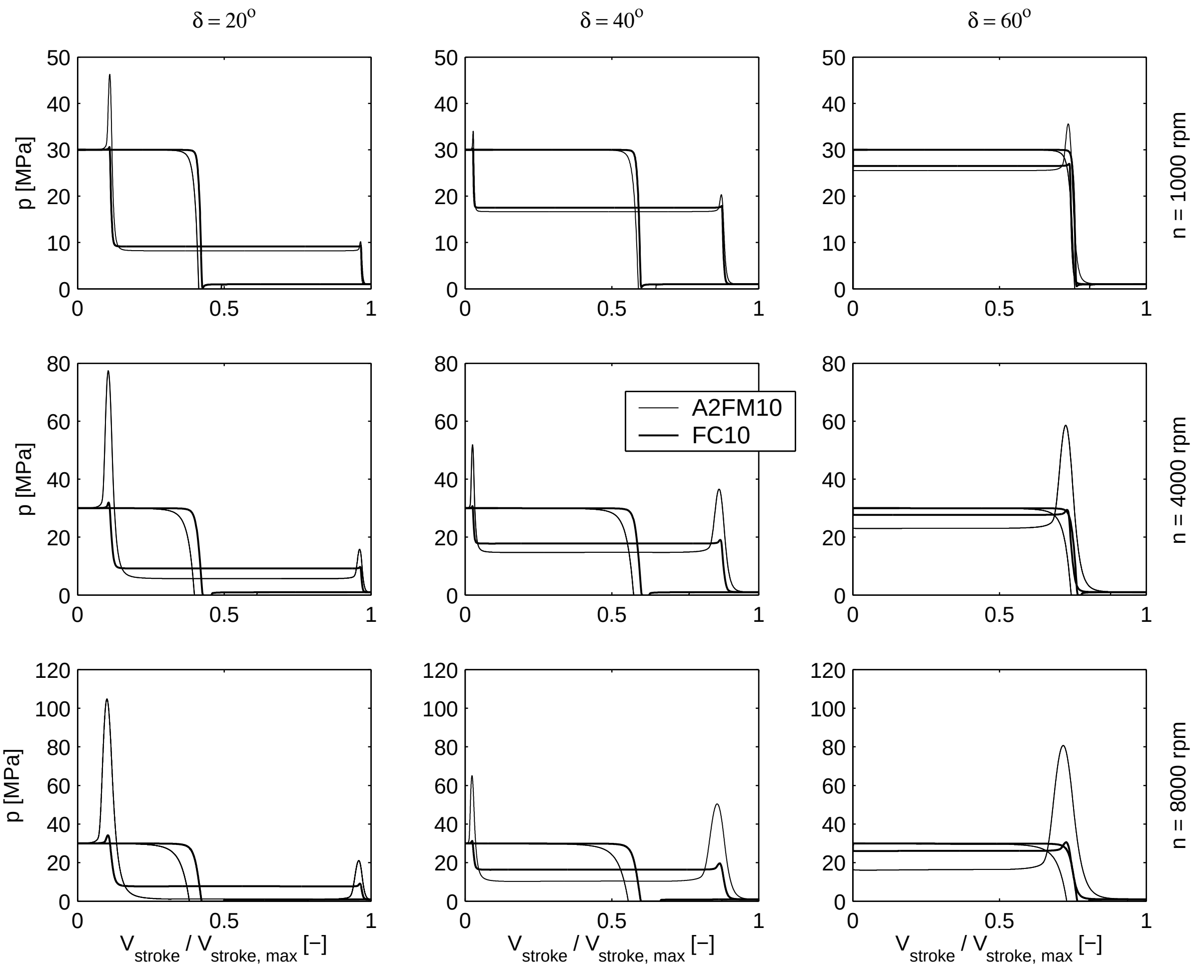

Figure 12: Comparing the FC IHT to an A2FM based IHT

spikes at port change over, the flow from each piston and the port opening area have to be expressed as functions of the geometrical parameters.

If the number of pistons N and the unit angle β have been chosen, the piston flow at a certain port change over – at a given port plate angle δ and unit speed n – is directly proportional to the maximum flow from the unit, Q_{max}. Thus, the scaling law for the piston flow to the unit size is identical to equation 9:

$$Q_p \sim V_g^{\frac{3}{4}} \tag{19}$$

The port opening area is determined by the geometry of the ports and the kidneys in the interface between port-plate and barrel. As has been shown in the previous section, in this interface, the contact force has to be minimized by balancing two large forces: the resultant cup force and the hydrostatic forces in the interface between barrel and port plate. The goal of the balancing is to reach a resulting contact force that is about two orders of magnitude smaller than the resultant cup force. As a good approximation, it can therefore be postulated that these two forces should be equal. Referring to the balanced kidney section in figure 6 and postulating that each kidney contains $\frac{N}{3}$ of these sections, this leads to the following equation for a kidney at a pressure p:

$$\frac{N}{3} \cdot \frac{\pi}{4} \cdot B^2 \cdot p = 2 \cdot \frac{\pi}{3} \cdot Rp \cdot 2 \cdot \left(rp + \frac{bs}{2} \right) \cdot p \tag{20}$$

In which Rp is the radius of the pitch circle of the kidneys in the port plate, rp is the end radius of the kidneys, which is half the kidney width, and bs is the width of the seal lands around the kidneys. For the scaling the radius Rp may be taken approximately equal to R_r, so equation 20 can be written as:

$$\left(rp + \frac{bs}{2} \right) = \frac{N \cdot B^2}{8 \cdot Rr} \tag{21}$$

If N and β are fixed and if the ratio between bs and rp is taken to be constant, equations 6 and 8 can be used to derive a scaling law between between rp and Vg. This yields:

$$rp \sim V_g^{\frac{1}{3}} \tag{22}$$

The flow area at port change over scales with rp^2, so a scaling law for the pressure increase resulting from the temporarily blocked port flow can be derived using the well known formula for the flow through a turbulent restriction in combination with equations 19 and 22:

$$\Delta p \sim \frac{Qp^2}{Ap^2} \sim \frac{Qp^2}{rp^4} \sim V_g^{\frac{1}{6}} \tag{23}$$

With this scaling law, the pressure increase in the cups at port change over, which were calculated for a 10 cm^3 FC IHT (see figure 12), can be predicted to rise with 30% when the unit is scaled to 50 cm^3, using the equations derived in section 4. This is a marginal effect, compared to the pressure peaks in the A2FM10 based IHT.

7 CONCLUSION

In this paper, equations have been derived for the scaling of the geometry of the FC principle to larger displacements. It has been shown how these scaling laws affect two important design aspects of any IHT: the port plate balance and the valving land phenomena.

The port plate balance strongly influences the port plate control torque. In the 50 cm^3 FC IHT prototype, the port plate control torque will be minimised by introducing a port plate bearing. A split port plate design ensures that production tolerances of the port plate can be kept at normal levels.

The valving land phenomena get slightly worse, when the FC IHT design is scaled to larger displacements. Compared to a conventional IHT design, however, the FC IHT design already improves the valving land phenomena that much, that the scaling effects may be considered to be marginal.

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