THE DESIGN OF THE FCT80 HYDRAULIC TRANSFORMER

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ABSTRACT

Hydraulic transformers and CPR-systems are a promising alternative for current valve-controlled systems. Instead of throttling, the control is realised in a non-dissipative manner. Furthermore, energy can be recuperated and delivered to other loads or stored in hydropneumatics accumulators. Hydraulic transformers are the cornerstone for these systems. In this paper, a new design of a hydraulic transformer, the FCT80, is described. The design is based on the concept of the Innas Hydraulic Transformer (IHT) combined with the floating cup displacement principle. In the past twenty years, the floating cup principle has first been developed and introduced in hydrostatic pumps and motors. This pump and motor knowhow is now applied in the design of the new 80cc Floating Cup Transformer, the FCT80. The key elements in the new design are the servo control, the new port plates, and the oil flow through the main shaft, which has been made hollow.

KEYWORDS: Hydraulic transformer, floating cup principle, CPR-system

1. INTRODUCTION

For many decades, the market of non-road mobile machines has been steady and calm. But, *the times, they are a-changin'*, also for this market. Climate laws, especially in Europe, force the industry to strongly increase the efficiency of their machines, and even come up with zero-emission solutions. The times of relatively cheap fossil fuels are over, and the alternatives (batteries and sustainable fuels) are much more expensive. Because of this, the efficiency has become a number one priority. Current hydraulic systems, with a system efficiency often below 50%, are no longer acceptable.

Common Pressure Rail (CPR) systems, with accumulators and hydraulic transformers, offer a solution which is in many ways more efficient than current valve control systems:

- Throttle control is replaced by transformer control;
- Energy can be recuperated;
- The hydraulic circuit is simplified to a high-pressure and a low-pressure rail. These lines can have a larger diameter and hence reduce the flow losses;
- The conventional variable displacement pumps can be replaced by a smaller and more efficient fixed displacement pump;
- The hydraulic accumulators can also be used for power management, which allows the prime mover and the pump to be much more efficient.

Many of the advantages of current hydraulic systems are retained. Cooling (or heating), filtering and filling are still centralised, as well as filling of the system and deaeration of the fluid. The accumulators will add extra weight, volume and costs to the system, but the coolers will be much smaller due to the strongly reduced losses [1].

An important advantage of transformer control is the low moment of inertia of the rotating parts combined with secondary control [2]. This is important for mobile machines, for which high productivity is often crucial. The accumulators are not only necessary for power management and energy recuperation, but can often also be used for important features like ride control [3, 4]. This way, terrain induced bumps can be absorbed while driving, without the need of a separate ride control accumulator and system.

Many previous studies have acknowledged the advantages of transformer-based systems, but also concluded that there are no hydraulic transformers commercially available [5-9]. In 1996, INNAS proposed a fundamentally new design principle (the Innas Hydraulic Transformer or IHT) [10, 11]. The first designs of the IHT were converted bent axis motors, and although the principle of the IHT could be proven, it also became apparent that a new hydrostatic displacement principle was needed. In 2002, INNAS presented the new floating cup principle, specifically for hydraulic transformers [12]. Since then, several transformer designs have been designed, build, and tested [13-15], but none of these designs were introduced into the market. In parallel, the floating cup principle was further developed for pumps and motors [16-28]. In 2019, the floating cup principle was introduced by Bucher Hydraulics as the AX-series of pumps and motors [29].

Soon after, INNAS again started the development of a new hydraulic transformer, thereby utilizing all the knowledge gained during the development of the various floating cup pumps and motors. This paper is the first paper to describe the design of the new Innas Hydraulic Transformer, with the working title 'FCT80'. The new transformer is designed to control the pressure and flow for a differential cylinder with a maximum flow capacity of about 180 ℓ/min. The maximum pressure is 350 bar.

2. THE PRINCIPLE OF THE INNAS HYDRAULIC TRANSFORMER

Around 1980, the first designs of hydraulic transformers emerged in the literature, and even, for a short while, in the market. These designs were based on combinations of two pump/motors, of which at least one is a variable displacement design.

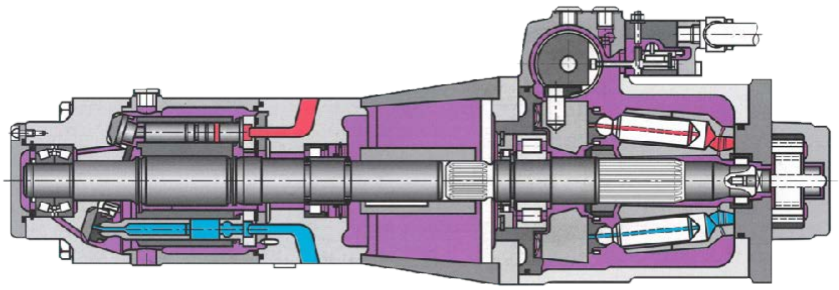


Figure 1. Hydraulic transformer build as a combination of a fixed and a variable displacement pump/motor [30]

Transformers are connected to a high-pressure rail, a low-pressure rail, and a connection to the load. There are several pump/motor-combinations possible [31, 32], but only a few configurations are meaningful [32].

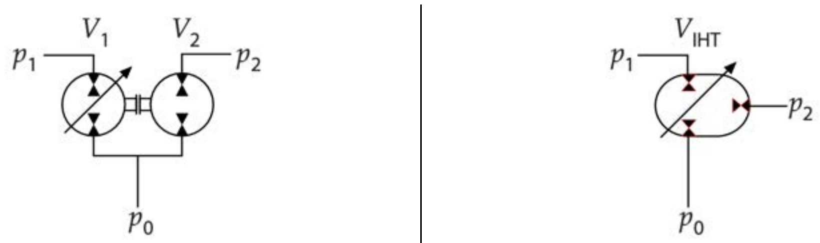
In many applications, like in an excavator or a loader, the hydraulic differential cylinders operate in four quadrants [1, 33, 34]. Directional valves can be used to determine to which side of the differential cylinder the load port of the transformer is connected:

- to the piston side,
- to the ring side,
- or to both sides.

Cylinders continuously work in a stop-and-go operation, which means that the transformer will also need to start from zero, often at high breakaway loads. In addition, a high productivity is one of the most important demands. The flow, pressure and power levels of the cylinders are therefore extremely dynamic and very demanding. To reduce the size of the transformers, the maximum speed needs to be high.

Another important demand for hydraulic transformers is the maximum pressure amplification from the pressure level of the high-pressure rail to the demanded pressure level of the load. Considering that the accumulators will be used for power management and energy recuperation, the pressure level in the high-pressure accumulator will change constantly. The situation can thus occur that, at a certain moment, the pressure in the accumulator is only 175 bar, whereas the load demands a pressure of 350 bar. This would result in a required pressure amplification of a factor two when supplying oil from the high-pressure rail at 175 bar to the load demanding a pressure of 350 bar.

Hydraulic transformers can amplify pressures, but the maximum amplification ratio is set by the configuration. In Fig. 2, two different designs of hydraulic transformers are compared: a combination of a variable and a fixed displacement machine (CHT), and the Innas Hydraulic Transformer (IHT).



$$\frac{p_2}{p_1} = \xi_1 \frac{V_1}{V_2}, \quad 0 \leq \xi_1 \leq 1 \tag{1}$$

$$\frac{p_2}{p_1} = \frac{\sin\left(\frac{\alpha}{2}\right)}{\sin\left(\frac{\beta}{2}\right)} \frac{\sin(\delta)}{\sin\left(\delta + \frac{\gamma}{2}\right)} \tag{2}$$

a. Combination of two pump/motors (CHT)

b. Innas Hydraulic Transformer (IHT)

Figure 2. Transformation ratio p_2/p_1 of two transformer designs, assuming no losses, and assuming the pressure in the low-pressure rail $p_0 = 0$. The line with pressure p_1 is connected to the high-pressure rail. The line with pressure p_2 is connected to the load. The parameter ξ_1 sets the displacement of the variable displacement machine with a maximum displacement V_1. The parameters of the IHT in Eq(2) are explained in Fig. 3.

In the CHT, the displacement of the first pump/motor can be varied between 0 and maximum displacement by changing the parameter ξ_1 between 0 and 1 [35]. The maximum transformation ratio is set by the ratio of the geometrical displacements V_1/V_2 of both machines. If a maximum transformation ratio $p_2/p_1 = 2$ is demanded, then V_1 needs have twice the displacement of V_2:

$$V_1 = 2\, V_2 \tag{3}$$

The IHT does not need two machines. Instead, a single machine is used, which in theory could be any positive displacement principle which uses a port plate or similar kind of distributor. Assuming an axial piston principle, the IHT features a port plate with three ports. In addition, the port plate can be rotated around its own axis. The parameters in the equation of Fig. 2b are defined in Fig. 3. For the IHT, the transformation ratio is no longer dependent on the total geometrical displacement, but only on the length of the individual ports and the rotational position of the port plate [36].

The flow output (in ℓ/min) of the CHT is determined by the product of the rotational speed (in rpm) and the displacement V_2 (in [cc]):

$$Q_{2,CHT} = n \frac{V_2}{1000} \tag{4}$$

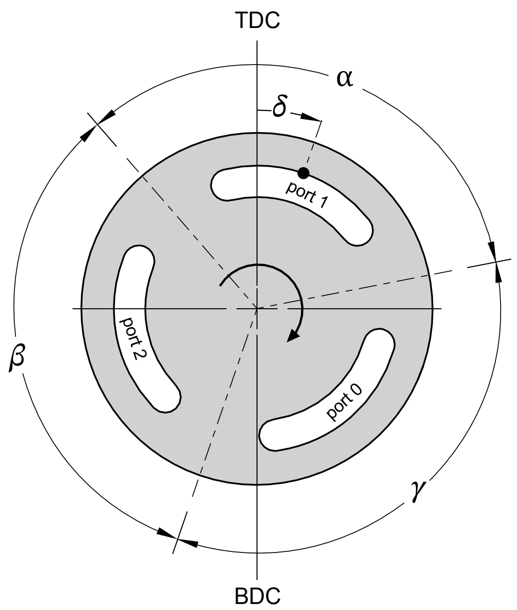


Figure 3. Example of a port plate of the IHT, showing the definition of the angles. Port 1 is connected to the high-pressure rail, port 0 to the low-pressure rail and port 2 to the load. The control angle δ is defined as the angle between the top dead centre (TDC) and the middle of port 1.

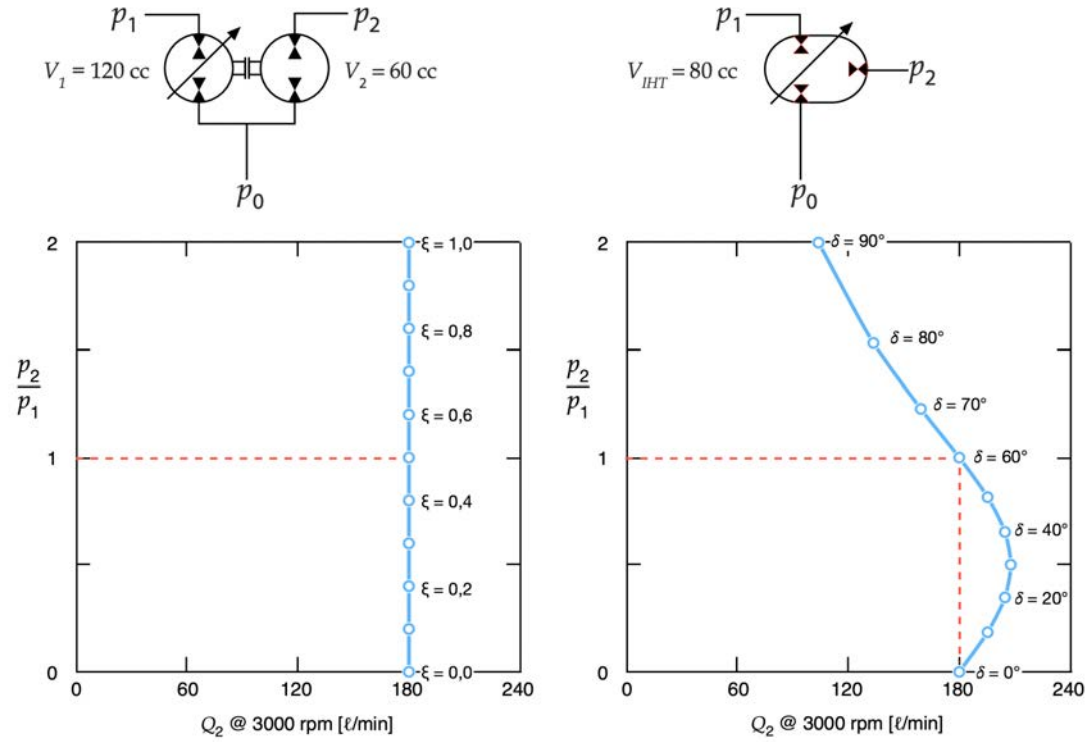
The flow output of the IHT is also dependent on the sizes of port 1, 2, and 3, and on the control angle δ:

$$Q_2 = \frac{n\, V_{IHT}}{1000} \frac{\cos\left(\delta - \frac{\gamma}{2}\right) - \cos\left(\delta - \frac{\alpha}{2} - \beta\right)}{2} \tag{5}$$

With these equations it is possible to calculate how large the geometrical displacement of the CHT and the IHT need to be, assuming a certain flow demand at a certain rotational speed. The transformation ratio p_2/p_1 is set to be $p_2/p_1 = 1$ in this reference point, and it is assumed that there are no losses. If, for instance, a flow of 180 ℓ/min needs to be achieved at a rotational speed of n = 3000 rpm, then it can be calculated that:

- V_1 = 120 cc
- V_2 = 60 cc
- V_{IHT} = 80 cc

In this calculation, the IHT is assumed to have three equally sized ports with an effective arc length of 120°. The calculation shows that the CHT needs a relatively large variable displacement pump/motor with a maximum displacement of 120 cc, combined with a 60 cc fixed displacement pump/motor, which results in a total displacement of 180 cc. In comparison, in this example, the IHT only needs a design with total geometrical displacement of 80 cc, less than halve the total displacement of the CHT. Fig. 4 shows the calculated transformation ratio versus the flow Q_2 at various values of ξ_1 for the CHT and δ for the IHT. Although the IHT has the same flow output as the CHT at $p_2/p_1 = 1$ and n = 3000 rpm, the flow output of the IHT is less at higher transformation ratios ($p_2/p_1 > 1$), assuming the rotational speed to remain constant. But, in most applications, this meets the requirements of a lower operating flow output at higher loads, and vice versa. For $p_2/p_1 \leq 1$ the IHT delivers a higher flow than the CHT.



a. Combination of two pump/motors (CHT)

b. Innas Hydraulic Transformer (IHT)

Figure 4. Calculated geometrical displacements of the CHT and the IHT, assuming a demanded flow of 180 ℓ/min at $p_2/p_1 = 1$ *and* n *= 3000 rpm*

The smaller total geometrical displacement of the IHT is not only favourable with respect to the weight, size, and costs of the transformer, but also results in a lower mass moment of inertia, which is needed for achieving a fast and dynamic operation of the transformer. Furthermore, the losses of the CHT are expected to be higher at the same operating conditions, simply because of the larger component sizes and bearing interfaces. One of the reasons why CHTs were not successful was the rather disappointing overall efficiency [35, 37], having an overall efficiency which was often lower than of simple throttle control by means of valves. For energy recuperation, the overall efficiency is even more important since the transformer will be used twice, during energy delivery and while recuperating the energy to be stored in the accumulator or used for another load. Nevertheless, also these CHTs could offer advantages in some market niches, especially in terms of peak-shaving.

3. THE FLOATING CUP PRINCIPLE

Although the floating cup principle offers considerable advantages for being applied in hydrostatic pumps and motors, the principle was originally designed for hydraulic transformers, more specifically for the IHT [12]. One of the first advantages is the high number of pistons of the floating cup principle. The torque of the IHT-axle is not constant due to the commutation outside the top and bottom dead centres (TDC and BDC) [38-44]. The resulting torque and flow ripple is strongly reduced by increasing the number of pistons. The floating cup principle is by design a multi-piston principle [20] having three to four times as much pistons as conventional axial piston pumps and motors.

A second advantage is the strong reduction of side forces in the contact between the pistons and the cups. The friction losses are further reduced by adopting a new design of the sealing lands in the contact between the barrels and the port plate [23]. The volumetric losses have been reduced by creating a cavity in the piston crown [22]. When being pressurised, the piston crown follows the radial expansion of the cups, thereby closing the gap between the two components. By means of reducing the wall thickness, the mass of the cups has been reduced. This reduces the tipping torque on the barrel which is induced by the centrifugal forces of the cups [28]. This is beneficial for increasing the maximum rotational speed, also when the floating cup principle is applied in a hydraulic transformer. Due to the mirrored design, the axial load on the shaft bearing is relatively small and the friction losses of the roller bearings are small, especially compared to bent axis pumps and motors.

The development of the floating cup principle has resulted in a high overall efficiency, higher than of other types of pumps and motors [45].

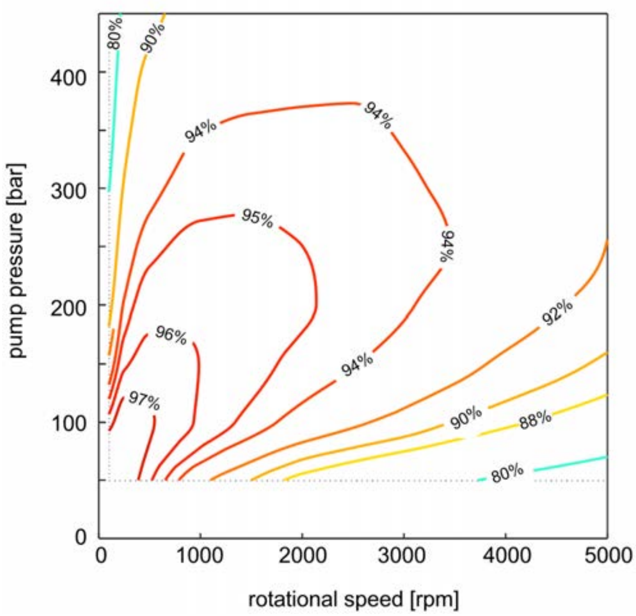


Figure 5. Measured overall efficiency of the FCM24 [45]

Aside from a high maximum rotational speed, the floating cup principle also offers the opportunity to be operated from standstill while having a near 100% torque efficiency and with lower volumetric losses than comparably sized pumps and motors. This is of great importance for application in hydraulic transformers, since one port is always connected to the high-pressure line and the transformer (like the hydraulic cylinder it is connected to) frequently operates in a stop-and-go operation, also when the load pressure is high.

4. DESIGN OF THE FCT80

The FCT80 is the latest design of INNAS of its hydraulic transformer, combining all the knowledge and experience of the previous designs, and the detailed knowledge of the various floating cup designs. The new transformer is designed to operate between -4000 and +4000 rpm, and to a maximum pressure level of 350 bar. Some of the key design parameters are summarized in Table 1.

Two hydraulic transformers will be used in a Common Pressure Rail (CPR) system (see Fig. 6), one controlling a single cylinder, and the other a pair of parallel cylinders. The cylinders are differential cylinders, and valves will be integrated into the transformer (Fig. 7) to determine which side of the cylinder will be connected to the load port of the transformer. In addition, two check valves and a pressure relief valve are added.

Table 1. Design parameters

geometrical displacement V_{IHT}	80 cc
number of pistons	30 (2 x 15)
swash angle	8°
effective port sizes	α = 168° β = 72° γ =120°
control angle	0° ≤ δ ≤ 70°
rotational speed	-4000 ≤ n ≤ 4000 rpm
$p_{0,max}$	15 bar
$p_{1,max}$	350 bar
$p_{2,max}$	350 bar
maximum power	180 kW
maximum load flow Q_2	187 l/min

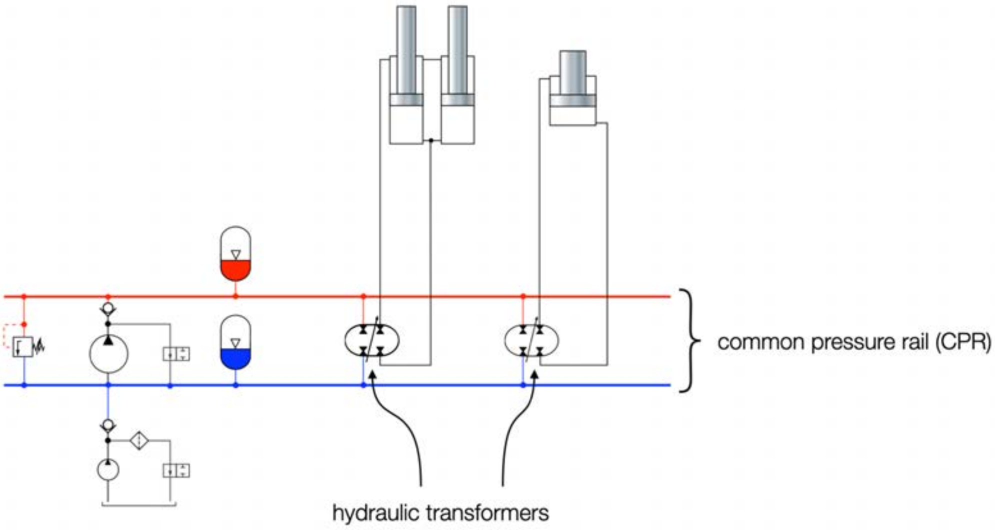


Figure 6. Example of a hydraulic CPR system with transformers and accumulators

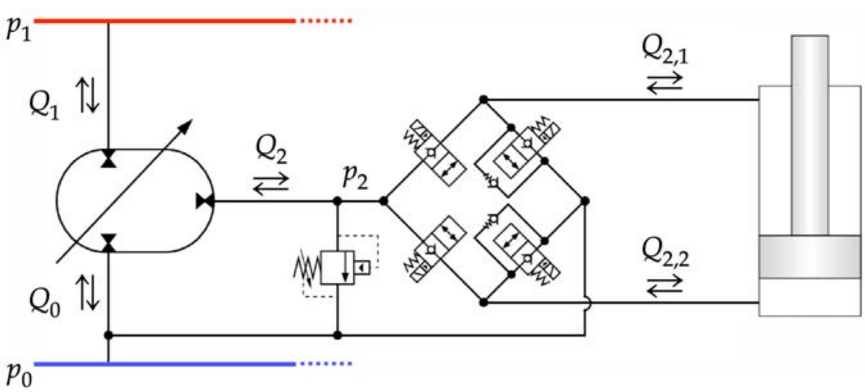


Figure 7. Hydraulic schematic of the transformer with the integrated valves, being connected to a differential cylinder

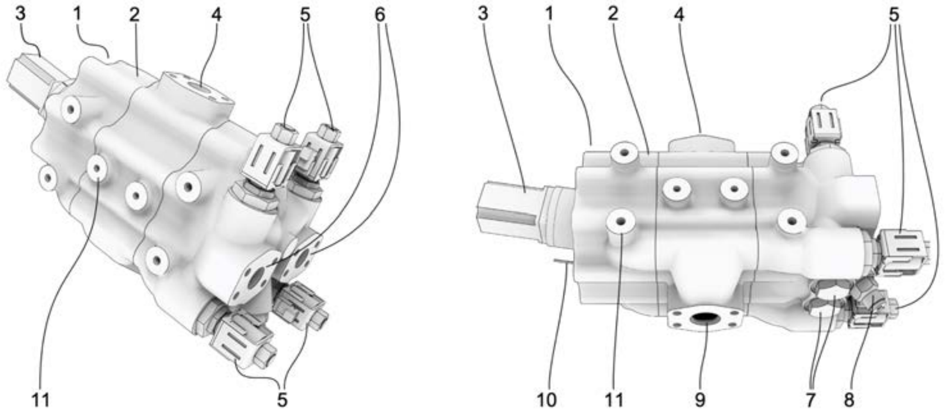


Figure 8. Drawing of the FCT80

1. the hydraulic transformer FCT80
2. case
3. closed loop stepper motor
4. connection to the high-pressure rail
5. directional valves
6. connections to the hydraulic cylinder
7. suction check valves
8. pressure relief valve
9. connection to the low-pressure rail
10. sensor for the port plate position
11. points for mounting the transformer

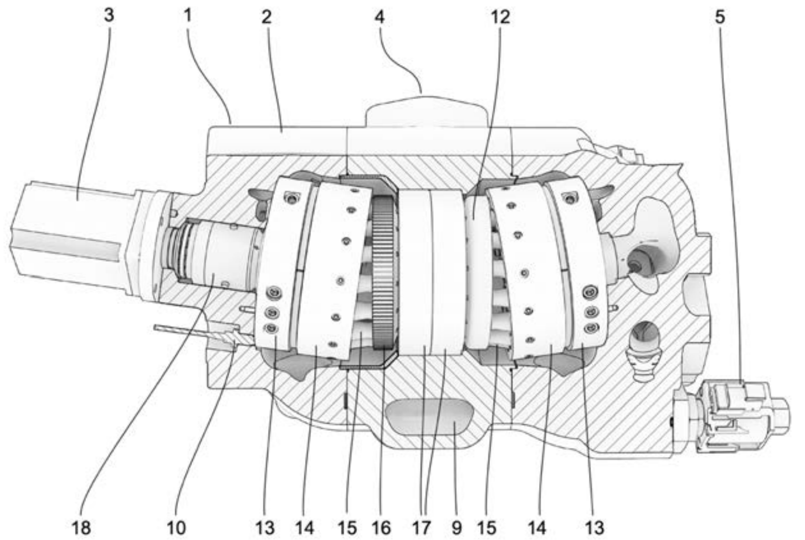


Figure 9. Cross section of the FCT 80

1. the hydraulic transformer FCT80
2. case
3. closed loop stepper motor
4. connection to the high-pressure rail
5. directional control valve
9. connection to the low-pressure rail
10. sensor for the port plate position
12. rotor disc with pistons
13. port plate
14. barrel and cups
15. pistons
16. toothed rotor for measuring the rotational speed
17. roller bearings of the main shaft
18. hydraulic servomotor

Figure 8 shows two views of the new transformer. Since the transformer does not have a drive shaft, the mounts of the transformer do not have to absorb the reaction torque, and the mounting can be much softer than for pumps or motors. The transformer has an electric servomotor which is combined with a hydraulic servo-actuator to control the rotational position of both port plates simultaneously, which will be described in section 4.3. The demand is to be able to rotate the port plates 60° within 10 milliseconds.

The inner parts of the transformer are shown in Fig. 9. The main shaft is supported in the middle by two angular contact roller bearings. On each side of the bearing pair, there is a flange on which the pistons are mounted, 15 pistons on each flange. One of the flanges is toothed for measuring the rotational speed of the transformer.

4.1. Port plate design

In the FCT80, the port plates can rotate over an angle of 70°. In a conventional axial piston pump or motor, the port plate position is fixed, and the hydrostatic balance is such that the plates will be pushed harder to the case when the pressure is increased. Pins and slots are used to avoid a rotation of the port plate because of the friction torque between the cylinder block and the port plate.

However, in the FCT80, the port plates need to be 'floating' on the case, and the friction between the port plates and the housing needs to be reduced to a minimum. For this reason, the ports on the front side of the port plate are largely copied to the back side. Figure 10 shows a drawing of one of the port plates, looking at the front and the rear. For experimental reasons, each port plate is a combination of a top and a base plate. By copying the ports, the pressure forces at the front of the port plate are nearly identical to the hydrostatic forces at the back. However, the forces are not 100% identical since a barrel rotates on top of the port plate, whereas the pressure fields at the back are stationary. Without any design changes, this would result in rather strong contact forces between the port plate and the case [46], thereby creating a high torque load to rotate the two port plates. To reduce these contact forces, a series of small boreholes (marked 24a in Fig. 10) are made at the front side of each port plate. These holes connect to small pockets at the back of the base port plate (marked 24b in Fig. 10).

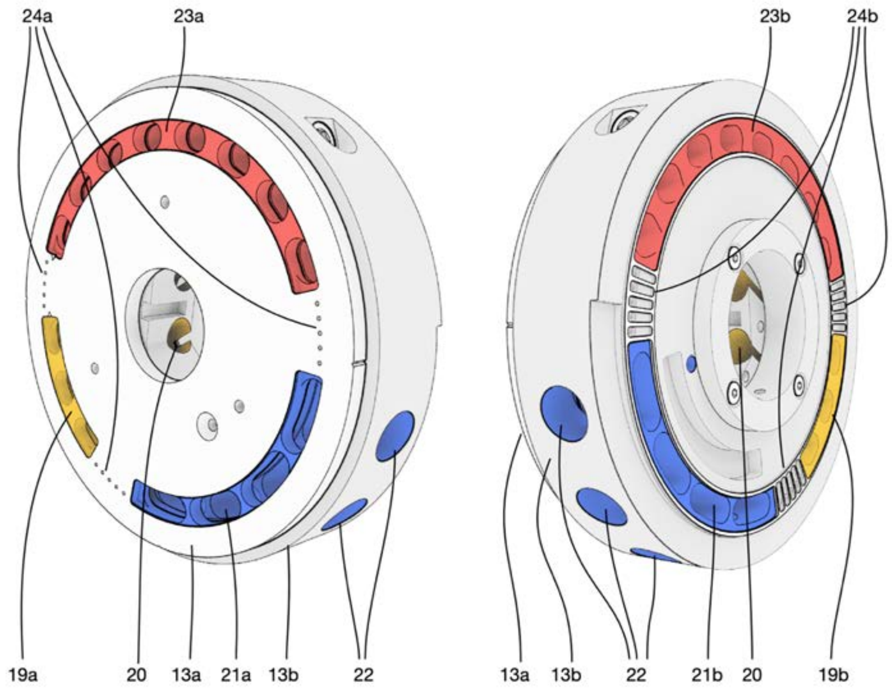


Figure 10. Front (left) and backside (right) of one of the two port plates of the FCT80

13a	top port plate	21b	rear low-pressure port
13b	base port plate	22	radial connection low-pressure port
19a	front load port	23a	front high-pressure port
19b	rear load port	23b	rear high-pressure port
20	centre connection load port	24a	front balancing boreholes
21a	front low-pressure port	24b	balancing pockets

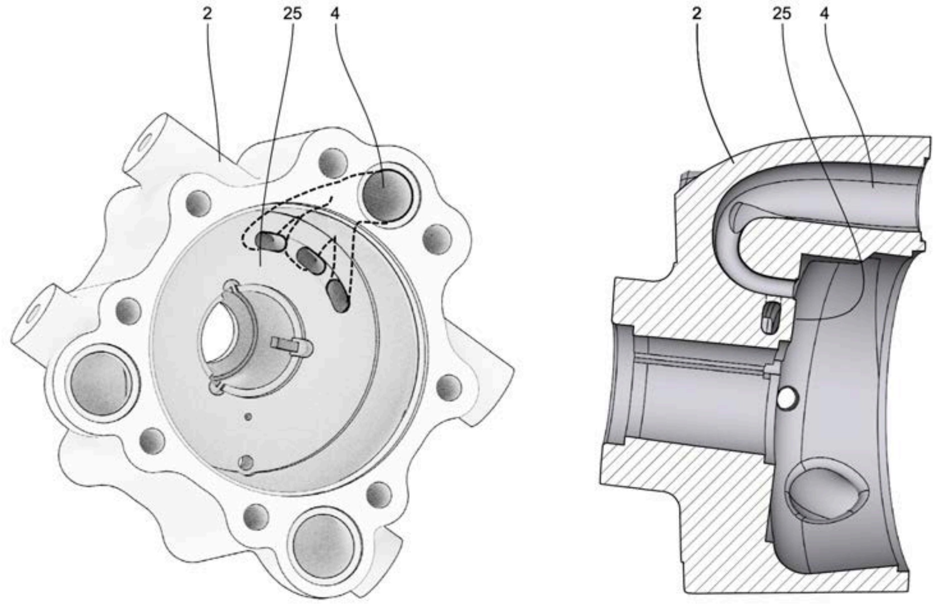


Figure 11. Part of the housing of the FCT80 showing the connection to the high-pressure port

2 case
4 connection to the high-pressure port
25 support plane for the port plate

Only the high-pressure port has a connecting port in the contact area between the port plate and the case (Fig. 11). The other two ports don't have such a connection. Instead, the port plate has radial passages (marked 22 in Fig. 10) that connect to the low-pressure port of the case. Consequently, the complete inside of the housing is used as a supply line for the low-pressure oil flow.

The load ports (marked 19a and 19b in Fig. 12) also don't have a connecting port in the housing. Instead, radial connections are made, like with the low-pressure port. The difference is however that the oil is now directed to the centre of the port plates instead of to the outer circumference.

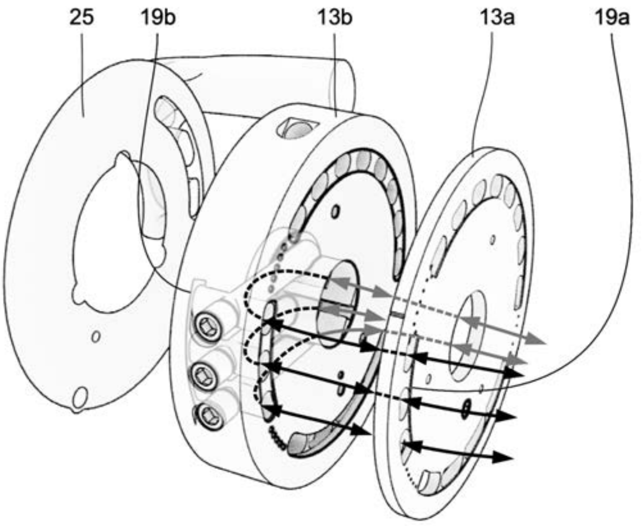


Figure 12. Oil flow with a connection to the load. The flow can go in both directions. During the recuperation, the flow direction is reversed

13a top port plate
13b base port plate
19a front load port
19b rear load port
25 support plane for the port plate

The new design allows a rotation of the port plates over an angle of 70° without any moving and stationary ports overlapping (Fig. 13). The only port that has a connection to the housing via a stationary port in the support plane is port 1, which is connected to the high-pressure supply line. The other ports are running blind. One of the reasons for making port 1 rather large is that it allows a relatively large stationary port in the housing.

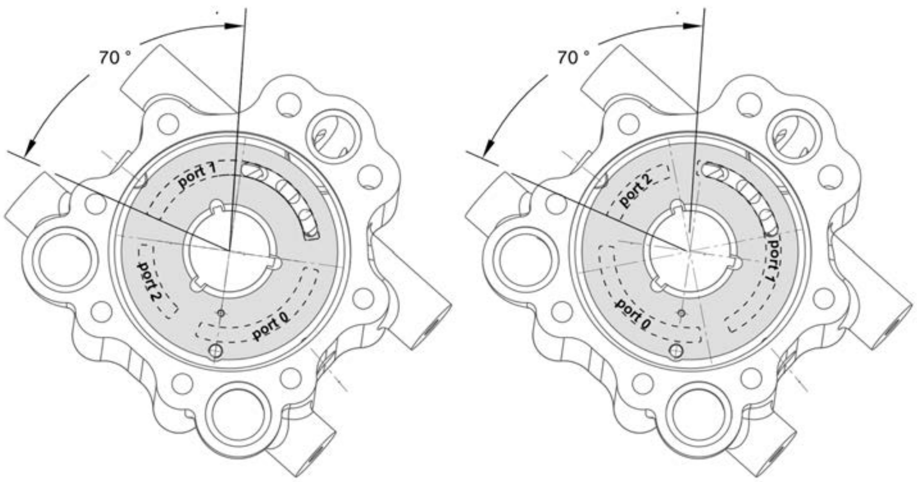


Figure 13. Part of the housing showing the port plate plane and two extreme positions of the port plate

4.2. Hollow main shaft

The novelty in the new FCT80-design is that the main shaft is made hollow (Fig.14 and 15) and is used as a connection for the load-flow between the two sides of the transformer. These centre ports are also balanced from the front to the back of the port plate. Due to the tilted position of the port plates and the barrels, a swivelling connection is made consisting of a kind of piston and cup (nr. 26 in Fig. 15).

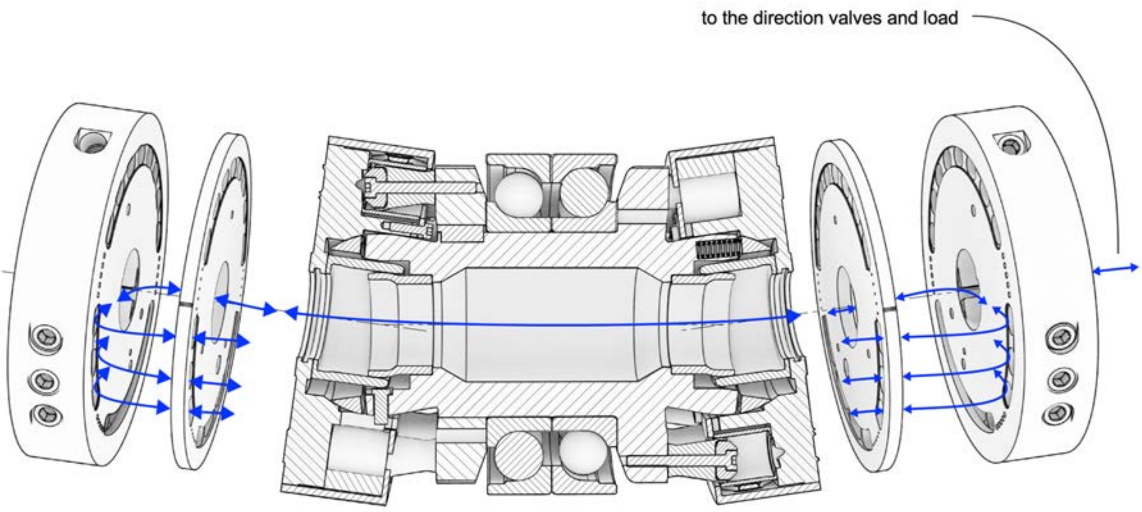


Figure 14. Oil flow with a connection to the load. The main components (the rotating group, the top port plates, and the base port plates) are shown separated, as an exploded view. The blue arrows show the flow path, which can go in both directions.

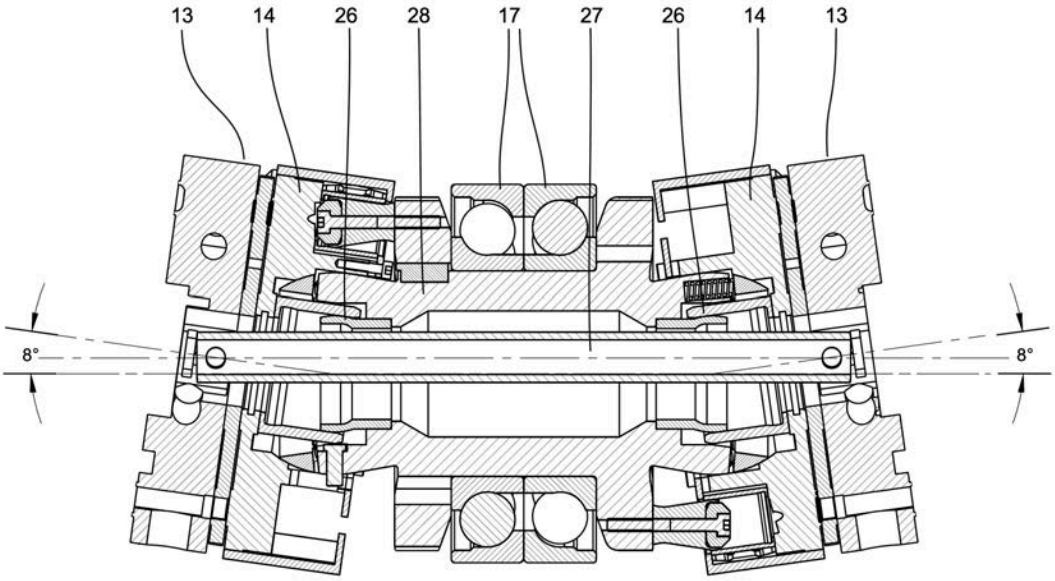


Figure 15. Cross section of the rotating parts and the port plates

13 port plates
14 barrels
17 main bearings
26 swivelling connections and sealing of the load pressure flow
27 shaft connecting the port plates
28 main shaft

The hollow shaft is also used to connect the two port plates mechanically by a mechanical joint connection (see Fig. 15). The construction is possible due to the relatively large space in the centre of the floating cup design, and due to the centre bearing location, which is possible because the transformer does not need a shaft going in or out of the case.

4.3. Servocontrol of the port plate position

The port plates are rotated by means of a hydraulic servomotor (nr. 18 in Fig. 9). Firstly, the motor needs to be strong enough to resist the friction torque of the barrel running on top of the swash plate. Next, there will always be some remaining friction between the port plate and the housing, which needs to be overcome by the servomotor. The third torque load is the moment of inertia. The design demand is to rotate the swash plate over an angle of 60° within 10 milliseconds. In the FCT80, the port plates are mechanically connected. The actuator for the port plate control therefore needs to overcome the torque load from both port plates.

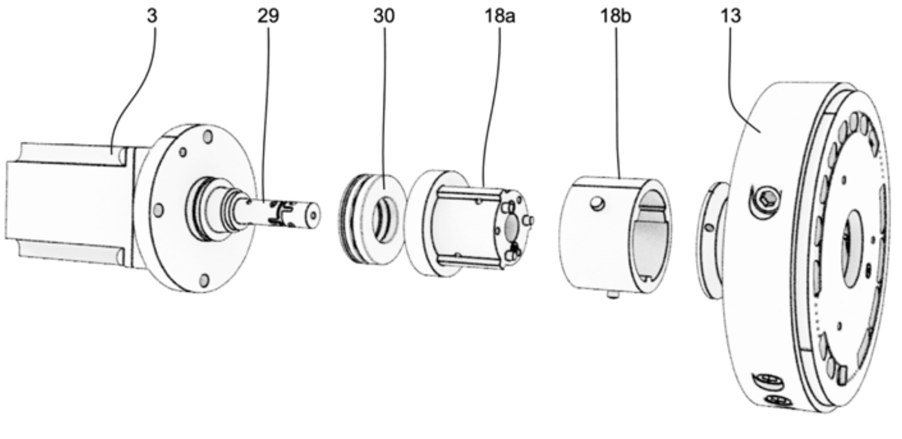


Figure 16. Servo-actuator for positioning of the port plates

3 closed loop stepper motor
13 port plate
18a rotor hydraulic servomotor
18b stator hydraulic servomotor
29 servo control axis
30 axial roller bearing

The FCT80 uses a combination of an electric servomotor and a hydraulic servomotor. The electric motor rotates a small control axis with ports that connect to the low-pressure and the high-pressure lines inside the transformer. The hydraulic servomotor has a rotor and a stator part, both having three ribs. When assembled, the stator and the rotor form a hydraulic motor with six displacement chambers. The pressure in these chambers is controlled three by three by means of the servo control axis. The electric servomotor has its own position feedback. Alternatively, the position sensor 10 in Fig. 9 can be used. The chambers of the hydraulic servomotor are connected to either the high-pressure p_1 or the low pressure p_0, depending on the position of the servo control axis. In combination with the closed loop stepper motor, the hydraulic servomotor allows a strong but also accurate and fast position control of the port plates.

5. CONCLUSION

The design of the FCT80 utilises all the floating-cup-knowledge INNAS has gathered in the past 20 years designing floating cup pumps and motors. This knowledge is combined with the concept of the Innas Hydraulic Transformer (IHT) which allows a much more compact and efficient design than combinations of two pump/motors (CHTs). New features in the new FCT80-design are:

- The central bearing of the main shaft between two flanges
- The hollow main shaft, which is used as a passage for the load flow
- The mechanical connection of the two port plates via the hollow main shaft
- The electro-hydraulic servomotor which controls the rotational position of both port plates
- The design of the port plates which are nearly 100% hydrostatically balanced in the axial direction

The FCT80 is designed for application in a real mobile machine and will be tested later this year.

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