FUNDAMENTALS OF HYDRAULIC TRANSFORMERS

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ABSTRACT

In Common Pressure Rail systems (CPR-systems), it is no longer possible to control the loads at the primary side by means of a variable displacement pump. Instead, the loads need to be controlled directly at the load (also referred to as secondary control). Rotating loads could be controlled by variable displacement motors, but hydraulic cylinders need to be controlled by means of hydraulic transformers. The problem is, however, that these transformers are not yet available on the market.

This paper discusses and analyses the principal design options for hydraulic transformers, thereby comparing several combined hydraulic transformers (CHTs), which are combinations of two pump/motors, and the Innas Hydraulic Transformer (IHT). The emphasis is on hydraulic power transformers which can be used as a general hydraulic control component, having a continuously variable control.

Keywords: Hydraulic transformer, CPR-system

1. INTRODUCTION

Most current hydraulic systems are inefficient. There are several reasons for this inefficiency. Although pumps and motors may have a high peak efficiency, the average efficiency is much lower, especially in variable displacement machines. Hydraulic systems could also be much more efficient if energy would be recuperated. Hydro-pneumatic accumulators are excellent storage devices for capturing the recuperative energy, but in almost all hydraulic applications, energy is not recuperated. Another reason for the inefficiency of hydraulic systems are the valve losses. In most hydraulic applications, a variable displacement pump is supplying oil to several loads at different pressure levels. The resulting pressure differences are throttled away and dissipated in the valves.

It is known for many years, that hydraulic transformers could eliminate many of these losses, and would strongly increase the efficiency of hydraulic machinery. First of all, they can eliminate most valve losses. Furthermore, they offer the opportunity to recuperate energy by constantly converting the pressure in the accumulator to the pressure needed at the load. The transformers will be connected to a Common Pressure Rail or CPR. This rail separates the energy supply side from the loads. It offers the opportunity for advanced power management, thereby increasing the efficiency of the pumps (which can now have a fixed displacement) and the prime mover. Previous studies have shown that these CPR-systems can reduce the energy consumption by more than 30% [1-6].

The main reason, why hydraulic transformers are generally not used, is that there are hardly any hydraulic transformers that have reached the market [7-11]. The market is still waiting for a winning design. This paper investigates the fundamentals of hydraulic transformers, to understand what is needed to make a feasible hydraulic transformer for a market introduction. The analysis will be focused on transformers which can replace conventional valve control systems for most hydraulic applications. Such a transformer is a dynamic pressure and speed controller at a power level of 10 kW to several 100 kW, output flows in the order of 100 to 400 □/min, and pressure levels up to 420 bar.

2. BASIS EQUATIONS AND DESIGN GUIDELINES

Hydraulic transformers convert hydraulic power without any principal losses: they convert a flow Q_1 at a certain pressure difference Δp_1 into another flow Q_2 at another pressure difference Δp_2, thereby keeping the product of flow Q and pressure difference Δp (i.e. the hydraulic power) constant:

$$(p_1 - p_0)Q_1 = -(p_2 - p_0)Q_2 \tag{1}$$

In this paper, oil flowing into the transformer is defined as positive. If the pressure at the low-pressure side is defined to be zero, then Eq. (1) can be written as:

$$p_1 Q_1 = -p_2 Q_2 \tag{2}$$

The equation shows that a difference in pressure level also results in a difference in flow level. A flow of for example 100 □/min at 300 bar can be transformed to a flow of 300 □/min at 100 bar. This differs from valve operation, where the flow through the valve remains constant and energy is dissipated. In a hydraulic transformer, the flow at the output Q_2 will in general be different from the flow at the input Q_1. Only if $p_1 = p_2$, then $Q_1 = -Q_2$.

In addition to the energy continuity equation (2) there is also a flow continuity equation. Aside from Q_1 and Q_2 , the continuity equation demands that there needs to be a third flow Q_0 which is coming from the low-pressure side of the hydraulic circuitry. If we assume the density to be constant, then:

$$Q_0 + Q_1 + Q_2 = 0 \tag{3}$$

Considering this third make-up flow Q_0, the energy continuity equation can be rewritten as:

$$\sum_{i=0}^{2} p_i Q_i = p_0 Q_0 + p_1 Q_1 + p_2 Q_2 \tag{4}$$

Combining Eq(4) and (3) results in Eq.(1), whereas assuming $p_0 = 0$ results in Eq(2).

One of the earliest [12] and most simple designs of a hydraulic transformer is a multi-chamber cylinder (Fig. 1). These transformers are often called pressure amplifiers [13]. The amplifier has three chambers at three different pressure levels. If the middle ring chamber has a pressure $p_0 = 0$, then:

$$p_1 A_1 v = -p_2 A_2 v \tag{5}$$

By means of valves, the piston in the amplifier can move back and forth, for instance by means of ports in a rotating device [14, 15]. It is also possible to make the cylinder symmetrical [16], having two ring chambers. This creates the possibility for a somewhat continuous operation.

Pressure amplifiers have a few disadvantages. Due to the limited switching frequency, the power density is poor. Moreover, the transformation ratio is fixed and defined by the area ratios of the piston. It is possible to create a multi-step cylinder [17-19], which allows multiple transformation steps, but this makes the design even larger.

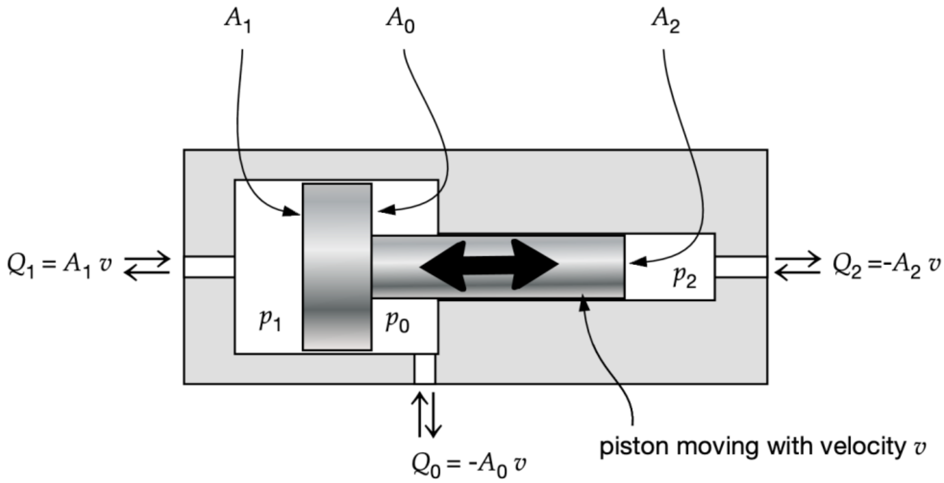


Fig. 1: Pressure amplifier

In this paper, these fixed transformers and pressure amplifiers are ignored. The focus is on hydraulic power transformers which can be used as a general hydraulic control component, i.e. as a replacement of current valve control:

1. The transformer needs at least three connections at three different pressure levels;
2. The transformer must be continuously variable;
3. The transformer should be able to regenerate hydraulic power, that is, to return hydraulic power from the load (at p_2 and Q_2) back to the high pressure rail at pressure p_1;
4. The transformer needs to be able to move oil from a lower to a higher pressure level, for instance when energy needs to be recuperated from a load pressure, which is lower that the pressure level in the high pressure accumulator. Q_0 also often needs to flow from the low pressure level p_0 to the higher pressure levels of p_1 or p_2. The flow direction of Q_0 depends on the transformation ratio p_2/p_1. If for instance $p_2 < p_1$ and Q_1 is flowing into the transformer, then Q_2 will be larger than Q_1. As a consequence, an additional flow Q_0 needs to be supplied to the transformer from the low pressure side. Also in this case, oil needs to go from a low to a higher pressure level.

In order to make oil flow from a lower to a higher pressure level, oil needs to be ‘pushed’ or displaced to this higher level. This requires some kind of pump or motor principle. Therefore, hydraulic transformers are hydrostatic machines, like hydrostatic pumps and motors. Taking into consideration the power, pressure, and flow demands, they need to be built around some kind of rotating displacement principle. Pressure amplifiers, such as the example in Fig. 1, do not fulfil these demands. They are not continuously variable, and their power and flow capacity is too small for most hydraulic applications.

Figure 2 shows the field of operation (the blue area) that the transformer must cover. The horizontal axis shows the load flow Q_2 relative to the maximum flow demand Q_{max}. The vertical axis shows the pressure or transformation ratio p_2/p_1. Hydraulic systems deliver their energy to either hydraulic motors or hydraulic cylinders. In most cases, these loads have stop-and-go operation, which means that the displacement principle used in the transformer needs to be operated frequently at these conditions as well. In the end, the transformer needs to be able to deliver any flow between $-Q_{max}$ and Q_{max}.

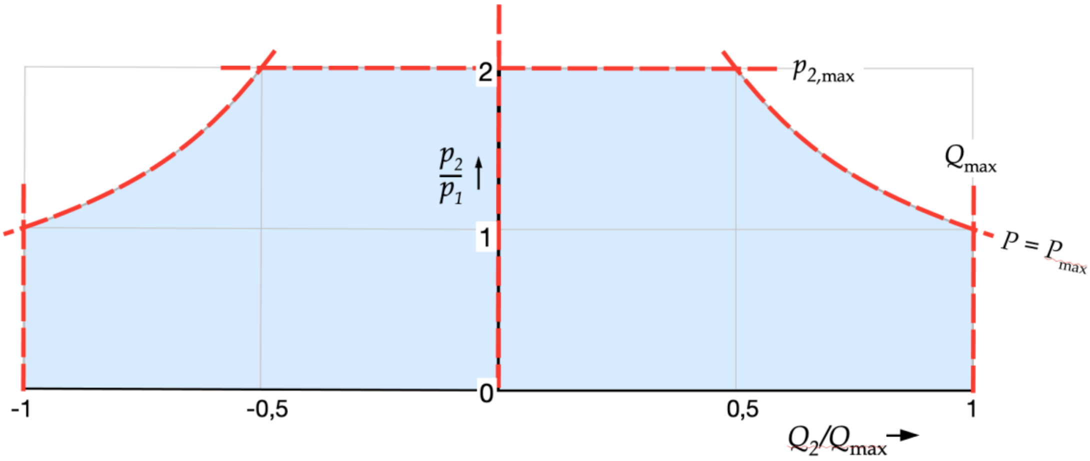


Fig. 2: Field of operation of the hydraulic transformer

The maximum pressure and power demands are also set by the load. Many, if not most hydraulically driven machines, are production machines, meaning that these are demanding applications which go to the limit of the maximum power and pressure capacity.

3. HOW TO MAKE THE TRANSFORMER VARIABLE?

There are various possibilities to make a transformer variable. One way is to connect the shafts of several fixed displacement pumps and motors, and decide by means of valves which of these pumps and motors will be active [20, 21]. Aside from becoming bulky, this design would not allow a continuously variable operation.

Another option would be to change the rotating speed. Equation (2) can be rewritten as

$$p_1 n_1 V_1 = p_2 n_2 V_2 \tag{6}$$

The minus sign on the righthand side of Eq(2) is now included in the definition of the displacement V_2. If the ratio between n_1 and n_2 could be varied, then the pressure ratio p_2/p_1 would be varied accordingly. But this would require a continuously variable mechanical transmission with the same power rating as the transformer (i.e. 10 kW to several 100 kW). Because of the size, weight, complexity and costs of such a transmission, this option will not be investigated further. In this paper the rotational speed will be considered the same for V_1 and V_2 ($n_1 = n_2$). Eq(6) is then reduced to:

$$p_1 V_1 = p_2 V_2 \tag{7}$$

The only remaining way to make the transformer variable is to change the displacement volumes:

$$p_1 \xi_1 V_1 = p_2 \xi_2 V_2 \tag{8}$$

The parameters $\xi_1$and $\xi_2$reflect the reduction of the maximum geometrical displacement ($|\xi| \in [0,1]$). The transformation ratio equals p_2/p_1:

$$\frac{p_2}{p_1} = \frac{\xi_1}{\xi_2}\frac{V_1}{V_2} \tag{9}$$

Two different concepts of such a transformer have been studied in the past:

1. A transformer based on a combination of two hydraulic machines (CHT), of which at least one has a variable displacement;
2. A transformer based on a single hydraulic machine with three instead of two pressure ports, as has been designed by Innas (IHT) [22-24].

Both concepts are evaluated in the next two sections of this paper.

4. COMBINATIONS OF TWO PUMP/MOTORS (CHT)

First designs of a hydraulic transformer based on two hydrostatic pump/motors showed up around 1970 [25, 26]. These designs were not variable. The first variable hydraulic transformers with two hydrostatic machines started to be designed around 1980 [27].

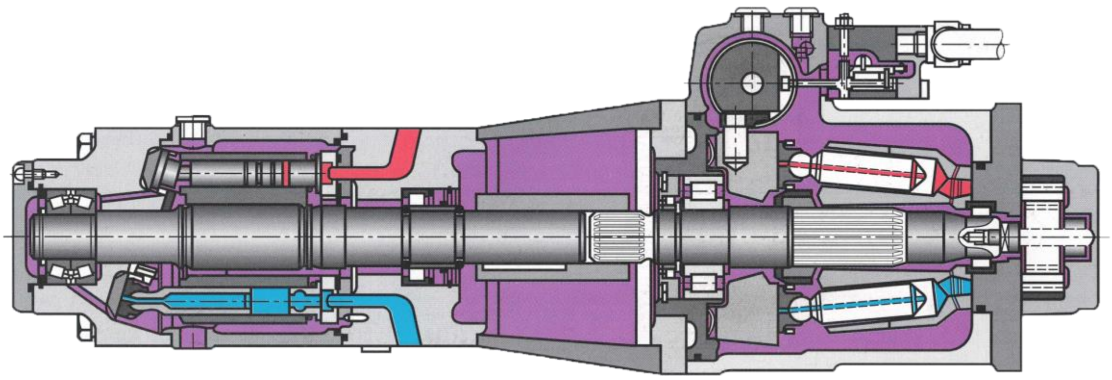


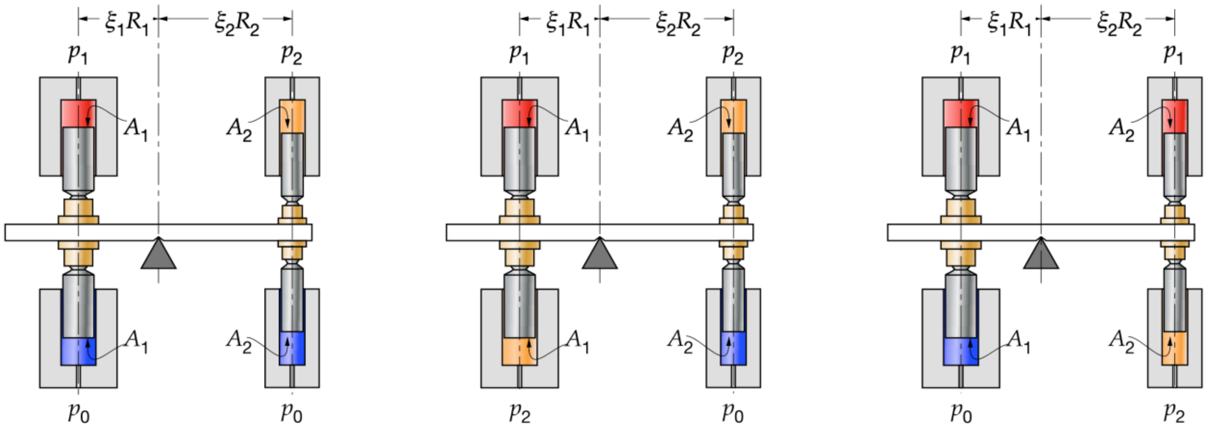
Fig. 3: Variable hydraulic transformer build as a combination of a variable displacement pump/motor and a fixed displacement pump/motor [28]

Most of these transformer designs are based on combinations of two axial piston machines, but there are also designs based on radial piston machines [29], external gear pumps [30], geroler motors [31] and vane pumps [27, 32].

These transformers can have one or two variable displacement machines. Examples of both machines being variable can be found in [30, 33]. There are various ways to connect the two machines to the p_0, p_1 and p_2-lines [29]. Figure 3 shows an example of a design in which a fixed and a variable pump/motor are combined.

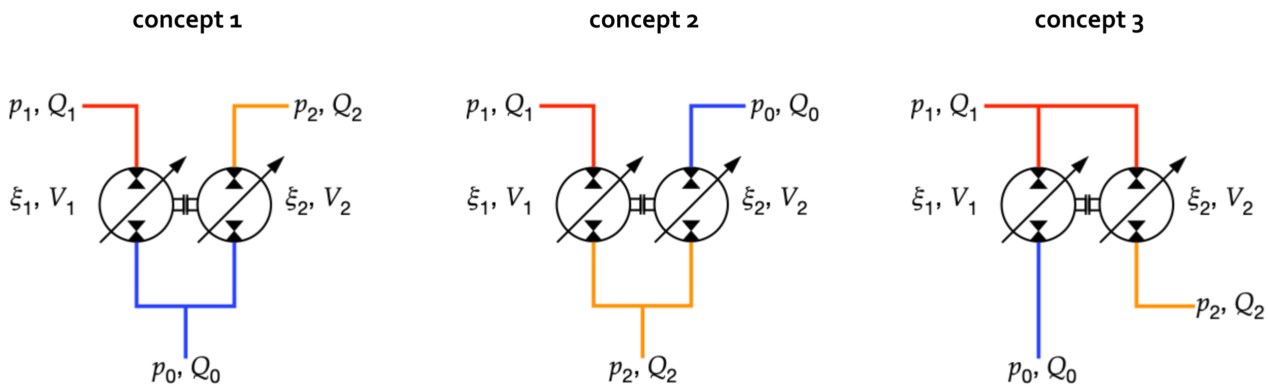
A simple way, to understand a hydraulic transformer as a combination of two machines, is to first look at a transformer which is constructed as several pistons acting on a torque beam or seesaw. Figure 4 shows three configurations of a beam which is supported in the middle by a pivot, on which four pistons act. The arm lengths $\xi_1 R_1$ and $\xi_2 R_2$ of the pistons can be changed. The two pistons on the left side of the pivot point and the two pistons on the right side move in pairs. The pressure levels in Fig. 4 change between the concepts. The torque equations and the transformation ratios of each concept are shown below the drawings (assuming no friction and $p_0 = 0$).

The same idea is realized when combining two pump/motors (see figure 5). In theory, there are even more combinations possible, but those are not meaningful [34]. Figure 5 also shows the stationary torque equations, assuming no friction and other hydromechanical losses and assuming no accelerating or decelerating inertia effects.



$p_1 A_1 \xi_1 R_1 = p_2 A_2 \xi_2 R_2$	$(p_1 - p_2) A_1 \xi_1 R_1 = p_2 A_2 \xi_2 R_2$	$p_1 A_1 \xi_1 R_1 = (p_1 - p_2) A_2 \xi_2 R_2$
$\frac{p_2}{p_1} = \frac{\xi_1}{\xi_2} \frac{A_1 R_1}{A_2 R_2}$	$\frac{p_2}{p_1} = \frac{1}{1 + \frac{\xi_2}{\xi_1} \frac{A_2 R_2}{A_1 R_1}}$	$\frac{p_2}{p_1} = 1 - \frac{\xi_1}{\xi_2} \frac{A_1 R_1}{A_2 R_2}$

Fig. 4: Hydraulic transformers with 4 pistons achting on a torque beam. The torque equations and transformation ratios p_2/p_1 assuming no friction and $p_0 = 0$



$p_1 \xi_1 V_1 = p_2 \xi_2 V_2$	$(p_1 - p_2) \xi_1 V_1 = p_2 \xi_2 V_2$	$p_1 \xi_1 V_1 = (p_1 - p_2) \xi_2 V_2$
$\frac{p_2}{p_1} = \frac{\xi_1}{\xi_2} \frac{V_1}{V_2}$	$\frac{p_2}{p_1} = \frac{1}{1 + \frac{\xi_2}{\xi_1} \frac{V_2}{V_1}}$	$\frac{p_2}{p_1} = 1 - \frac{\xi_1}{\xi_2} \frac{V_1}{V_2}$

Fig. 5: Three different concepts to combine 2 hydrostatic machines. The torque equations and transformation ratios p_2/p_1 are shown below each concept, assuming no friction and no volumetric losses, and assuming $p_0 = 0$ bar

In principle, the displacement can be varied between $-1 \leq \xi \leq 1$. However, this does not make sense for concept 1, since $\xi_1/\xi_2 < 0$ would result in negative pressure ratios. All concepts in Fig. 5 can result in infinite pressure ratios (Table 1).

Table 1: Conditions when p_2/p_1 becomes infinite

concept 1:	$\xi_2 = 0$
concept 2:	$\frac{\xi_2}{\xi_1}\frac{V_2}{V_1} = -1$
concept 3:	$\xi_2 = 0$

Concept 2 results in a strong non-linear relationship between the control parameters ξ_1 and ξ_2, and the transformation ratio p_2/p_1. Concept 2 also has a limited transformation range and is unable to cover the entire field of operation as shown in Fig. 2. Both concepts 1 and 3 can operate in the entire field of operation as described in Figure 2, but concept 3 would need a much more complicated over-centre design ($-1 \leq \xi_1 \leq 1$) then concept 1, which only needs ξ_1 to be varied between 0 and 1.

Because of the weaknesses and disadvantages of concepts 2 and 3, only concept 1 with V_1 being variable and V_2 being a fixed displacement machine ($\xi_2 = 1$) will be evaluated in this paper. Concepts 2 and 3 are sometimes considered as modes of concept 1 [35-37]. By means of valves, it should then become possible to choose which mode to choose, which would give the opportunity to improve the efficiency. This paper will not further pursue these options.

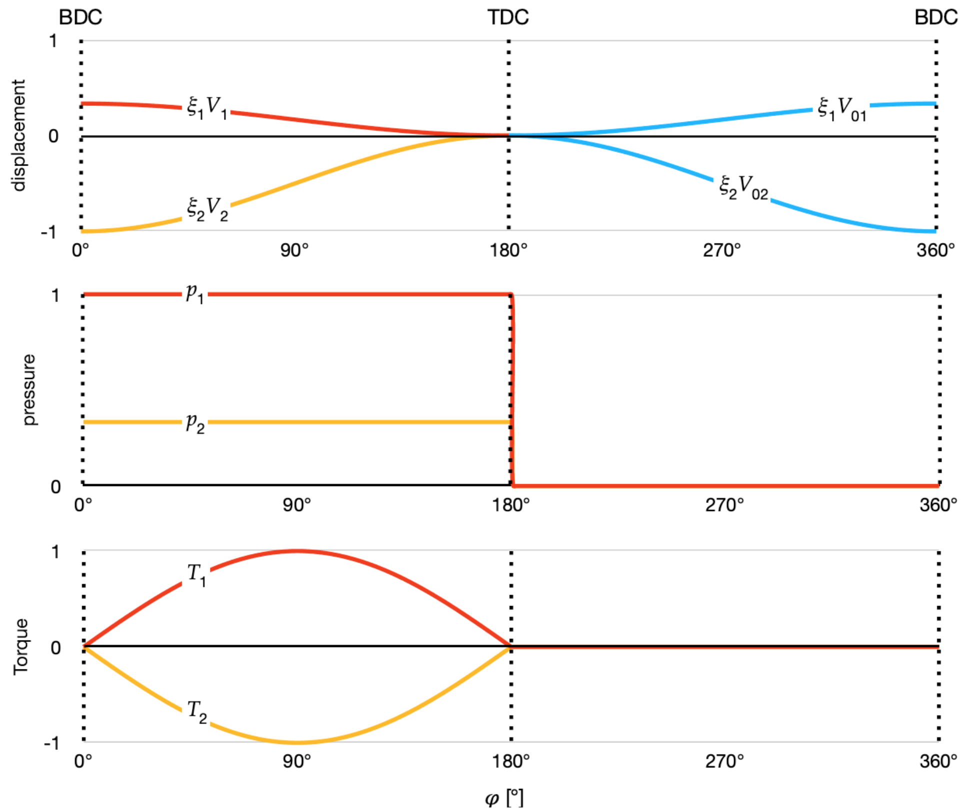


Fig. 6: Relative displacement, pressure level and torque generate of a single piston during one rotation. The diagram shows a transformer as a combination of two machines, in which machine 1 has a displacement which is 1/3 of the displacement of machine 2, and in which the pressure level during the first half of the rotation is three times as high in machine 1 as in machine 2

For transformers, which are built as a combination of two pump/motors, the principle can be explained further by plotting the volumetric displacement, the pressure level and the torque of a single displacement chamber during a single rotation (See Fig. 6). During each rotation of the shaft, each displacement chamber is connected sequentially to different ports at different pressure levels. For instance, in an axial piston pump, the cylinders are connected to the low-pressure port for about half of the rotation, and to the high-pressure port during the other half. In the example of Fig. 6, the two units have a difference of a factor 3 in displacement: $\xi_1 V_1$ of machine 1 is three times smaller than $\xi_2 V_2$ of machine 2. On the other hand, the pressure level during the first half of the rotation is three times as high for machine 1 as for machine 2. This results in a product pV i.e. a torque, which is equal in amplitude, but different in sign. The make-up flow is shown in Fig. 6 as two individual displacements $\xi_1 V_{01}$ and V_{02}. Since it is assumed that $p_0 = 0$, there is no torque generated or demanded during this second half of the rotation.

The curves which are shown in Fig. 6 are simplifications of the reality. There are no losses and the commutation in the top and bottom dead centres (TDC and BDC) are assumed to be instantaneous. In addition, in Fig. 6, the curves of machines 1 and 2 are exactly in phase. In reality, the displacement in one machine will most of the time not be synchronized with the displacement in the other. But having multiple displacement volumes (i.e. pistons and cylinders in an axial piston machine), will not change the principle of operation.

5. THE INNAS HYDRAULIC TRANSFORMER (IHT)

The conventional method of changing the displacement of piston pumps and motors is to change the amplitude of the sinusoidal piston movement, for instance by reducing the angular position of the swash plate in an inline axial piston pump (see Fig. 3). This is also the way in which, in the previous section, the transformers are controlled.

There is however another method to change the displacement, without changing the swash plate angle. In conventional pumps and motors, the filling and the delivery stroke are defined between the top and bottom dead centers (TDC and BDC). As an example, in Fig. 6, both machines have a high pressure stroke between 0 and 180°, and a low pressure stroke between 180 and 360°. But these ‘windows’ can be changed to different angles. Part of the stroke will then occur while passing one of the dead centres, and the effective displacement will be reduced.

Essential is, that in current pumps and motors, the port plate has a fixed position in the housing. But it is also possible to create a pump (or motor) in which the port plate can rotate around its own axis. The commutation can then occur outside the TDC and BDC areas, when the pistons are already moving with a considerable speed. As a result, the high and low pressure port of the port plate will rotate past the TDC and BDC. Both the effective suction stroke and the effective delivery stroke will become shorter, and the effective volumetric displacement will be reduced. This will effectively create a variable displacement pump or motor [38-42]. In case of a digital displacement pump or motor, the valve timing can offer the same control option [43, 44].

The Innas Hydraulic Transformer (IHT) uses the same principle. In addition, the design features a port plate with three ports instead of two (see Fig. 7). The ports are connected to the low-pressure line of the CPR-system at a pressure p_0 (port 0), the high pressure line of the CPR-system at a pressure p_1 (port 1) and the load line at a pressure p_2 (port 2).

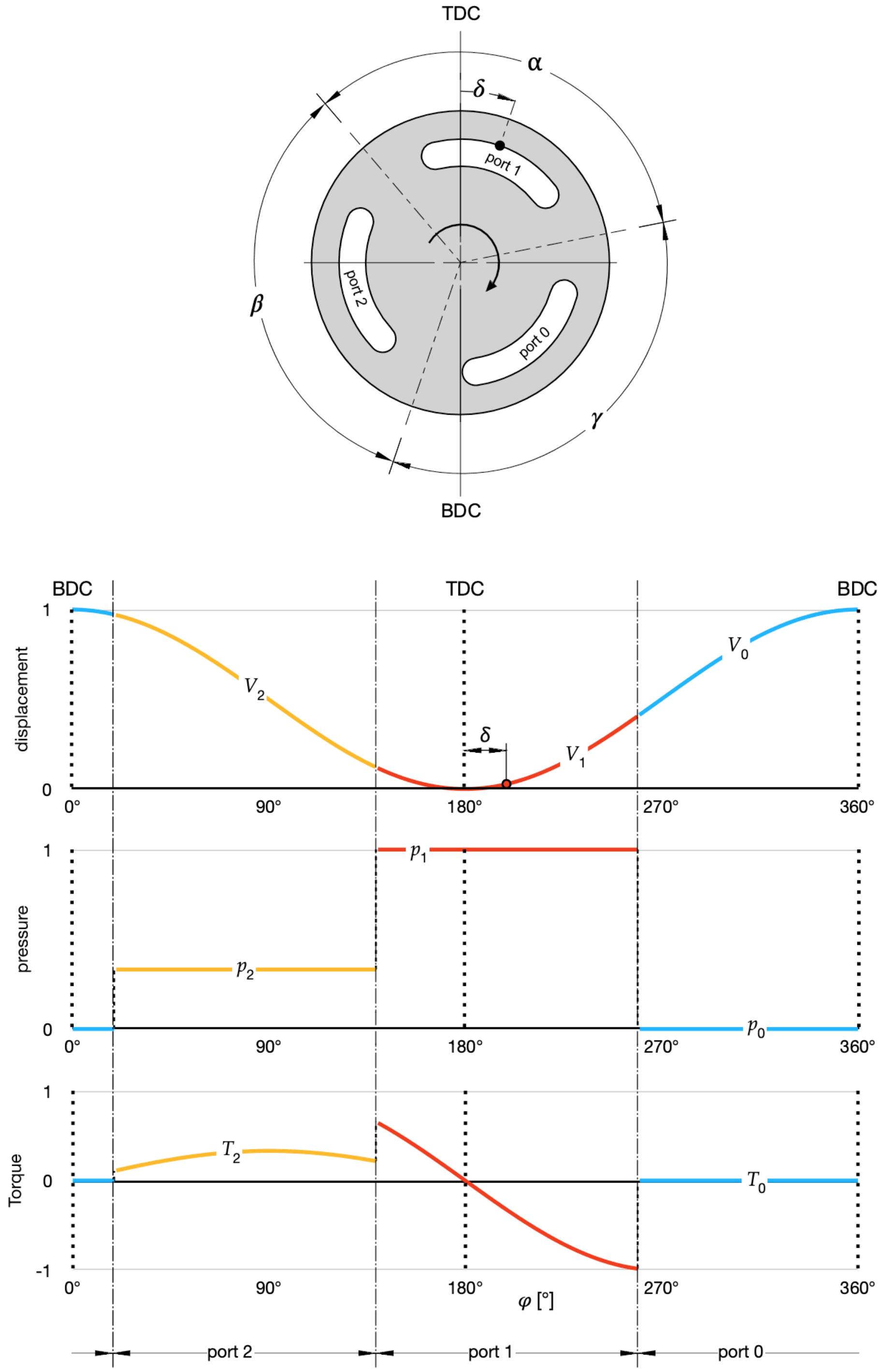


Fig. 7: The IHT which is controlled by the angular position of the port plate, and in which the port plate has three ports. The curves show the relative volumetric displacement, the relative pressure level in each of the ports and the torque generation during one revolution.

This is illustrated in Fig. 7, which shows a design of the IHT for an axial piston design. In this example, each port has the same length. Between the ports are the sealing lands which are necessary to avoid short-circuit leakage during commutation. For the port plate which is shown in Fig. 7, a cylinder block with 9 pistons is assumed, resulting in a commutation zone of about 360°/9 = 40°. Considering the movement and the size of the ports in the cylinder block, each port in the port plate has an effective length of 120°. Other configurations and designs, with different port lengths, are also possible. For marking the rotational position of the port plate, a point in the middle of port 1 has been chosen as a reference. The angular distance between this point and the TDC-BDC-line is the control angle δ. In this drawing, the cylinder block, which rotates on top of this port plate, is assumed to rotate in clockwise direction.

The transformation ratio p_2/p_1 is not anymore dependent on the geometrical displacement V but on the effective arc lengths α, β and γ of the ports and the control angle δ:

$$\frac{p_2}{p_1} = \frac{sin\left(\frac{\alpha}{2}\right)sin(\delta) - \frac{p_0}{p_1}sin\left(\frac{\gamma}{2}\right)sin\left(\delta - \frac{\beta}{2}\right)}{sin\left(\frac{\beta}{2}\right)sin\left(\delta + \frac{\gamma}{2}\right)} \tag{10}$$

When $p_0 = 0$:

$$\frac{p_2}{p_1} = \frac{sin\left(\frac{\alpha}{2}\right)}{sin\left(\frac{\beta}{2}\right)} \frac{sin(\delta)}{sin\left(\delta + \frac{\gamma}{2}\right)} \tag{11}$$

The diagram of Fig. 8 shows the calculated relationship between p_2/p_1 and δ for two different port plate designs and $p_0 = 0$.

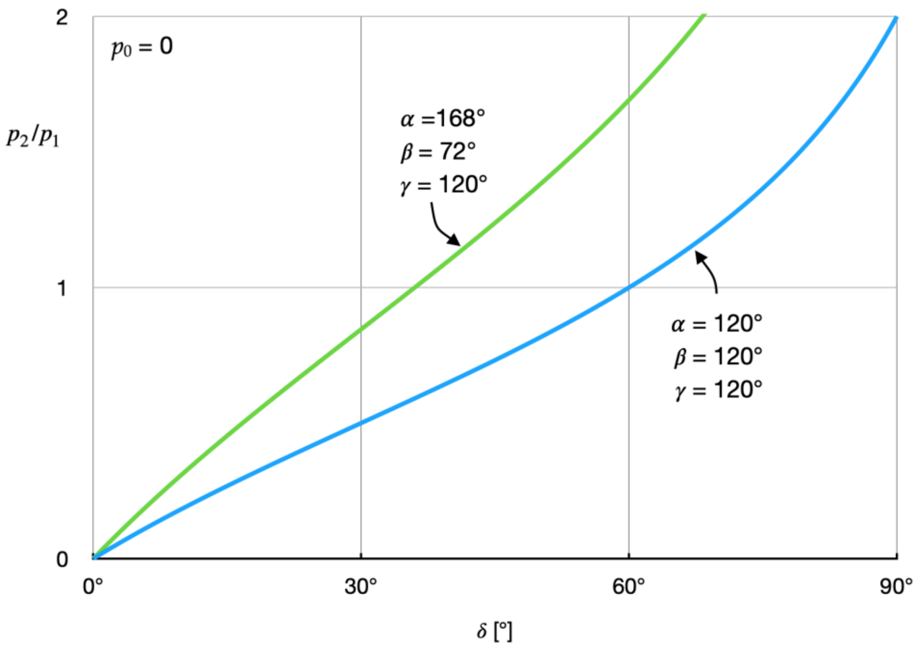


Fig. 8: Transformation ratio p_2/p_1 for the Innas Hydraulic Transformer (IHT) for two different configurations as a function of the control angle δ.

The IHT can control the complete range of transformation ratios from 0 to an amplification ratio of $p_2/p_1 = 2$, and even higher if needed. To achieve this, the port plate needs to be rotated to a maximum angle of 68° to 90°, depending on the port plate design.

Because there is no ISO-symbol for this type of hydraulic transformer, INNAS has designed and defined this symbol for the IHT:

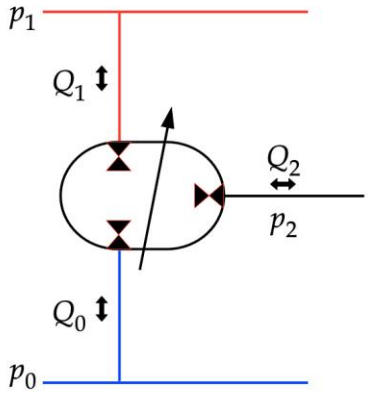


Fig. 9: Symbolic representation of the IHT

6. TRANSFORMER SIZES

The analysis of the previous section resulted in two principle designs of a hydraulic transformer that fulfil the following basic requirements:

- being continuously variable;
- can be operated in the entire field of operation, as defined in Fig. 2;
- can recuperate energy

The two principles are shown below:

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

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

a. Combination of two pump/motors (CHT)

b. Innas Hydraulic Transformer (IHT)

Fig. 10: Hydraulic transformer designs. The transformation ratio p_2/p_1 is valid for $p_0 = 0$

Both transformers need to achieve a transformation ratio $p_2/p_1 = 2$. For the CHT, this means that $V_1 = 2V_2$ to get to this maximum pressure ratio at a maximum displacement of V_1 (i.e. $\xi_1 = 1$). For the Innas Hydraulic Transformer (IHT), the transformation ratio is not dependent on the maximum volumetric displacement, but on the effective arc lengths of the three ports. Assuming all ports are equal ($\alpha = \beta = \gamma = 120°$) the control angle δ needs to be 90° to make $p_2/p_1 = 2$.

The flow output of the transformer based on two pump/motors is:

$$Q_2 = \frac{nV_2}{1000} \tag{12}$$

For the Innas Hydraulic Transformer the flow output equals:

$$Q_2 = \frac{nV_{IHT}}{1000\cdot 2}\left[cos\left(\delta-\frac{\alpha}{2}-\beta\right)-cos\left(\delta-\frac{\alpha}{2}\right)\right] \tag{13}$$

In these equations the rotational speed n is defined in rpm and the flow Q in □/min. The volumetric displacement V is defined in cc/rev.

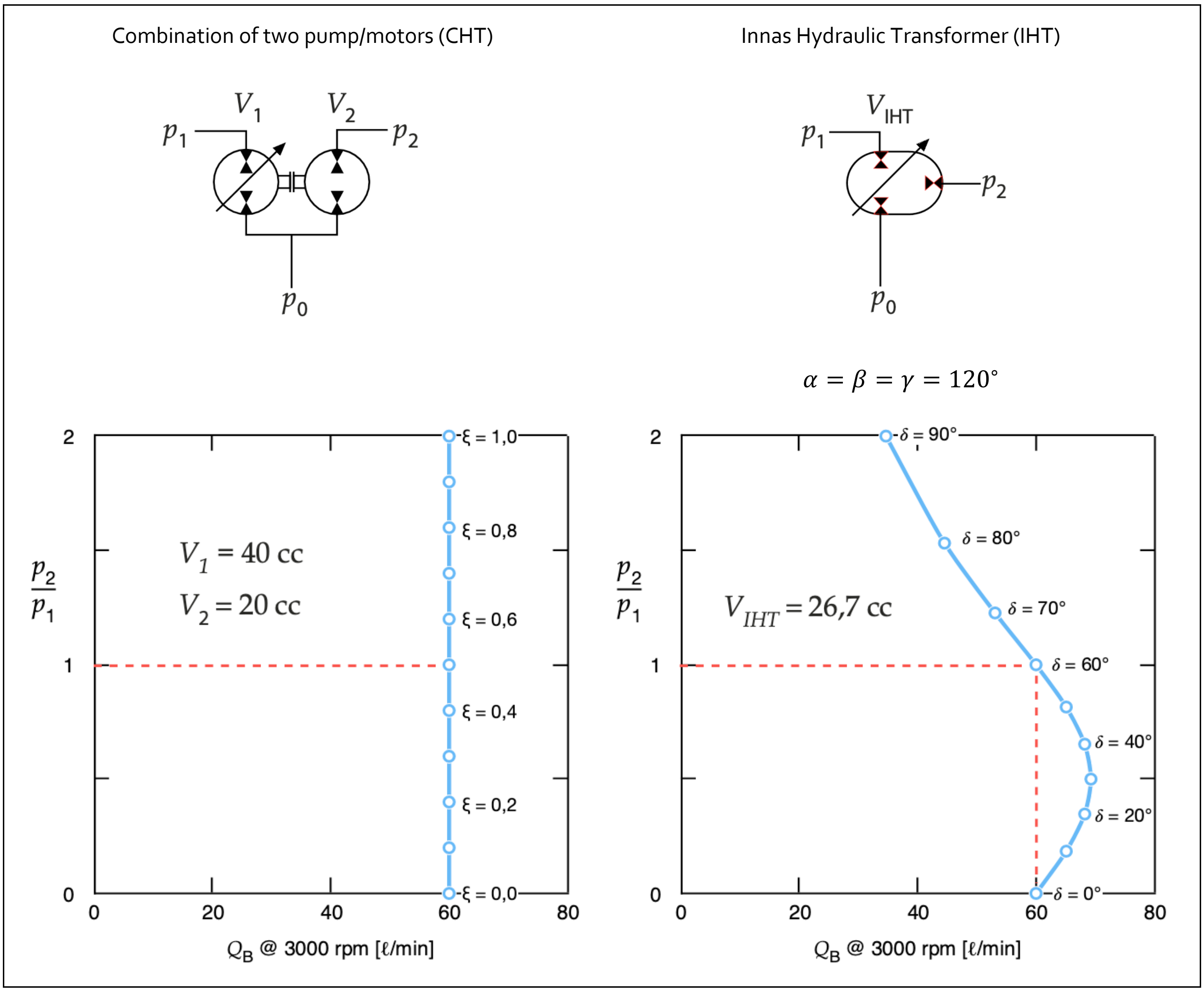


Fig. 11: Calculated flow Q_2 and transformation ratio p_2/p_1 for the two considered transformers ($p_0 = 0$). The geometrical displacements V_1, V_2 and V_{IHT} are calculated under the conditions that the transformers:

- can achieve a transformation ratio $p_2/p_1 = 2$
- can create a flow $Q_2 = 60$ □/min at $n = 3000$ rpm and $p_2/p_1 = 1$

To compare the two designs a flow capacity of 60 □/min is chosen as an example, to be reached at a rotational speed of 3000 rpm and a pressure ratio $p_2/p_1 = 1$. As before, the ports in the Innas Hydraulic Transformer are assumed to be equal ($\alpha = \beta = \gamma = 120°$). To get to the demanded flow of 60 □/min, the CHT needs a variable displacement pump/motor with a maximum displacement of 40 cc and a fixed displacement pump/motor with a displacement of 20 cc per revolution. In contrast the IHT needs a rotating group with a total geometrical displacement of only 26,7 cc per revolution. The diagrams of Fig. 11 show the calculated relationship between the transformation ratio p_2/p_1 and the output flow Q_2.

The comparison clearly shows that, in principle, the Innas Hydraulic Transformer only needs a single rotating group which has a geometrical displacement of 26,7 cc/rev, whereas the combination of a variable displacement pump/motor and a fixed displacement pump/motor needs a total displacement of 60 cc/rev. In contrast, the IHT has a smaller flow output at the highest transformation ratio. This corresponds to the pressure and flow demands of most hydraulic applications, where the flow demand is reduced at higher load pressures. The relationship between flow and pressure demand more or less follows the hyperbolic constant power line.

If the displacement of the IHT would be increased to 46,2 cc/rev then the flow output at the maximum transformation ratio of the IHT would be equal than of the CHT. But then the maximum flow output of the IHT would go up to 120 □/min at $\delta = 30°$, twice as high as for the CHT.

Also, the IHT can result in very high transformation ratios:

$$p_2/p_1 \rightarrow \infty \text{ when } \delta \rightarrow -\frac{\gamma}{2} + k\pi \qquad (14)$$

Considering that $p_2/p_1 \in [0,2]$ the control angle will always be smaller than the critical value.

7. OTHER DEMANDS

Next to the four design requirements that were discussed in section 2, there are several other more practical demands for the application of a hydraulic transformer: There are several demands for the application of hydraulic transformers:

- The hydrostatic principle needs to be variable
- It must be possible to run the transformer at close to zero speed and breakaway, even at high pressure levels and loads, without excessive wear and torque losses
- Volumetric losses need to be as small as possible, especially at low rotational speeds
- It should preferably also be possible to run the transformer at high rotational speeds
- The noise, vibration and harshness (NVH) levels need to be low
- The transformer must be capable of handling loads up to 420 bar continuously
- The overall efficiency needs to be high in a wide range of operating conditions. This includes possible losses of the displacement control

The first demand is that the displacement can be varied. This excludes all gear pumps. There is some research making gear pumps variable [45-47], but these designs are rather immature. Aside from a limited pressure, flow and power range, the most important disadvantages are the limited range in which the displacement can be varied. Other principles, which cannot be varied, are the geroler and gerotor principle, helical pumps, and multi-lobe radial piston machines.

This study focusses on hydraulic transformers as a general substitute for current valve control based hydraulic systems, i.e. to control hydraulic cylinders and possibly also hydraulic motors that have a frequent stop-and-go operation. Consequently, hydraulic transformers will also require having frequent stops and breakaways, often even at high pressure loads. Most hydrostatic principles rely on hydrodynamic lubrication and cannot run for an extended period at low rotational speeds.

Another concern is the volumetric loss and internal leakage at low rotational speeds. These losses result in a dead band for pumps: an operating range in which the pump does not deliver any net effective flow because the volumetric losses are higher than the pump displacement. In a motor, the volumetric losses result in slippage. Volumetric losses cannot be eliminated completely, but in hydraulic transformers, the volumetric losses need to be as small as possible.

From a size perspective, it will be impossible to be as small and compact as a valve segment of a load-sensing block. Nevertheless, the transformers should preferably be small. Aside from the size constraints mentioned in the previous section, this implies that the maximum rotational speed of the transformer needs to be as high as possible. Doubling of the rotational speed not only halves the volume of the rotating parts, but also of the size of the housing and the actuators which are needed to change the displacement. Hydrostatic principles have different speed constraints. Some are due to cavitation, but also centrifugal forces and barrel tipping [48-50] in axial piston machines can limit the maximum rotational speed.

Another requirement of hydraulic transformers is noise, vibration and harshness (NVH). Most certainly in battery driven applications, but also in CPR-systems with accumulators, the noise level of the hydraulic system is of much bigger importance than in conventional drives, in which the diesel motor dominates the noise level. The noise level is generally dependent on the rotational speed, the pressure level, and the number of displacement chambers (i.e. pistons) of the rotating group.

Obviously, the hydrostatic principle for a hydraulic transformer needs to fulfil the pressure demands of current hydraulic applications. Pressure levels of 350 to 420 bar are currently the peak pressure demands. Not all hydrostatic principles can handle these pressure levels.

Finally, it is important that the hydrostatic principle has a high overall efficiency in a wide range of operating points and conditions. For instance, if, in a CHT, both machines have an efficiency of 90%, then the total efficiency of the transformer will only be 81%. If this same transformer also recuperates energy with the same efficiency, then the overall cycle efficiency will be limited to 65,6% ($0{,}9^4 \times 100\%$). Many pumps and motors don not have an overall efficiency of 90%, especially variable displacement pumps. For hydraulic transformers, the highest possible efficiency is needed.

For these reasons, INNAS has developed the new floating cup principle. It is a multi-piston principle which strongly reduces the torque ripple and improves the controllability of the transformer. Aside from having a high efficiency, it is also possible to operate floating cup machines starting from standstill and at low rotational speeds, even when the pressure is high (which it always is at the high pressure CPR-port).

8. CONCLUSION

This article derives the most important, fundamental equations for hydraulic transformers: the equations for the transformation ratio between output and input pressure, and the flow equations of the transformer. It has been shown that the principle of the Innas Hydraulic Transformer (IHT) is much more compact than a combination (CHT) of a variable displacement pump/motor and a fixed displacement pump/motor.

The demands of hydraulic transformers are different from conventional pumps and motors. The efficiency needs to be higher (also at low rotational speeds, in combination with high pressure levels), the dynamic response needs to be higher, and the noise level needs to be lower. The floating cup principle is specifically designed for these demands and is a better fit than other hydrostatic pump and motor principles. In addition. That being said, both the floating cup principle and the IHT-concept might not be the only solutions, and better designs might be found in the future. This paper gives an overview of the guidelines and demands for such new designs.

NOMENCLATURE

A	Area	m^2
n	rotational speed	rpm
p	pressure	bar
p_0	pressure level low pressure CPR-line	bar
p_1	pressure level high pressure CPR-line	bar
p_2	pressure level load line	bar
Q	flow	□/min
Q_0	flow low pressure CPR-line	bar
Q_1	flow high pressure CPR-line	bar
Q_2	flow load line	bar
R	arm length	m
v	velocity	m/s
V	geometrical displacement	cc/rev
α	effective arc-length of port 1	
β	effective arc-length of port 2	
γ	effective arc-length of port 0	
δ	control angle of the IHT port plate	
ξ	displacement ratio	-
φ	angular position	
CHT	Combined Hydraulic Transformer	
CPR	Common Pressure Rail	
IHT	Innas Hydraulic Transformer	
BDC	Bottom Dead Center	
TDC	Top Dead Center	

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