2000-01-2561

The Innas Hydraulic Transformer The Key to the Hydrostatic Common Pressure Rail

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ABSTRACT

Through a series of incremental innovations, load sensing (LS) has become the current state-of-the-art in hydrostatic transmission technology. The Common Pressure Rail (CPR) is an alternative way to create a hydrostatic transmission. Although CPR systems offer considerable advantages over LS systems, they never really broke through. The main reason for this was the lack of a good solution to drive linear loads from a common pressure rail. The Innas Hydraulic Transformer (IHT) has been developed to fill this gap. It is a young component but has matured to a stage where a series production within the next two years should be possible. Because of its simple construction, its dynamic capacity, its low control power requirement and its efficiency it can be used to control both linear and rotary loads from a common pressure rail. With the IHT, the potential of CPR systems can finally be unleashed.

INTRODUCTION

The vast majority of modern hydrostatic drivelines are based on the ‘imposed flow’ hydrostatic transmission type. In this transmission in its purest form, a positive displacement pump sends an oil flow to a load. The output flow is set by controlling the speed of the pump and sometimes also its displacement. The load is forced to move with a speed corresponding to this flow. The load pressure is not controlled but is a result of the loads reaction to the enforced speed.

When used to drive only one load, this system layout results in a good efficiency and controllability. In most practical systems, however, more loads have to be driven by the same pump. In that case, priority and control problems arise. Driven by the need to solve these problems hydrostatics have evolved to the load sensing (LS) system type. Although LS is an elegant solution, it resolves control and priority issues only partly and at the cost of a significant reduction in efficiency and a substantial increase in costs.

A fundamentally different approach to the hydrostatic driveline is the hydrostatic common pressure rail (CPR). In this system type, a pump supplying flow to a hydrostatic rail is controlled in such a way that the CPR is kept to a predefined pressure level. All loads are controlled directly at the load, from the same predefined rail pressure. This explains why this transmission is often referred to as being of the ‘imposed pressure’ type. The CPR effectively separates the power source from the loads and the loads from each other, much like the electrical net separates the power plant from the end users and the end users form each other.

Notwithstanding the advantages associated with the CPR approach, it is not widely used. This is mainly due to the lack of a good and efficient way to drive linear movements from the (semi) constant rail pressure.

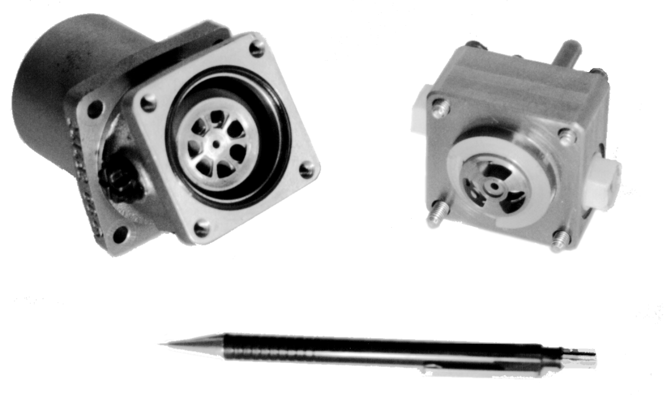


Figure 1: A 20 kW IHT prototype based on a 5cc Mannesmann Rexroth axial piston unit (A2FM5).

In 1996, the Dutch engineering company Innas BV designed the Innas Hydraulic Transformer in order to fill this gap. This ‘IHT’ could well prove to be the missing link to the hydrostatic common pressure rail.

THE HISTORY OF HYDROSTATIC SYSTEMS

THE STARTING POINT

The basis for modern oil-based hydraulics was the hydrostatic transmission developed in 1905 in the US by Williams and Janney [1]. Its first application was in the elevation and control system for the guns of the USS Virginia [2]. In figure 2 the layout of the USS Virginia is given, as well as a picture of its sister ship, the USS New Jersey.

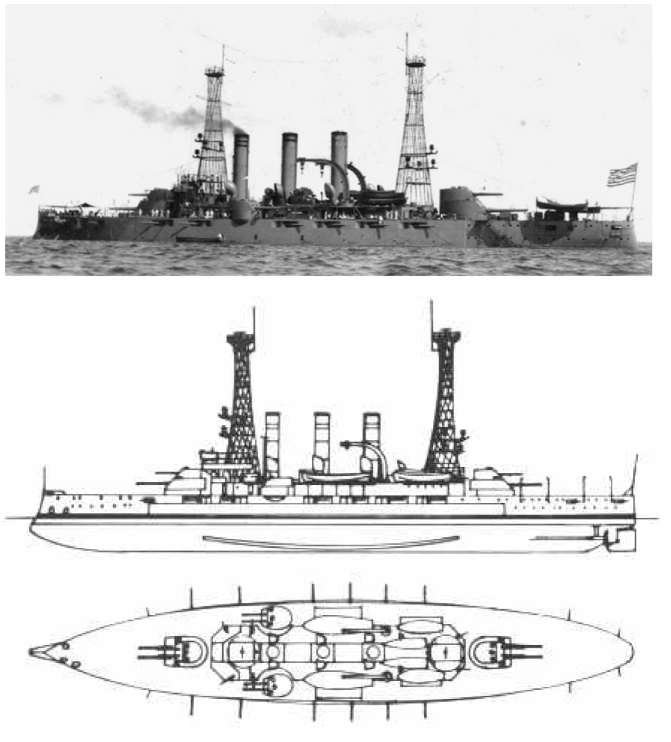


Figure 2: The USS Virginia (drawing) and its sister ship, the USS New Jersey (picture).

Until after the Second World War, military and aircraft applications were the predominant field of use for oil based hydrostatic transmissions. The spatial freedom they introduced and their superior controllability made them the better choice over mechanical transmissions, even if their power density was still somewhat poor and oil leakage was considerable. Because of their poor power density, electrical transmissions could not compete in these applications.

HYDROSTATICS IN MOBILE MACHINERY

The breakthrough of hydrostatic transmissions in the mobile machinery market is of a much later date. Until after the Second World War, not much of a mobile machinery market existed. Excavators are the most distinct exception. Large, cable actuated shovels and excavators have been built since the first half of the nineteenth century. They were used in civil engineering projects, in sewer and piping contracting and in strip-mining. In these fields, hydrostatically actuated systems could not compete because they had a smaller reach and –at that time- a lower power density.

During the economic uplift starting directly after the War, mobile machinery suited for smaller infrastructural projects was in great demand. This lead to the breakthrough of hydrostatic transmissions in mobile machinery. The emerging of the market for smaller mobile machines, the maturing process of mobile hydrostatics and the way eventually also the large excavator market was taken, are described by Christensen in [3].

The hydrostatic systems used in mobile machinery started their evolution from the systems used in the military and aircraft applications. Essentially, these were of the ‘imposed flow’ system type described in the introduction. The general layout of this system type can be seen in figure 3. This so called ‘pump-control’ results in good efficiency and controllability.

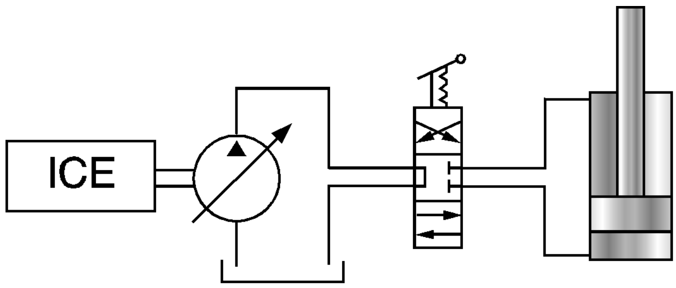


Figure 3: The ‘imposed flow’ hydrostatic system type.

This system layout applies to a situation where one pump controls one load. When an application requires more loads to be powered from the same energy source at the same time, a truly primary control scheme can still be maintained. In that case, more (variable) pumps have to be coupled to the energy source and each has to be assigned to a load. The pumps can be split between loads that do not have to operate concurrently. The system costs, however, increase substantially with the increasing number of (variable) pumps. Therefore, a ‘one pump’ strategy is often preferred.

When all loads are simply coupled to the same pump, a fundamental problem arises. Flow follows the path of the least resistance, so the load requiring the lowest pressure will move first or fastest. Only when this pressure rises, a different load can start moving or accelerating. With this kind of system behavior, precise control is virtually impossible.

If in this one-pump set-up throttling valves are introduced between the flow source and each load, the loads can be synchronized by increasing small loads artificially. This system layout requires great operator skills and introduces substantial throttling losses. It is still used in applications like the mini excavator, where system costs have to be minimized. In these vehicles, the problems mentioned are partly circumvented by introducing more pumps and grouping non-concurrent functions around these pumps.

LOAD SENSING SYSTEMS

Still fairly recently, ‘load-sensing’ (LS) hydrostatics were introduced in order to improve the efficiency and controllability of hydrostatic drivelines. The term ‘load-sensing’ is subject to definition and is used for many different system types. All LS systems have in common that the load pressures are sensed and used in a control scheme. Two control strategies based on the sensed load pressures are important:

1. The highest load pressure can be used to control the displacement of the supply pump in such a way that its output pressure is always just above the highest load pressure. In this way, all load pressures can be realized by using valves to throttle down from the pump output pressure. The throttling losses are minimized but can still be considerable, when other loads require pressures clearly below the highest load pressure.
2. The former strategy can be elaborated by using each sensed load pressure to control a pressure balance upstream of the control valve for that load. Each pressure balance is connected in such a way that the pressure drop over the corresponding control valve is kept constant. Thus, the relationship between valve position and load speed is independent of the load pressure. This provides a constant ‘stick’ feel to the operator, which greatly enhances controllability.

Figure 4 gives an example of a modern LS system of the second type [4]. Apart from its control valve (1), each section contains a valve (2) that combines the pressure balance function with the pressure comparison function. The highest load pressure is send to the LS signal circuit (3).

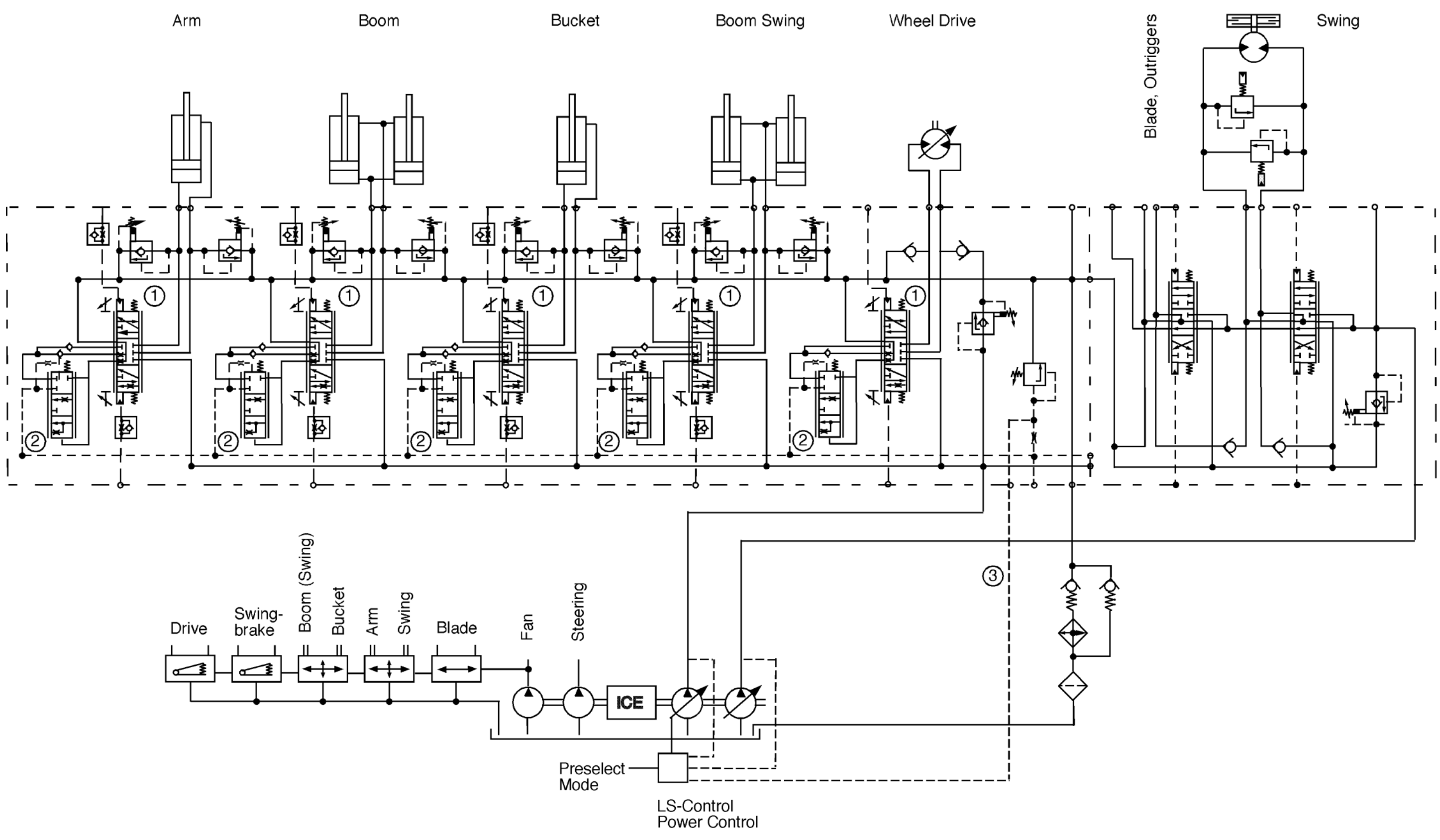


Figure 4: Load sensing hydrostatics in a NOBAS excavator [4].

COMMON PRESSURE RAIL SYSTEMS

A fundamentally different approach to the hydrostatic driveline is the common pressure rail (CPR) concept. In a CPR system, all loads and at least one energy source attach to the common pressure rail. The energy source is controlled in such a way that the pressure in the rail is kept to a predefined level. The loads connected to the rail are controlled at the secondary side, directly at each individual load. Hence, these systems are also referred to as 'secondary controlled' systems. An example of this system type is given in figure 5.

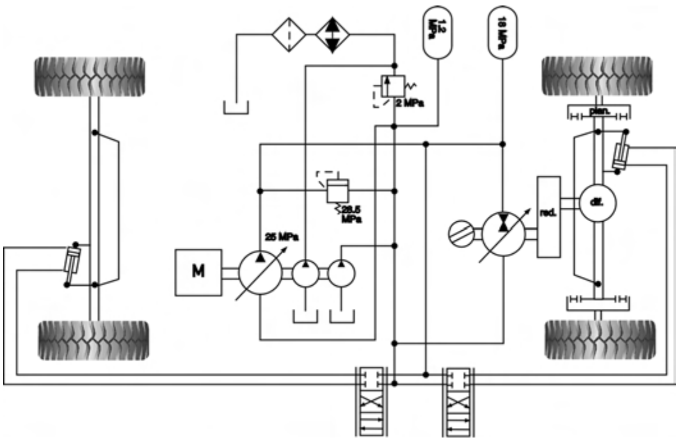


Figure 5: The CPR system of a container carrier

The advantages of CPR systems have been known for a long time:

- The pressure rail effectively separates the loads from the energy source and from each other. Because an accumulator is attached to the CPR, the energy source does not directly interact with the loads. It can be controlled with a simple and slow control strategy. All loads are controlled individually, from the same predefined rail pressure. Therefore the loads are, by principle, mutually independent.
- As there are no throttling losses, secondary controlled systems have a very high efficiency potential.
- When appropriate in an application, the accumulator can be laid out for power peak shaving and recuperation of energy from the loads. This implies a further increase of the efficiency and a smaller energy source, as it can be laid out for the average rather than the maximum power required.
- Because control is done directly at the loads, the path between controlling element and load is stiff. Low frequency oscillations, common in purely flow-controlled systems with long hydraulic circuits between pump and load, do not occur. Furthermore, the dynamics of the energy source do not influence the loads. For these two reasons, very precise control is possible. This is also the main reason why for the container carrier of figure 5, a CPR system with secondary control was chosen. Only with this system, the automatically guided vehicle can be positioned with the required accuracy.

Work on secondary controlled systems started in 1977 and mainly focused on control strategies for components attached to the CPR and on the components themselves.

SECONDARY CONTROL STRATEGY

If, like in figure 5, a hydrostatic motor is used to drive a rotary load from a rail with a predefined pressure, the driving torque is directly determined by the units displacement. With a variable unit, the displacement and thus the driving torque can be adjusted. Most loads, however, require the load's speed or position to be set rather then the load torque. To achieve this in a CPR system, the load speed must be sensed and used in a feed back loop with the displacement as the control input.

From the introduction of the concept of secondary control, a lot of research has been directed at achieving a quick and stable speed control. The typical secondary control strategies resulting from this research will not be treated in the context of this paper. Information can be found in [5, 6, 7, 8, and 9].

COMPONENTS FOR SECONDARY CONTROL

In order to be able to drive rotating loads from a CPR in a secondary control scheme, slightly adapted variable motors were designed. This mainly implied ensuring that the swash plate angle could be changed fast enough to accommodate the requirements of secondary control. This requires fast actuators and because of that, secondary controlled motors are expensive.

One of the strongest points of hydrostatic transmissions, especially in mobile machinery, is the unequalled force density of the hydraulic cylinder. In order to present a real system option, CPR systems have to offer a competitive solution to drive translating loads from the common pressure rail. This solution proved hard to find.

Throttling the rail pressure down to the level required by the cylinders is a possibility. This, however, would introduce energy losses higher than in the LS alternative, as the rail pressure will generally be higher than the controlled pump output pressure in a load sensing system.

Mannesmann Rexroth developed a hydraulic transformer with which linear loads could be driven from a CPR without throttling losses. In this paper, this transformer type will be called the 'conventional' transformer. With the conventional transformer, in theory, the rail pressure can be transformed to any level required by the cylinder load. The transformation factor Π, defined as the theoretical ratio of the load pressure p_l to the rail pressure p_0, is continuously variable between zero and a maximum value. The transformer consists of a fixed axial piston unit, coupled mechanically to a variable axial piston unit. Figure 6a shows the construction of the transformer. Figure 6b shows one of the most common ways to connect it to the rail, the tank and the load.

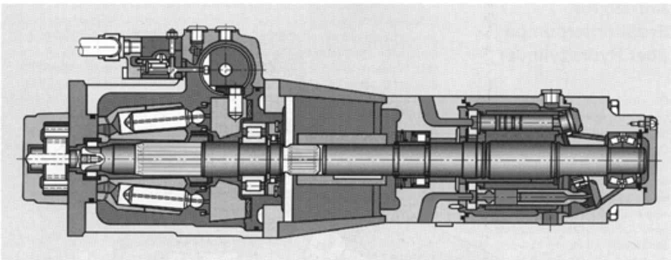


Figure 6a: Construction.

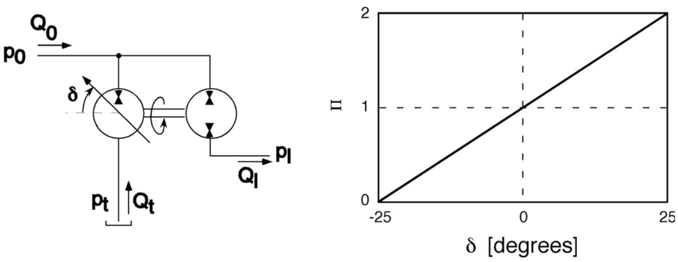


Figure 6b: Schematic and transformation curve.

Figure 6: The conventional hydraulic transformer

A detailed description of the way this hydraulic transformer functions is given in the Appendix. Here only the pressure transformation curve is given, i.e. the transformation factor Π as a function of the variable unit's swash plate angle δ. Assuming the maximum displacement of the variable unit to be equal to the displacement of the fixed unit, neglecting internal losses and assuming a pressureless tank connection, Π can be calculated to the curve given in figure 6b.

At first sight, the transformer concept described above looks promising. The transmission of energy through two axial piston units, however, leads to a poor average efficiency. In almost all points of operation, at least one of the units operates under part load conditions. Consequently, the losses associated with this solution are only marginally lower than the losses in the throttling solution. Furthermore, this type of transformer is bulky, heavy and complex. It requires the same large control forces and expensive actuators as secondary controlled motors.

Considering all this, it is not surprising that this conventional transformer did not provide the solution for linear loads, that CPR technology needed to break through. As has been argued before, rotary loads could be powered from a CPR by use of secondary controlled motors, but these were rather expensive. For these reasons, CPR systems with secondary control were only used in a small number of niche applications, like the one presented in figure 5, where their superior controllability provided a competitive edge. The IHT was developed to change this situation.

THE WORKING PRINCIPLE OF THE IHT

The basic principle of operation of the IHT is identical to the transformer described above. Again, the details are explained in appendix A. Here only the functional results are presented. The construction of the IHT is quite different from that of the conventional transformer and much simpler. This simplicity is the main reason why the IHT has a potential the conventional transformer never had.

IHT prototypes so far, have all been based upon positive displacement axial piston units. The essential difference of an IHT to a standard axial piston unit, is in the design of the port plate:

- The port plate in a standard axial motor has two kidneys, one connected to the supply line, the other to the tank line. The IHT has three kidneys, one is connected to the rail pressure, the second is connected to the tank pressure and the third is connected to the hydrostatic load.
- The port plate in a standard hydrostatic unit is fixed. In order to be able to vary the transformation factor Π, the port plate in the IHT can be rotated over a limited angle.

The construction of the first prototypes is shown in figure 7a. Figure 7b shows how the IHT is connected to the rail the tank and the load. The schematic also introduces the symbol that was coined for the IHT.

Assuming a pressureless tank connection and neglecting internal losses, the transformation factor Π as a function of the port plate angle δ, can be calculated. The curve for a port plate configuration with three 120-degree kidneys is shown in figure 7b. The curve is almost linear up to a transformation factor 1 at δ = 60 degrees, and progressively rising in the amplification region ($\delta > 60$ degrees).

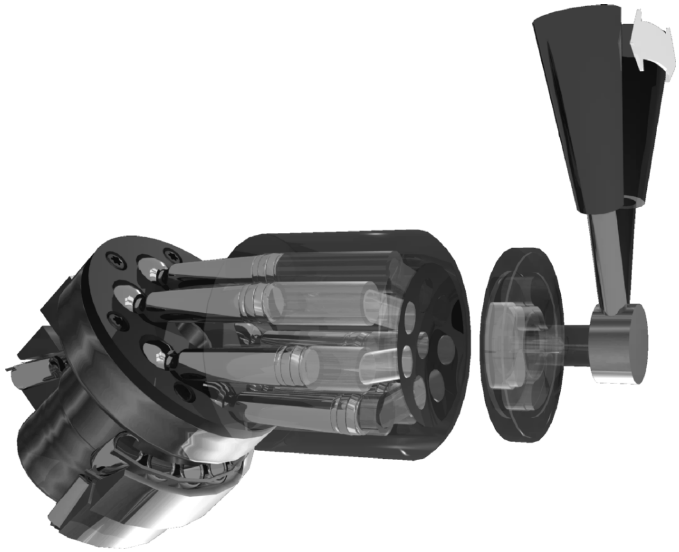


Figure 7a: Construction.

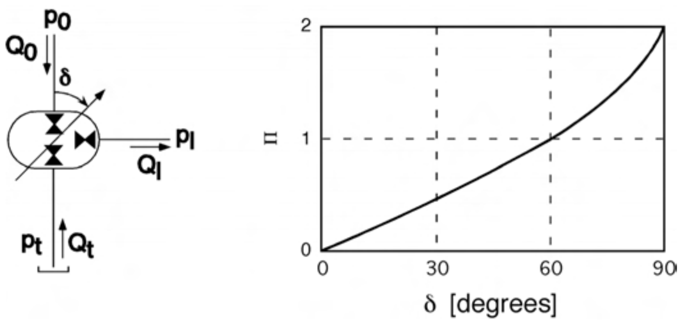


Figure 7b: Schematic and transformation curve.

Figure 7: The first IHT prototype, the port plate has three kidneys of 120 degrees each.

The unit is symmetrical to rotation of the port plate in positive or in negative direction. Therefore, the curve for δ-angles between zero and +90 degrees, may be mirrored with respect to the Π-axis to give the curve between zero and –90 degrees . When the port plate angle is turned over zero, the load kidney and the tank kidney switch function.

For the IHT, but also for the conventional transformer type, the law of conservation of power must be valid. For a stationary situation and neglecting internal losses and the power contribution of the tank connection, this means that the following equation must hold:

$$p_0 \cdot Q_0 = p_l \cdot Q_l$$

From this equation the ratio of the load flow and the supply flow can be calculated to:

$$\frac{Q_l}{Q_0} = \frac{p_0}{p_l} = \frac{1}{\Pi}$$

the inverse of the transformation factor $\prod$.

The transformer is also a flow node, for which the law of continuity of mass must hold. This explains the necessity of the third connection to the tank. This connection supplies the make-up flow, which is necessary, because mostly Q_l differs from Q_0 by the factor calculated above.

Figure 8 contains the graphic representation of these pressure and flow relationships. It presents the fundamentals of hydraulic transformation.

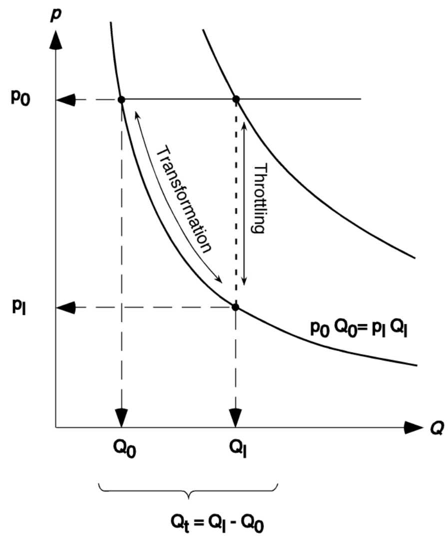


Figure 8: The fundamentals of hydraulic transformation.

ADVANTAGES OF THE IHT

Compared to the conventional transformer, the IHT's advantages are:

- The IHT consists of only one axial piston unit. It is small and light weighted. It should be noted here that the IHT concept is not limited to axial piston units. It can be applied to any positive displacement unit that uses a commutating element to separate its pressure connections. This means that for instance radial piston units and geroller type units can also be converted to hydraulic transformers based on the IHT principle.
- The IHT can be controlled by rotating the port plate over a relatively small angle. Only the friction torque between port plate and housing resists this rotation. With this torque and the maximum angular stroke of the port plate, a typical actuator control energy can

be calculated. In the same manner, the typical actuator control energy can be calculated for other types of variable units. Table 1 [10] lists the results of this calculation for (from top to bottom) a bent-axis axial piston unit, a swash-plate axial piston unit, a radial piston unit and for an IHT. The control energy of the IHT is much lower than that of the other units. A conventional transformer requires the same control energy as the variable unit it is based upon.

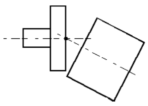
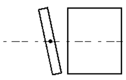
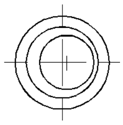
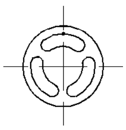
variable unit type	control displacement	control force	control energy
	0.040 m	2200 N	88 Nm
	0.020 m	510 N	10 Nm
	0.007 m	2000 N	14 Nm
	1 radian	< 5 Nm	< 5 Nm

Table 1: Theoretical control energy for different hydrostatic units [10].

- Calculations have shown that the power required to operate the IHT in a typical secondary control scheme is in the 30 – 50 W region for a transformer output power of 30 kW. This is an order of a magnitude lower than what a conventional variable unit would require. It should therefore be possible to use an electromagnetic actuator to control the unit.
- The inertia of the rotating parts in the IHT is very low, at maximum half the inertia of the rotating parts in a conventional transformer. Consequently, the IHT has an inherent capacity for a quick response to changes in the load or in the actuator input. This dynamic capacity provides even more room to realize a quick and stable secondary control.
- Because no real part load conditions occur, the efficiency is much higher than that of a conventional transformer.

IHT DESIGN DETAILS

The basic principle of the IHT is simple. It was invented and patented in 1996. As always, when developing an idea as simple as this to ripeness, the inevitable number of design issues emerged and had to be resolved.

VALVING LAND PHENOMENA

In a conventional axial pump or motor unit, the changeovers of the cylinder bores from one kidney to another, occur in the TDC and the BDC of the plunger movement, at zero flow delivery.

In the IHT, the changeovers of the cylinders occur at points were the pistons moving in the bores have a non-zero speed. During their changeover, the plungers are delivering oil to or drawing oil from the kidneys. This can be seen in figure 9, where the flow from one piston is given as a function of the unit's rotation angle φ. The positions of the kidneys are indicated for a certain port plate angle δ.

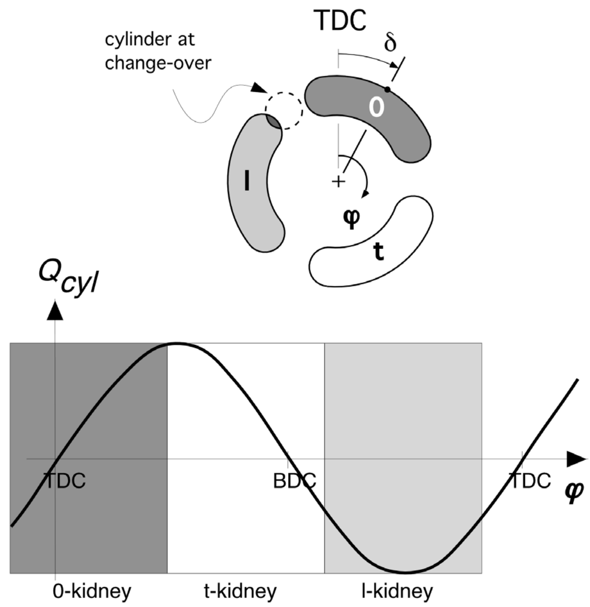


Figure 9: Piston flow as a function of the rotation angle ϕ, kidney positions at port plate angle δ.

Two of the design issues that had to be resolved arise from these cylinder changeovers at non-zero flow conditions. The associated valving land phenomena in the IHT have been described in detail in [11]. Here, only a brief summary is given.

Throttling losses

In the IHT, the decreasing opening area between cylinder bore and kidney at changeover conditions resulted in severe throttling losses. In a normal axial unit design, this opening is determined by the circular or elliptical edges of the kidneys and bores (see the piston at changeover in figure 9). This edge form results in a gradual change of the flow area between zero and the maximum area. When changeover takes place in or near the dead center positions, the flow through these decreased areas is very low or zero. When, as in the IHT, the changeover can occur at non-zero flow conditions, flows losses increase considerably, especially at high unit speeds.

A solution was found in the so-called concurrent port design, in which the edges of the bores are made congruent to the edges of the kidneys. In this way the flow passage is opened and closed much faster and flow losses are reduced to acceptable proportions. In the concurrent port design, the principle of concurrent edges is important. The exact form of the bore and kidney edges does not matter, as long as they are concurrent.

As can be seen in figure 1, for the concurrent ports of the 5-cc IHT prototype, square edges were chosen.

A 45-cc prototype unit with square concurrent ports was tested for its efficiency at the Linköping University of Technology. The results have been reported in [12] from which figure 10 has been taken. The importance of the concurrent edge design stands out.

These efficiencies have been realized with a prototype IHT that was produced by modifying only port plate and end cap of a 45-cc axial piston unit with seven pistons. Predicted and measured efficiencies show a close match. Calculations show that a nine-piston IHT design with three kidneys of equal arc length is optimal. The figure shows the calculated efficiency for such a design. The target efficiency also indicated in the figure, applies to an IHT designed from scratch. In that case, also the drive section and bearings can be designed for use in an IHT, which will bring an additional gain in efficiency.

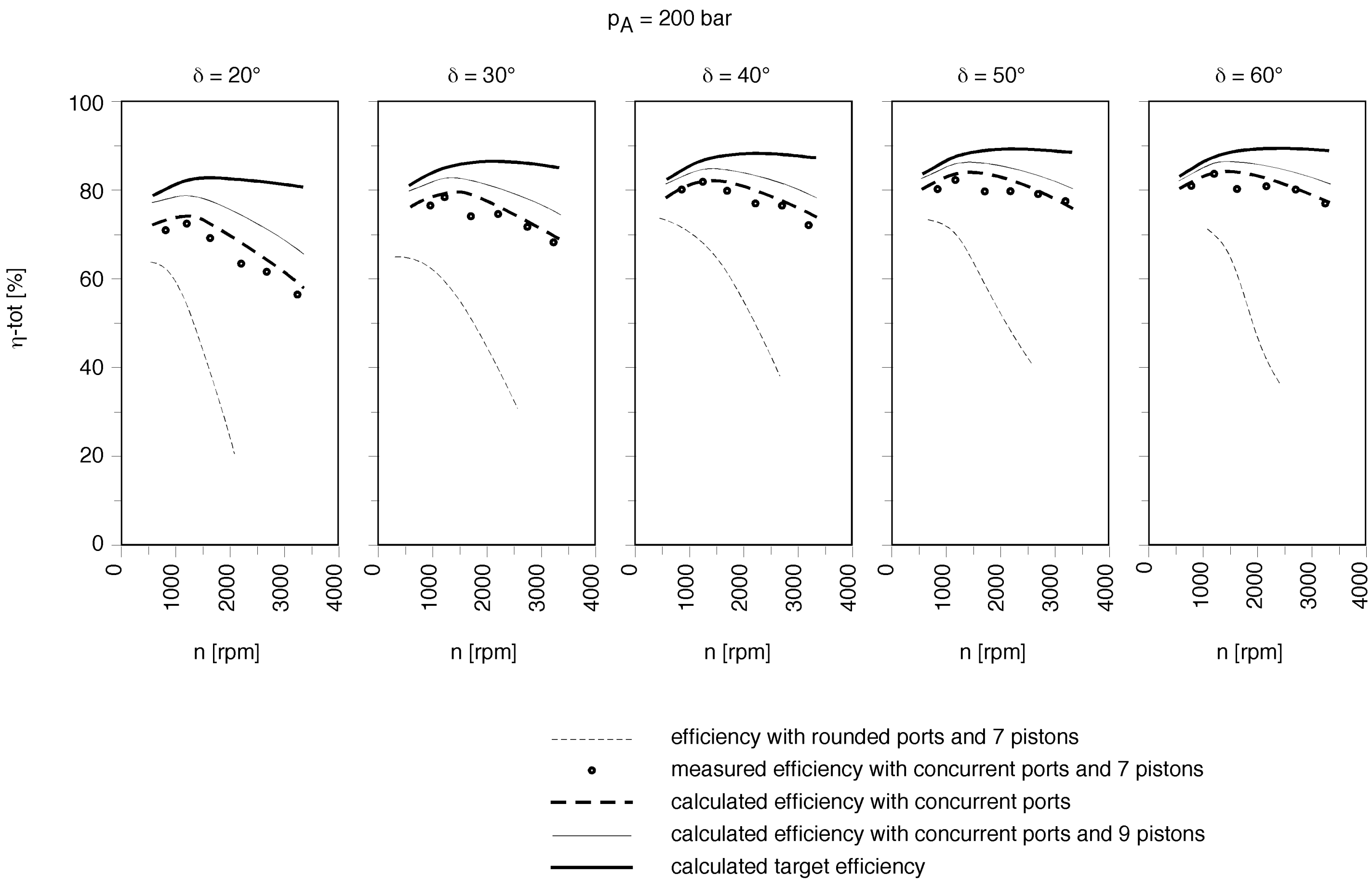


Figure 10: The efficiencies of IHT designs with and without concurrent ports [12].

Pressure peaks and cavitation

When the cylinders pass from one kidney to the other, the oil volume in the cylinder is briefly shut off by the valving land. Because the kidney changeover occurs outside the dead center positions, the cylinders experience a change of volume during the time that they are shut off. This implies that, depending on the position of the changeover, the oil in the cylinder volumes will pre-compress or pre-expand according to the well-known compressibility law:

$$\partial p = -K \cdot \frac{\partial V}{V_0}$$

Several companies developing axial piston units have experimented with rotatabe valve plate designs. Their goal was to create a new type of variable displacement axial piston unit. The consensus resulting from these experiments was that this technique leads to unacceptable pressure peaks and serious risks for cavitation.

For conventional, flow-imposed systems, this statement is true. Because the load pressure is a result of the imposed load speed and the load process, it can by no means be coupled to the operational circumstances of the axial unit alone. This implies that the pressure drop or rise between the kidneys cannot be predicted. The pressure changes that occur in the trapped cylinder volumes due to pre-compression or pre-expansion, on the other hand, are completely dependent on the unit's geometry and speed. Under most circumstances, the two effects will not match, and this leads to the pressure peaks mentioned. Thus, the rotatable port plate is not very suited for use in flow imposed systems.

In the IHT, the situation is different for two reasons:

1. The IHT displays a fundamental relationship between all kidney pressures and the port plate position. The supply and tank pressure are fairly constant and if they are known, the stationary load pressure can be predicted as a function of the port plate control angle (figure 7). As it happens, in the IHT this relationship coincides qualitatively with the compression and expansion curves for the cylinder volumes at all valving lands and for all port plate positions.
2. In a CPR system, the supply and tank kidneys of the IHT are connected to the high- and low-pressure rail respectively. As accumulators are attached to these rails, these connections are essentially 'soft'.

The first effect presents the possibility of optimizing the pre-expansion and pre-compression in the lands surrounding the load kidney for all values of the control angle δ. Because of the soft connections surrounding the land between supply and tank kidney, the pre-expansion or pre-compression in this land may partly be sacrificed to a better match in the other two.

In figure 11, the pressure in a cylinder is given as a function of its angular position. It was calculated for a unit in which the dead volume in the plungers was also optimized. It should be noted that apart from the geometrical overlap, a certain speed dependant overlap results, as the pressure in each plunger volume will start to change already when the flow area is getting smaller. Even in the concurrent port designs a small but noticeable effect remains. Therefore, the pressure match is not perfect for all rotational speeds. However, calculations show that the unit can be tuned to give a good match over the total speed range.

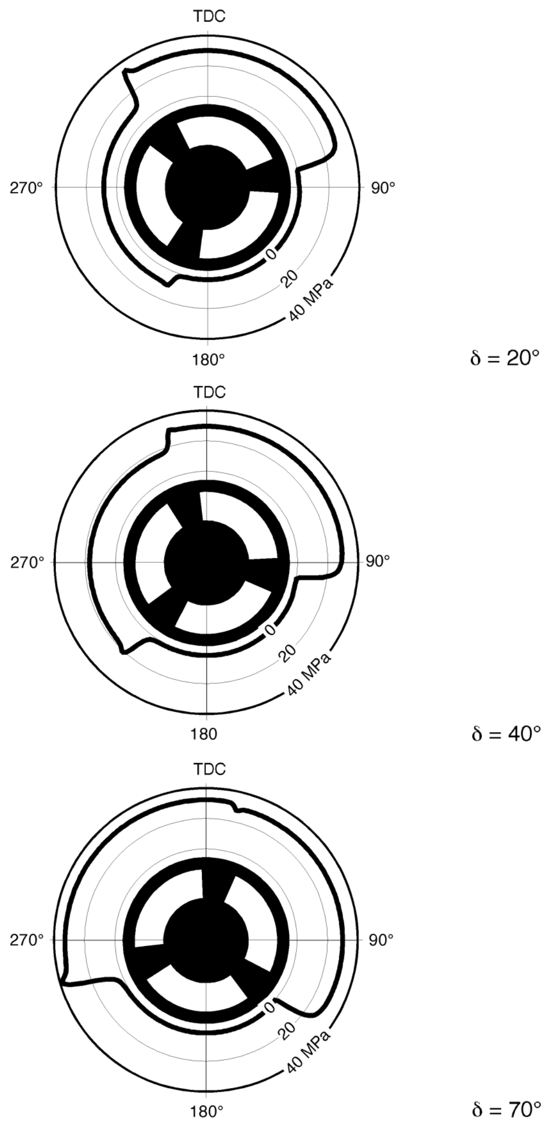


Figure 11: Calculated pressure in a single IHT cylinder, over one revolution of the barrel, at three different port plate control angles [11] (IHT: nine pistons, 45 cc/rev, 3000 rpm).

FOUR QUADRANT AMPLIFYING IHT DESIGN.

As has been explained in [13], using an IHT to supply pressure to two parallel, constant displacement hydrostatic hub motors, is the best choice for the wheel drive of a fork lift truck with a CPR system. Costs are lower, controllability is better and packaging is easier, compared to the option of a variable displacement motor coupled to a drive axle with a differential.

An IHT based wheel drive system was realized in a prototype forklift truck. The prototype contains a second IHT for the lift function and is powered by the 'Chiron' free piston engine described in [14]. The lay out of the wheel drive system is presented in figure 12.

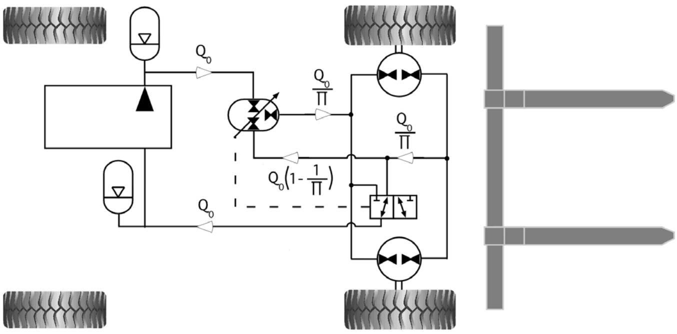


Figure 12: Lay out of an IHT based wheel drive for a lift truck

In figure 12, the flows in the connecting lines are indicated. As has been explained in figure 8, the flow out of the wheel motors differs from the required make up flow at the IHT's tank connection. The difference between both flows has to flow from or to the system's low-pressure line.

In this forklift truck application, the energy consumption can be reduced by more than 20% if the energy recuperation capabilities of the CPR technology are put to use. In that case, the wheel drive system has to operate in all four quadrants of the drive torque vs. drive speed diagram: it must be possible to reverse both the direction of rotation and the direction of the drive torque. The drive torque can be reversed by turning the port plate in the negative direction. As has been mentioned before, this causes the load and tank kidneys to change function, resulting in the load pressure being applied to the other side of the wheel motors. In that case, the make-up connection to the tank has to be switched to the other side of the wheelmotors. This is the function of the 3/2 valve in figure 12.

Because the maximum operational pressure of many standard hydraulic components is 30 MPa, a CPR rail pressure of approximately this value is a logical choice. For some applications, however, higher pressures should be possible. For the wheel drive of this forklift truck, a pressure of 43 MPa had to be realized. Here, the amplification capacity of the IHT was put to good use.

As can be seen in figure 7b, a pressure amplification capability of 1.5 implies that a normal port plate with three kidneys, each spanning a 120 degrees angle, has to be rotatable over approximately 80 degrees. To make-up for efficiency losses an extra 10 degrees are required. In combination with the required reversal of the direction of the drive torque, this means that it must be possible to rotate the port plate over 180 degrees. Each kidney requires a connection to a hole in the end cap and has to be able to rotate over this 180 degrees without connecting to the other two holes in the IHT's end cap. This would require 540 degrees of rotational freedom to be realized within the available 360 degrees.

The solution was found in the special design of the back of the port plate, shown in figure 13.

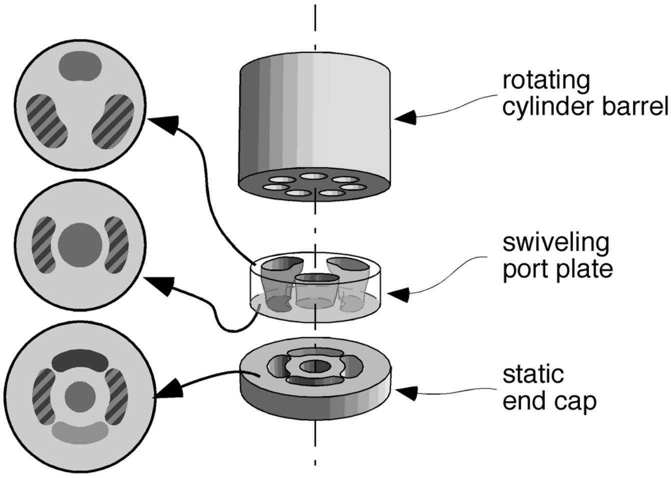


Figure 13: The four-quadrant port plate design.

In this design, the supply kidney is routed from its normal position at the front side of the port plate, to the middle of the port plate at the backside. In this way, room is created to put the other two kidney's opposite of each other at the back of the port plate. These kidneys span an arc length of about 90 degrees each.

The static connections in the end cap are split in four kidneys. Each kidneys spans the same 90 degrees. Two of the end cap kidneys are connected permanently to the load and the tank pressure respectively. The other two are switched when the load and tank kidneys have to switch function. As these kidneys have to be switched when the control angle changes sign, the switching function can be integrated in the make-up valve required for four-quadrant operation. This means that the 3/2 valve of figure 12 has to be replaced by a 4/2 valve. It should be noted that a free running position for the wheel drive - or any other load requiring one – can easily be realized by adding an extra position to this 3/2 or 4/2 valve. In this position, the wheel drive circuit can be switch to internal circulation.

Figure 13 illustrates the way the 4/2 valve switches the connections when the δ-angle changes sign. Note that in this way, not only the 540-degree problem is solved while keeping a sufficient flow area under all circumstances, but also the required switching of the low-pressure connection is realized. As the switching of this valve is dependant only of the sign of the port plate swivel angle, the valve may even be designed to be an integrated part of the port plate.

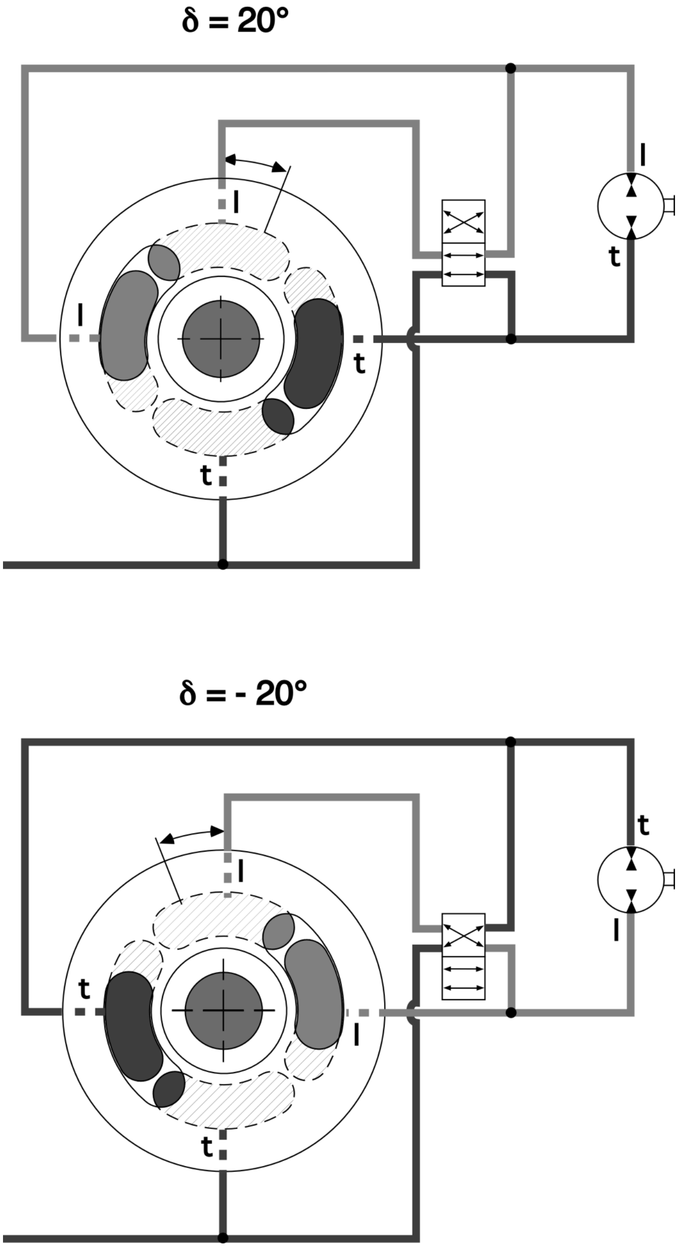


Figure 14: Switching the kidneys in a 4-quadrant IHT design

ADVANTAGES OF IHT BASED CPR SYSTEMS

The previous sections illustrate that the IHT is the preferred way to drive translating as well as rotating loads from a common pressure rail. In the four years following its invention, the IHT has developed to a state of ripeness where an introduction in the market should soon be possible.

The advantages of the CPR approach have partly been treated in the previous sections. Here the advantages on the system level are summarized. To illustrate these, a CPR alternative (figure 15) for the LS hydrostatic circuit of the NOBAS excavator (figure 4) has been worked out.

COSTS

The production costs of an IHT with actuator, built in normal axial unit series sizes, will be in the same range as the corresponding axial unit. Thus, they will be comparable to the production costs of load independent LS sections as used in figure 4. There is no cost penalty in using IHT's in stead of LS sections.

As can be seen by comparing the two figures, in the supply side of the CPR system considerable cost advantages will be reached:

- Because of the possibility to perform peak shaving and energy recuperation and because of the better overall efficiency, the installed engine power can be reduced substantially. In the forklift truck prototype, the engine power could be reduced from 28 kW in the conventional layout to 17 kW in the CPR layout. Engine costs will decrease correspondingly.
- The number of pumps is reduced from two variable and two fixed displacement pumps in the LS layout, to two fixed displacement pumps in the CPR layout. This will reduce the system costs substantially, as variable pumps are a lot more expensive than fixed displacement pumps.

The ideal energy source for a CPR system is the Chiron free piston engine [14]. If the conventional combination of internal combustion engine and pumps is replaced by such a free piston engine, the total system costs will decrease even further.

ENERGY CONSUMPTION

The energy consumption in the CPR layout will be substantially lower than in the LS layout:

- In the IHT based CPR layout, throttling is avoided.
- The efficiency of the IHT is high, especially in part load conditions.
- Energy can be recuperated to the high-pressure accumulator. (For the excavator this is particularly

advantageous when swinging the cabin and load back and forth. Usage of the dissipative swing brake can be avoided.

CONTROLLABILITY

In the CPR layout, there is no interaction between the loads and no interaction between the power source and the loads. When a speed control is necessary, the IHT's are secondary controlled. As is illustrated by the container carrier example, secondary controlled systems display an excellent controllability. In combination with the fast response characteristics of the IHT, this enables very precise control. In the LS system, the speed with which the supply pump can be varied influences the responsiveness of the LS system directly. This implies a slower control even if, at the penalty of increased costs, faster pump actuators are chosen.

The concept of secondary control matches with the trend towards bus systems in vehicle control. The fast control algorithms can be implemented in local controllers, who receive their set points and targets from a central master controller, over a bus system. This type of control set-up enables easy synchronization between functions, learning of repetitive tasks and adapting the system to changing circumstances or driver preferences.

MODULARITY

The combination of a system bus for the control of vehicle functions and a common pressure 'bus' for their power supply opens up the road to a really modular system design. For the OEM, this means a vehicle hydraulic system can be based on functional units that can be chosen from standardized series. The OEM will only have to determine the master control strategies, without having to bother with low level local control. If the installed power and accumulator capacity are carefully chosen, there is no risk that functions will interfere with each other. Consequently, there is no need to tune the hydraulic circuit. The design lead-time will be shorter and the level of standardization will be higher.

Because of its small size, it should be possible to integrate an IHT in the bottom of a cylinder. A cylinder module can be created that would functionally represent the variable cylinder. If a local secondary control algorithm can be realized, a truly 'intelligent hydraulic cylinder' results.

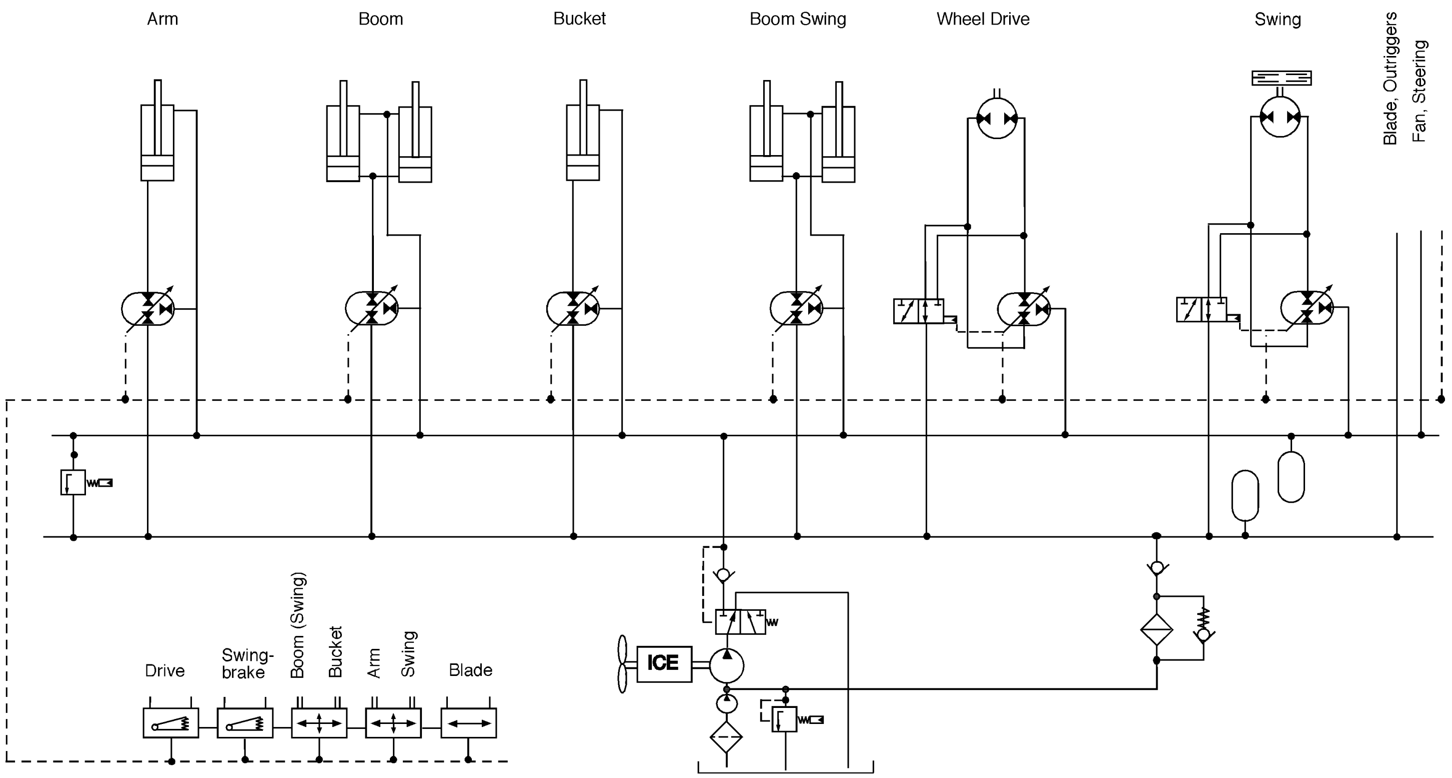


Figure 15: The CPR alternative for the excavator system.

CONCLUSION

With the Innas Hydraulic Transformer, both translating and rotating loads can be driven from a hydrostatic common pressure rail. The IHT development has reached a state where market introduction in the near future is possible. Thus, the IHT is indeed the key to the hydrostatic CPR system.

The CPR system type is superior to load sensing systems in terms of costs, energy consumption and functionality. The common pressure rail design is also beneficial to the OEM's design process as it will reduce the design lead-time and increase the level of standardization.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

IHT:
Innas Hydraulic Transformer

LS:
Load Sensing

CPR:
Common Pressure Rail

ICE:
Internal Combustion Engine

APPENDIX

THE OPERATING PRINCIPLE OF A CONVENTIONAL HYDRAULIC TRANSFORMER

The operating principle of the conventional hydraulic transformer can be explained using the hydraulic schematic of figure A1.

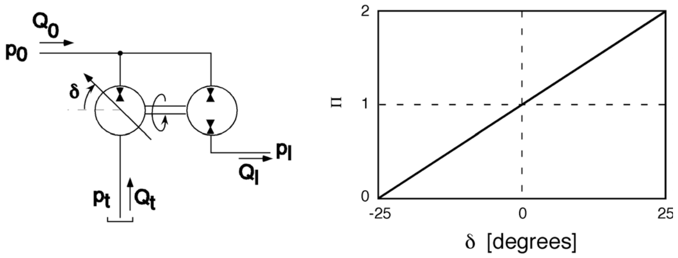

Figure A1: The conventional hydraulic transformer

For the sake of the explanation, all losses in the piston units are neglected.

Then, the torque introduced by the variable unit, in the direction of positive rotation is:

$$T_v = \frac{V_v}{2\cdot\pi}\cdot\left(p_0 - p_t\right) = \frac{V_{v,\max}}{2\cdot\pi}\cdot\frac{\delta}{\delta_{\max}}\cdot\left(p_0 - p_t\right)$$

In which:

V_v displacement of the variable unit

$V_{v,\max}$ maximum displacement of the variable unit

δ control angle of the variable unit

$\delta_{\max}$ maximum control angle

p_0 supply pressure

p_t tank pressure

The torque introduced by the fixed unit is:

$$T_f = -\frac{V_f}{2\cdot\pi}\cdot\left(p_0 - p_l\right)$$

In which:

V_f displacement of the fixed unit

P_l load pressure

The negative sign shows that this unit exerts a torque against the direction of positive rotation.

If the resultant torque is not zero, the total unit will accelerate or decelerate, according to:

In which:

J inertia of the rotary parts

The unit will be stationary only if:

$$T_v = T_f$$

Using this and putting the tank pressure (p_t) to zero, a relationship between control input y and the transformation factor Π can be determined:

$$\Pi = \frac{p_l}{p_0} = \left(1 + \frac{V_{v,\max}}{V_f}\cdot\frac{\delta}{\delta_{\max}}\right)$$

By varying δ, the transformation factor can be varied. The curve in figure A1 has been determined assuming the two displacements to be equal and the maximum control angle to be 25 degrees.

The formula shows that the transformation factor is fully determined by the ratio of the displacements of the two units.

It is important to realize that whenever the actual pressure ratio differs from the geometrically defined transformation factor (Π), the torque contributions from the two units are not in equilibrium. Consequently, the combination will accelerate or decelerate and the oil flow out of the transformer will deviate from the flow into the load cylinder or motor it is connected to. Therefore, the oil column between transformer and load will compress or expand and the pressure in that column will return to the equilibrium value, given by the transformation factor Π.

Two other factors have not been taken into account for this simplified explanation:

1. Mostly the tank connection is not at zero pressure but at an elevated pressure. This will expand the formula for the transformation factor with a small term that is dependent of the displacement ratio and of the ratio of tank and supply pressure.
2. When the transformer changes speed, also the flows at the supply and the tank kidney will change. In spite of the accumulators coupled to these

connections and in spite of the fact that the flow source is controlled to keep the rail pressure to a predetermined value, the pressures at these connections will vary slightly.

For the understanding of transformation fundamentals, however, these terms are not important.

THE OPERATING PRINCIPLE OF THE INNAS HYDRAULIC TRANSFORMER.

In the IHT, the pumping and motoring parts of the conventional transformer have been combined in one axial piston unit. In order to realize this, the fixed port plate with two kidneys of a standard axial piston unit, has been changed for a port plate with three kidneys, that can be swivelled over a limited angle. The kidneys connect to supply, tank and load pressure respectively. The construction of the rotatable port plate can be seen in figure 7. The schematic for this unit is contained in figure A2.

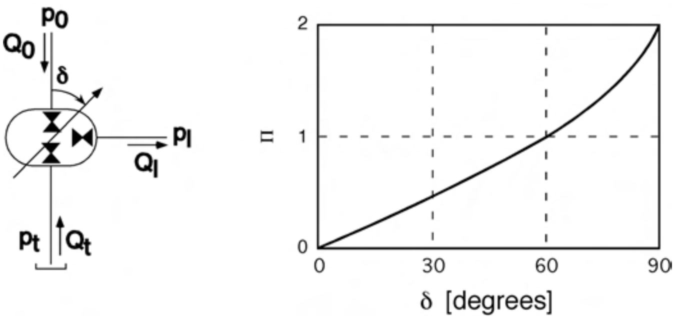

Figure A2: The Innas hydraulic transformer

When the barrel rotates, the cylinders pass the three kidneys sequentially. Depending on the pressure in each kidney and the cylinder position relative to the top dead center (TDC) of the plunger movement, the plungers contribute to the torque on the barrel.

Assuming a constant rotational speed, the average contribution of each kidney to the torque can be expressed by the general equation:

$$T_{k,av} = \frac{p_k \cdot V_{IHT}}{2 \cdot \pi} \cdot \left(\sin \frac{\varphi_k}{2} \cdot \sin \varphi_c \right)$$

In which:

V_{IHT} displacement of the base unit

P_k pressure in the kidney

φ_k nominal arc length of the kidney

φ_c position of the kidney relative to the TDC of plunger movement.

The plausibility of this formula can be illustrated by applying it to the two kidneys in a conventional axial piston unit. The angles defining the port plate and kidney positions are given in figure A3.

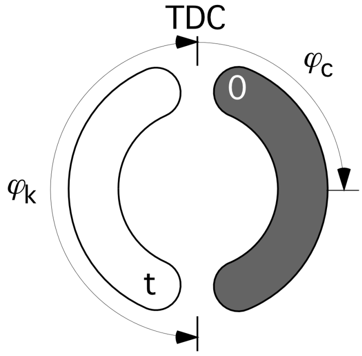

Figure A3: Port plate geometry of a standard axial piston unit.

Calculating the torque contributions of the two kidneys and adding them yields ($\varphi_k=\pi$, $\varphi_c=\pi/2$ and $\varphi_c=3\pi/2$ respectively):

$$T_{av} = \frac{p_0 \cdot V_{IHT}}{2 \cdot \pi} - \frac{p_t \cdot V_{IHT}}{2 \cdot \pi} = \frac{\Delta p \cdot V_{IHT}}{2 \cdot \pi}$$

This is the familiar formula for the motor torque.

For the port plate of an IHT, the angles defining position and geometry of the three kidneys, can be seen in figure A4. The torque contribution of each kidney can be calculated using these angles in the general formula.

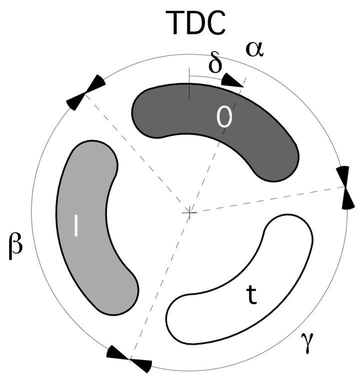

Figure A4: Port plate geometry of an IHT.

In equivalence to the conventional transformer described above, the IHT will move at stationary speed when the sum of the three kidney torque contributions is zero. Adding the three contributions and equating the sum to zero gives:

$$p_0 \cdot \sin\frac{\alpha}{2} \cdot \sin\delta - p_t \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right)$$

$$-p_l \cdot \sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right) = 0$$

This leads to the pressure transformation factor

$$\Pi = \frac{p_l}{p_0} = \frac{-\sin\frac{\alpha}{2} \cdot \sin\delta - \frac{p_t}{p_0} \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right)}{\sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right)}$$

This relationship has been drawn in figure A2, for a design with three kidneys spanning an equal arc-lent and assuming the tank pressure to be zero.

Just as in the conventional transformer type, if the actual pressure ratio differs from the transformation factor, the unit will accelerate or decelerate until the theoretical transformation factor has been reached again.

SUMMARY

Any hydraulic transformer will tend to keep the pressure ratio to the value defined by the ratio of displacements of the pumping and the motoring part. Hence, the output pressure can be set by varying this ratio.

The speed of the transformer is determined by the speed the load reaches at this load pressure.