Transforming future hydraulics: a new design of a hydraulic transformer.

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Abstract

Constant pressure or secondary controlled systems offer a number of advantages over current displacement controlled systems. Especially the energy usage in secondary controlled systems is superior to conventional systems. For implement systems there is however not an acceptable solution to convert the more or less constant pressure of the supply rail to the variable pressure and flow needed in the hydraulic cylinders. The classical 'solution' of a variable displacement motor connected to a constant displacement hydraulic pump is complex, bulky, heavy and expensive. The average conversion efficiency of the total package is in most applications not much better than of systems in which the pressure difference is completely throttled.

This paper describes a new concept of a hydraulic transformer, the Innas Hydraulic Transformer or IHT, called after the Dutch engineering company Innas that developed this concept. The basic structure of the IHT is a constant displacement hydraulic motor or pump with port control, such as an axial pump with a port plate. The design of the IHT however differs from the pump design in that it has a port plate with three ports in stead of two. Furthermore the angular position of the port plate can be changed relative to the top dead centre position of the pistons. The control angle of the port plate determines the pressure or transformation ratio as well as the output flow of the transformer.

A first prototype of the transformer has been built and tested in a fork lift truck application for lifting the fork. Also efficiency calculations have been performed and the dynamic response characteristics have been investigated. Furthermore the combination of the IHT with the Innas Free Piston engine has been analysed. The results of these analyses are described in this paper.

Keywords: Hydraulic Transformer, Design Description, Controllability, Efficiency, Secondary Controlled System

1 Nomenclature

A	area	[m^2]
d	damping coefficient	[N m s] or [N s/m]
E	oil bulk modulus	[Pa]
F	Force	[N]
J	inertia	[kg m^2]
m	mass	[kg]
M	torque	[Nm]
n	rotational speed	[1/s]
p	pressure	[Pa] or [bar]
P	power	[kW]
Q	flow	[m^3/s]
v	velocity	[m/s]
x	distance	[m]
δ	control angle	[°]
ε	angle	[°]
Π	amplifier ratio	[-]
ω	rotational speed	[rad/s]

Subscripts

A	pressure rail side
B	load side
T	low pressure side
L, IHT	leakage losses IHT
L, p	leakage losses pump
L, m	leakage losses motor
bal	balance
max	maximum
c	cylinder
b	cylinder barrel
ref	reference

2 Introduction

Most hydraulic systems are primary, displacement controlled systems. In hydrostatic drives the motor torque and speed are predominantly controlled by means of the control angle of the variable displacement pump and the governed speed of the engine driving the pump. Implement systems with hydraulic cylinders are throttle or load-sensing controlled.

Instead of displacement controlled systems, pressure determined systems could be applied. Hydrostatic motors and cylinders can then be connected to a pressure rail in which the pressure level is kept at a constant level, independent of the loads of the subsequent users connected to the rail. The load control is realised on the secondary side. In order to do this with a high efficiency, the machines on the secondary side have to be variable. For hydrostatic drives solutions have been found and analysed [1 to 8]. These so called secondary controlled systems are characterised by a relatively high efficiency and good controllability. Also energy recuperation is possible.

For implement systems a valve could be used to vary the load on the cylinder. However the efficiency of such a system can be rather low since the pressure difference between the constant pressure rail and the cylinder is completely throttled. As an alternative a hydraulic transformer has been suggested by different researchers [8, 9, 10]. The transformer consists of two hydrostatic machines of which at least one has a variable displacement (see figure 1). The efficiency improvement of this solution is however small compared to throttle control. Furthermore these types of hydraulic transformers are complex, bulky and expensive. It is also questionable whether these transformers can fulfil the dynamic response requirements needed for implement systems.

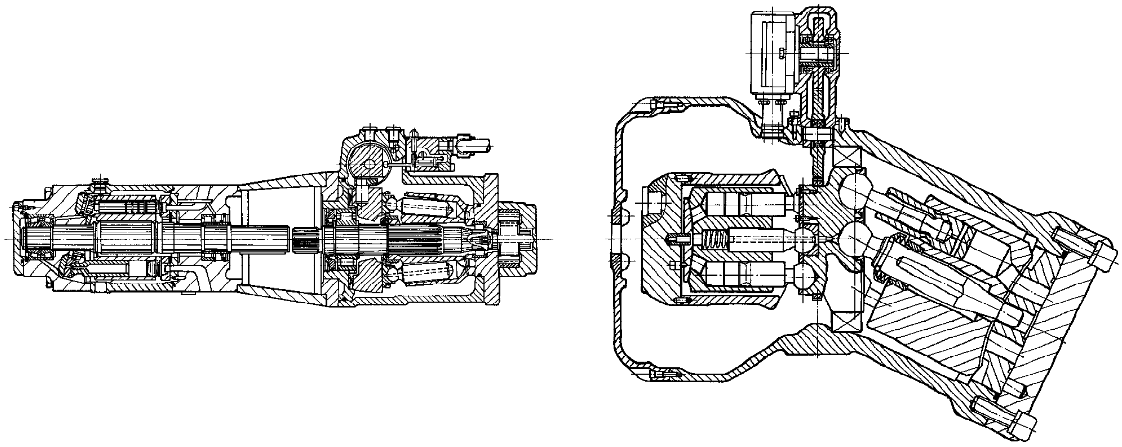


Figure 1. Classical hydraulic transformers [8]

In this article, a new concept of a hydraulic transformer is introduced: the Innas Hydraulic Transformer (IHT) called after the Dutch engineering company Innas that developed this concept. The new design is essentially based on a constant displacement hydraulic motor or pump. The key element is a port plate which has three kidneys instead of two and which can be rotated around the central axis. Next to a description of the working principle the efficiency and dynamic response capabilities are described. Although most of the analysis is done on the basis of computer simulations some preliminary tests have been performed. The analysis and tests have proved the feasibility of the concept.

The article will also include a discussion on the application potential of the transformer. Part of this discussion refers to the diesel-hydraulic free piston engine that has been developed by the same company and about which has been reported earlier [11 to 15]. The article will end with a preview on planned developments.

3 Working principle

Most hydraulic machines –whether pumps or motors– are of the displacement type in which a number of volumes subsequently expand and contract. The volumes are alternately pressurised and depressurised by means of ports. Pumps and motors generally speaking have two ports: a high pressure and a low pressure port (see figure 3).

The Innas Hydraulic Transformer operates in the same way as current pumps and motors with one elementary difference: instead of two different ports it has three ports. One port is used to supply high pressure oil to the transformer in order to drive the transformer. A second port is connected to the load side of the transformer. The third port is connected to a low pressure line or to an oil reservoir at ambient pressure.

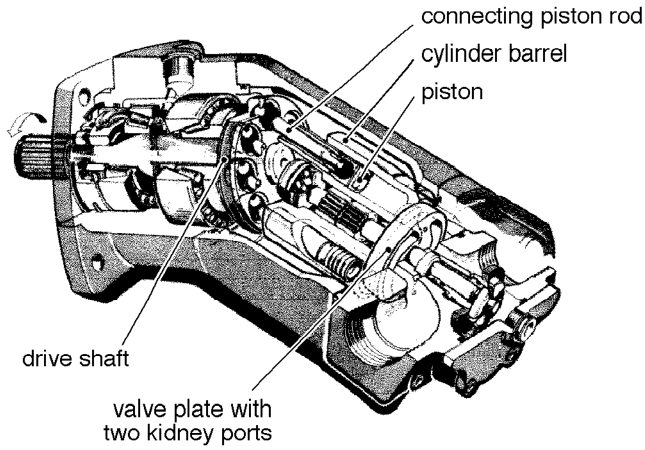


Figure 2. Hydraulic motor with port plate with two kidneys

In order to illustrate the necessity of this third port figure 3 shows the operation of the transformer in the pQ-diagram. Whenever hydraulic energy is transformed from a high pressure to a low pressure, while keeping the power level at constant, the flow has to increase. In order to maintain the flow balance over the transformer, a third flow has to be admitted to the transformer.

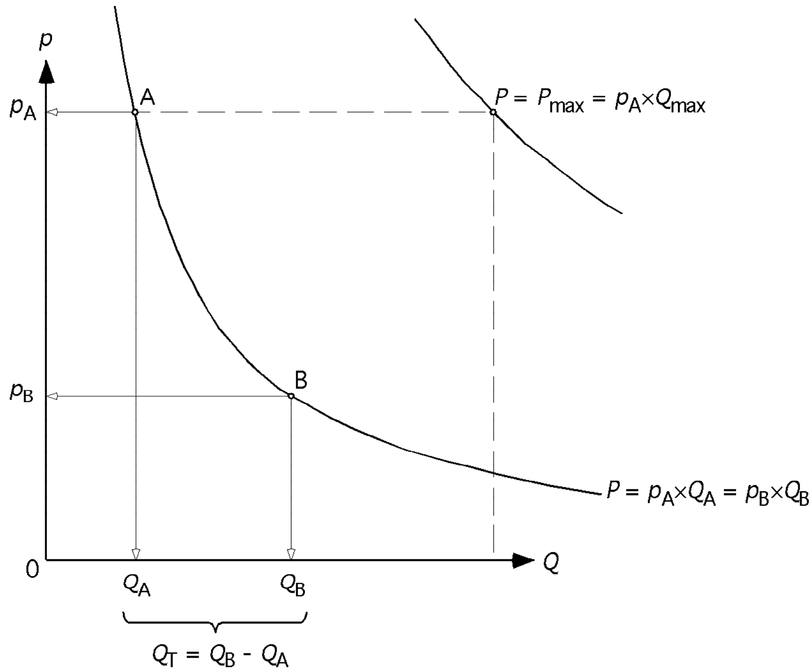
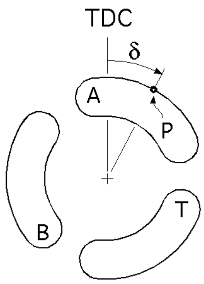


Figure 3. Power and flow balance of the hydraulic transformer ($p_T = 0$)

This paper will explain the operation of the IHT further on the basis of an axial piston design (although the IHT is not limited to this design principle). Figure 4 shows the port plate which, of course, for the transformer has three kidneys:

- the A-port is connected to the pressure rail that drives the transformer;
- the B-port is connected to the load side of the transformer;
- the T-port is connected to the low pressure side or tank.



A:	connected to the pressure rail (supply side)
B:	connected to the load side of the transformer
T:	connected to the low pressure side
P:	reference point on the port plate
TDC:	Top Dead Centre position of the pistons
δ:	Control angle of the transformer

Figure 4. Port plate of an axial piston type IHT

The transformation ratio Π (= p_B/p_A) as well as the output flow of the transformer can be varied by changing the angle δ between a reference point P on the port plate and the top dead centre (TDC) position of the plungers in the cylinder barrel. Point P is (arbitrarily) chosen on the middle of the arc of port A.

The rotational speed i.e. the flow of the IHT is dependent on the resultant of all forces that act upon the cylinder barrel of the transformer. Assuming a constant pressure on the A- and T-side of the transformer, for all pressure levels on the B-port a control angle δ_{bal} of the port plate can be found at which all radial forces that act upon the cylinder barrel result in zero torque. In that case the barrel will not rotate and the flow will be zero (see figure 5).

Aside from the pressure forces, friction forces also have an influence on the torque balance of the cylinder barrel. Furthermore the pressure forces are flow dependent since a higher flow will result in a change of pressure levels at the three ports due to flow restrictions in ports and lines. In order to overcome these flow losses the control angle δ has to be increased somewhat (to $\delta_{bal} + \varepsilon$) when changing the point of operation of the transformer from zero flow to a positive flow (see figure 5).

The cylinder barrel of the transformer can rotate in both directions: clockwise as well as counterclockwise. As a result the flow at the B-side of the transformer can be positive as well as negative (positive being the direction at which flow is delivered by the transformer). For a B-flow towards the transformer i.e. a negative B-flow, the control angle δ has to be somewhat less than the balance angle δ_{bal} (in figure 5: $\delta = \delta_{bal} - \varepsilon$). In this mode the transformer can be used to recuperate energy from the load side i.e. B-side of the transformer. The recuperated energy can then be stored in an accumulator.

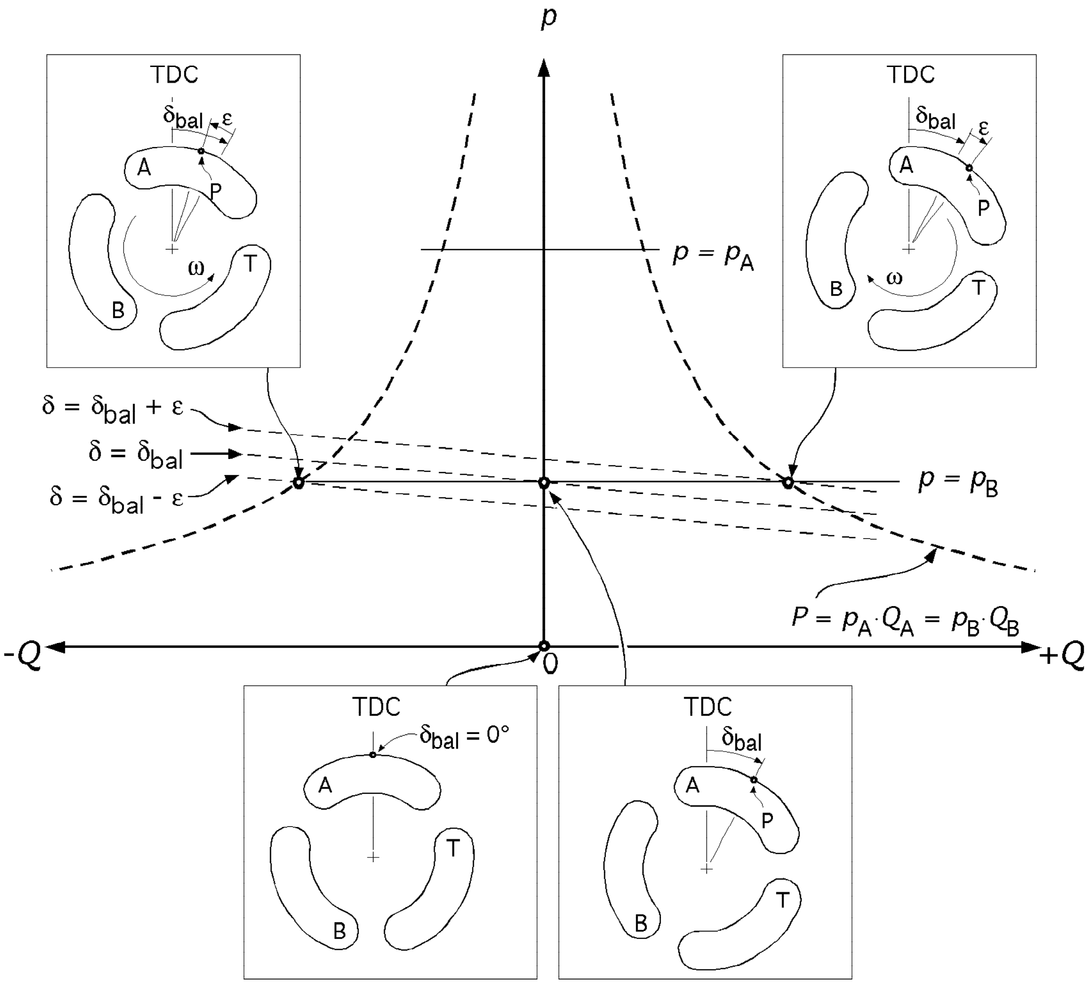


Figure 5. Influence of the control angle δ on the point of operation of the IHT in the pQ-diagram ($p_T = 0$).

The transformer can also work as an amplifier: the pressure at the B-port can be higher than the pressure at the A-port. Referring to figure 5 the port plate has to be rotated to an angle at which the B-port is going 'over-centre'. Thereby the effective arc-length of the B-port is decreased the more the B-port is rotated over the TDC-position of the piston in the cylinder barrel.

This 'over-center' point is also indicated in the next graph (figure 6). In this diagram three curves are shown as a function of the control angle δ:

- the pressure or amplifier ratio of the transformer (p_B/p_A);
- the output flow Q_B of the transformer relative to the maximum flow $Q_{B,max}$ both at a maximum rotational speed n_{max} of the transformer;
- the power P of the transformer relative to the maximum power P_{max}, also both at a maximum rotational speed of the transformer.

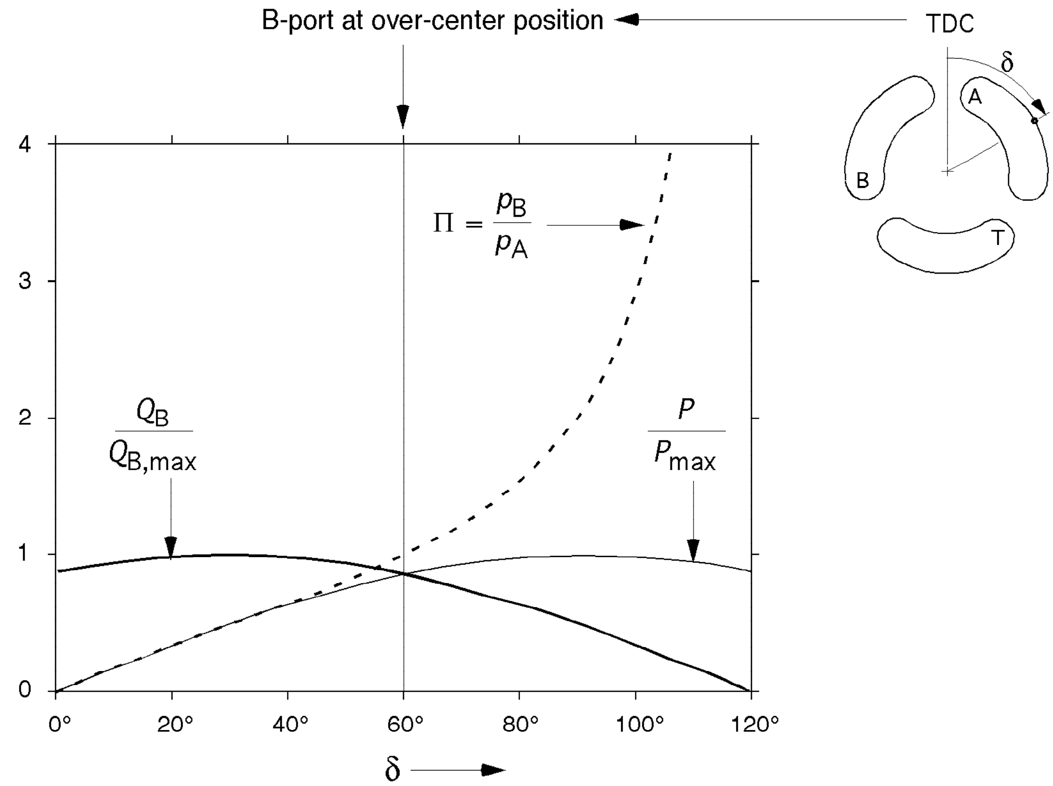


Figure 6. Pressure (p_B), flow (Q_B) and power (P) of the IHT as a function of the control angle δ for a transformer with three equal kidneys (120°) in the port plate. All parameters are made dimensionless. The flow and power curves are defined at the maximum rotational speed of the transformer i.e. the cylinder barrel.

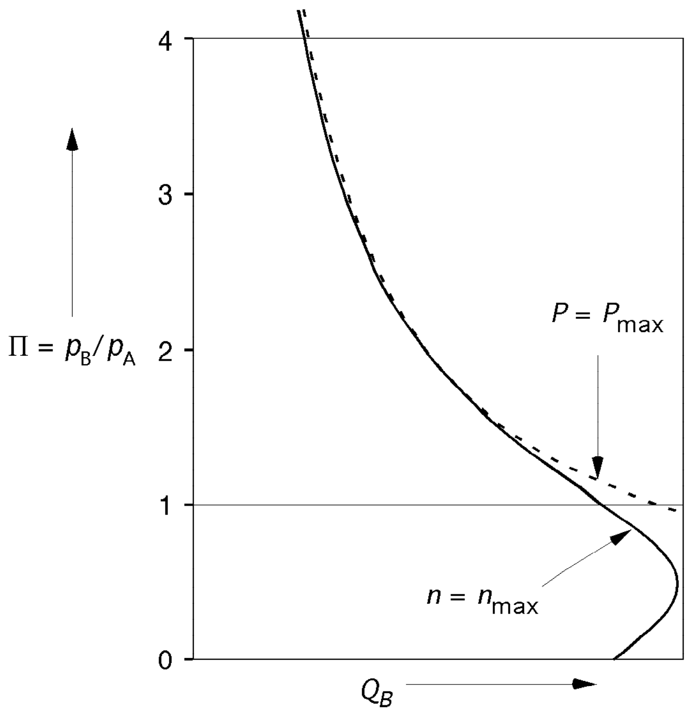


Figure 7. Maximum speed curve of the transformer. On the vertical axis stands the output pressure p_B relative to the supply pressure p_A. On the horizontal axis stands the output flow Q_B.

Figure 7 shows the line of operation of the IHT at maximum speed in the ΠQ_B-diagram. On the vertical axis the pressure amplifier ratio $\Pi = p_B/p_A$ is shown. The graph had been calculated for the same conditions as the graphs shown in figure 6 and is only valid for a port plate configuration with three equal ports with an effective arc length of 120° each. Other port configurations will result in different shapes of the maximum speed line in the pQ-domain.

The graphs from figure 6 shows that at small δ-angles –before the B-port is turned over center– the output flow is approximately constant. The power that is sent through the transformer however increases as the δ-angle is increased since the amplification ratio Π increases. The relationship between Π and δ is about linear. From the point the B-port is turned over the top dead center the amplification ratio increases and the flow decreases. As a result the maximum speed curve follows approximately the constant power curve as can be seen in figure 7. This is different than in current hydraulic motors and pumps where the line of maximum flow corresponds to the line of maximum rotational speed.

4 Testing

A first prototype of the transformer has been built by Innas in 1996 and preliminary functional tests have been performed. The prototype has been constructed on the basis of a Rexroth 40° bent-axis motor (see figure 2). The cover plate and the control plate of this pump have been removed and are replaced by a port plate with three kidneys and a cover plate with three connections (A, B and T). The three kidneys have an equal arc length of 120° including overlap areas between the kidneys. Also the port plate can be rotated by means of an axis connected to the port plate. Figure 8 shows a drawing of the first prototype and the rotating port plate with three kidneys. The port plate of this transformer is manually operated. Since the port plate is completely balanced in the radial plane and is almost completely balanced in the axial direction there is only a low torque needed to rotate the port plate (< 1 Nm). Table 1 shows the most important data of the Rexroth motor.

Table 1: Data of the Rexroth pump used for the first IHT-prototype.

Type:	A2FM10 (40° bent axis)
displacement volume:	10.3 cm^3
max. rotational speed:	8000 rev/min (8800 rev/min intermittend)
max. flow capacity:	427 L/min
max. pressure level:	40 MPa
Inertia rotational parts:	0.0004 kg m^2
weight:	5.4 kg

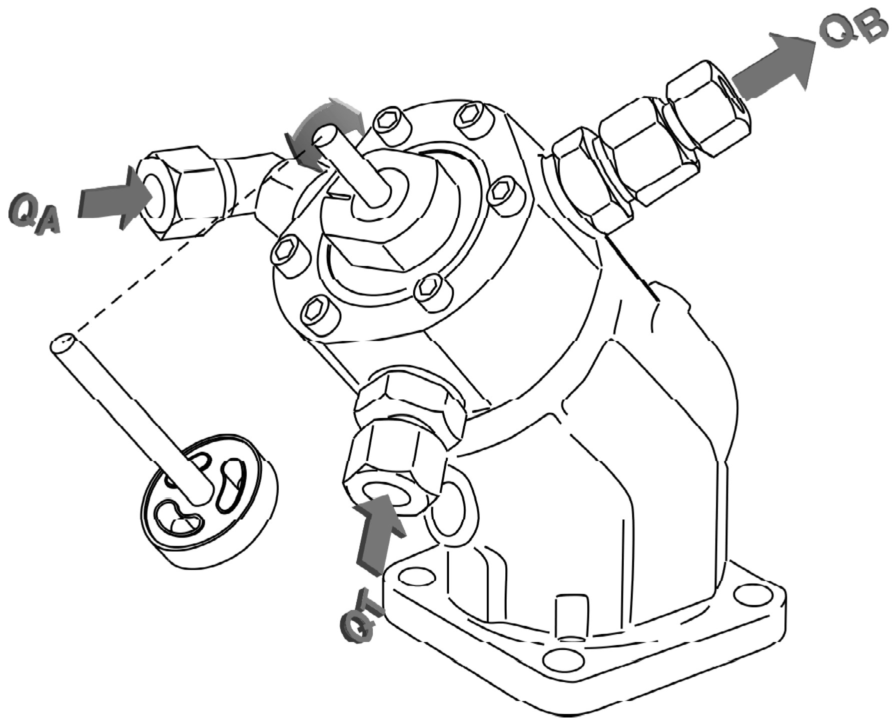


Figure 8. First prototype of the Innas Hydraulic Transformer showing the three oil flows and the rotatable port plate with its three kidneys.

Using a standard pump design as the basis for the first prototype of the IHT was a compromise. It offered a fast route to a prototype since the bearings, the sealing, the plungers, the cylinder barrel and the housing could be used again and did not need to be designed and build: only the port plate and the cover of the pump needed to be replaced. On the other hand the pressure and torque balances of the cylinder barrel and the port plate could not be optimized. Also the number of plungers of the transformer could not be optimized with respect to pressure and flow. And of course the transformer does not need an outgoing axis and the sealing and bearings that are needed for this axis.

The IHT-prototype has been tested in a forklift truck to drive the hydraulic cylinder of the fork. Figure 9 shows the system schematics of the testbed. Since there is yet no generally accepted symbol for the transformer a new symbol had to be created.

Only functional tests have been performed so far. No efficiency or performance measurements have been performed. The tests have proved the principle of the transformer. The torque pulsation's created by the varying number of pistons that are connected to the three ports result in a minimum speed requirement of around 200 rpm. The sound level is low and has a rather low frequency. The tests have also proved that it is possible to rotate the port plate with a very low torque while the transformer is in operation.

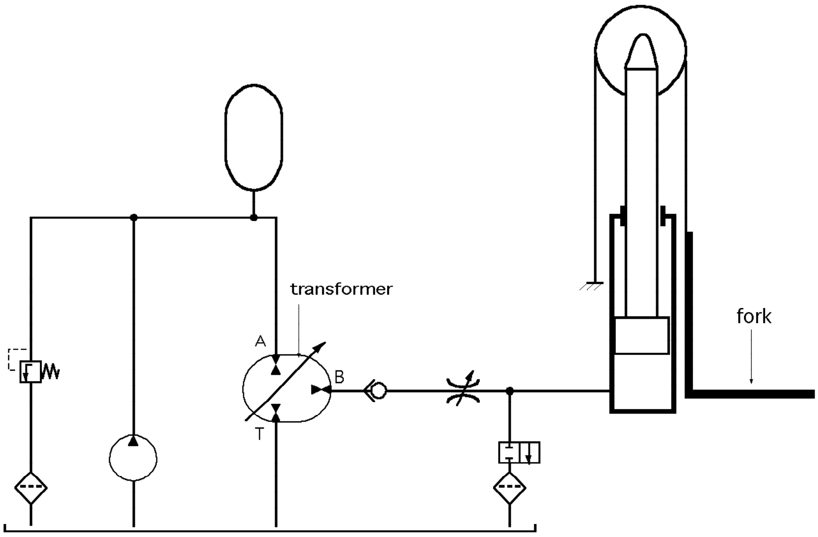


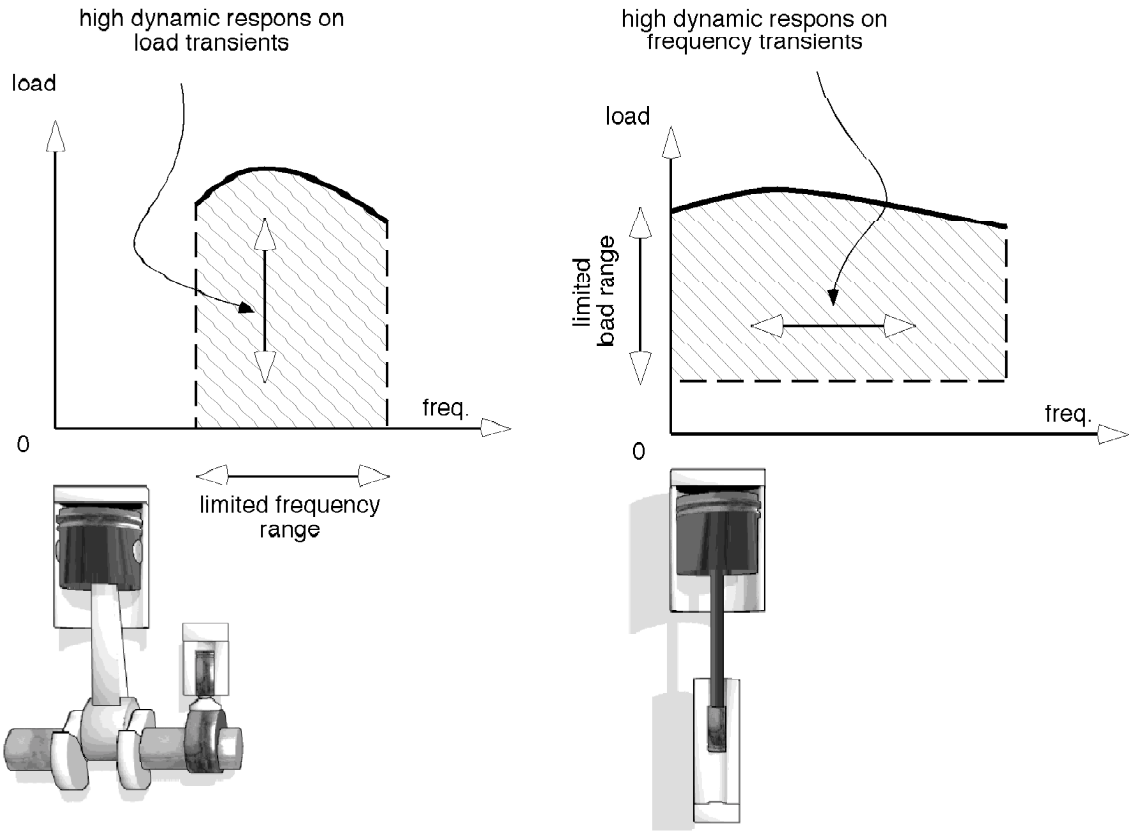
Figure 9. Innas Hydraulic Transformer testbed.

5 Application of the transformer

The transformer has been developed in combination with the ongoing development of the Innas Free Piston Engine of IFP engine. This engine is a combination of an internal combustion engine and a hydraulic pump, having the combustion piston and the hydraulic plunger combined to a single piston-plunger-combination. This ‘piston’ is not connected to any mechanism and is ‘free’ to move in the axial direction.

The IFP engine has characteristics that are quite different from standard engines and pumps. Figure 10 shows the typical torque-speed-diagram of an industrial diesel engine compared to the equivalent pressure-flow-diagram of the IFP engine. As shown the conventional engine has a limited speed range. The dynamic response of this engine is also rather slow in the speed domain but very fast in the load domain. The IFP engine has opposite characteristics: the piston frequency (and with that the flow) can be changed instantaneous and can be varied continuously between zero and maximum. On the other hand the IFP engine has a limited load range. Also the dynamic response is rather slow in the load domain. Both limitations are due to the accumulator the IFP engine needs to damp out the flow pulses.

The limitations of the IFP engine in the load domain are irrelevant when the engine is used as a supply unit for a constant pressure system since the load in such a system has to be kept constant. The high response capabilities in the flow domain than become an essential advantage, since all load and flow variations at the user side of the system are ‘translated’ into flow variations at the supply side by means of variable displacement hydraulic motors and hydraulic transformers.



10a. conventional internal combustion engine   *10b.* Innas Free Piston engine

Figure 10. Comparison of the load-frequency (torque-speed) diagram of an internal combustion engine and the equivalent load-frequency (pressure flow) diagram of the IFP engine.

One of the advantages of 'constant' pressure systems is the capability of recuperating energy that can be stored in accumulators which are connected to the pressure rail. When storing energy the pressure in the hydro-pneumatic accumulator increases. In those cases the IFP engine can stop working and the system is operating entirely on the energy supplied by the accumulators. Only when the pressure drops below a set value the IFP engine starts to work again.

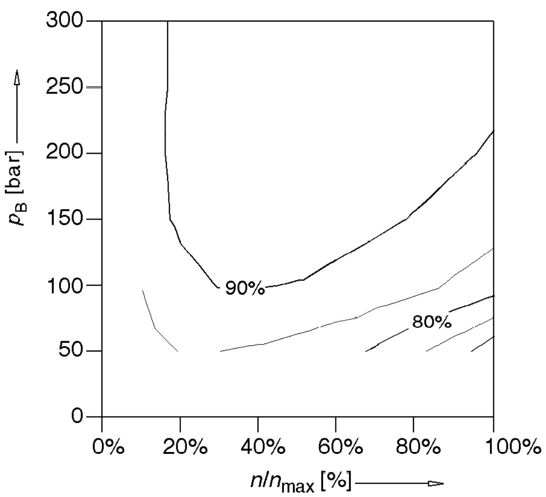
The transformer was originally developed for driving implement systems with a high efficiency in combination with a constant pressure rail. For hydrostatic drives the transformer was not necessary since (secondary controlled) variable displacement hydraulic units were already developed and available. The costs of these hydrostatic units is however more than three times the price of a constant displacement hydraulic unit. The difference even becomes bigger if the gear transmission, which is generally needed for the secondary controlled units is included in the price (constant displacement motors can be mounted directly to the wheel in the hub of the wheel).

When combined with a constant displacement unit, the hydraulic transformer could offer an alternative for the variable displacement motor/pump and gear transmission combination. In that case, the transformer would be the variable transmission for the constant displacement unit. The alternative is however only marketable if it offers sufficient advantages in terms of costs, weight, size, reliability, efficiency, etc.

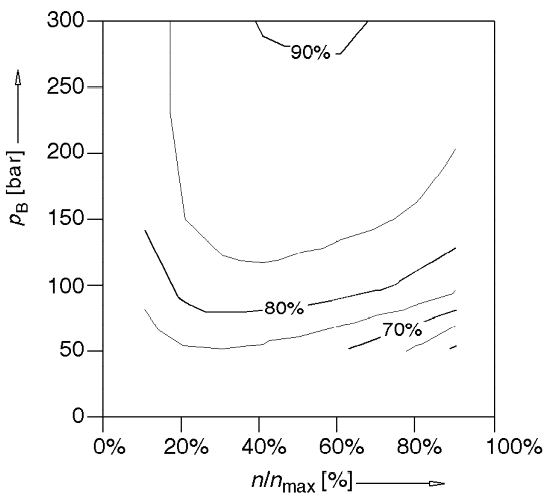
The cost of the transformer can be estimated as being equal to the cost of a constant displacement hydraulic pump or motor, which is equivalent in size and output as the transformer. The actuator needed to control the angle of the port plate will increase the cost. Since the port plate is almost balanced and free floating and the inertia of the port plate is small, only a small actuator is needed to control the port plate angle. Eventually a direct electro mechanical control can be used. The costs of the port plate control are probably more than offset by the cost reduction that is realised because of the elimination of an axis, and the bearings and sealings that belong to the axis.

6 Efficiency calculations

The efficiency analysis of the IHT is performed on the basis of the available efficiency data of the A2F series motors/pumps of Mannesmann Rexroth [16]. The flow losses of the IHT are estimated by scaling the losses of an axial piston motor (equivalent to the motor section of an IHT, i.e. A-port with a pressure of 300 bar) and an axial piston pump (equivalent to the pump section of an IHT, i.e. B-port with the load pressure). The torque losses are estimated on the basis of the losses of an axial piston unit (equivalent to the pump section, i.e. B-port with the load pressure) plus the losses due to the existence of an extra A-port with a pressure of 300 bar. Based on the flow and torque losses analysis, an estimate of the efficiencies of an IHT is performed. The results are shown in figure 11. All calculations are based on an IHT with three equally sized kidneys in the port plate with an effective arc length of 120°. The supply pressure at the A-port of the IHT is 300 bar. As shown the efficiencies of an IHT are somewhat lower than that of its baseline unit. The difference in efficiency ranges from 3% to 13%. The average difference in efficiency is 6%.



11a. Fixed displacement hydraulic motor



11b. Innas hydraulic transformer

Figure 11. Total efficiency of a fixed displacement axial piston unit and of an IHT based on the fixed displacement unit.

The decreased efficiency is due to an increase of the flow losses as well as of the torque losses. Roughly speaking the flow losses due to leakage can be calculated with the following formula:

$$Q_{L,IHT}(p_B) = \tfrac{2}{3}Q_{L,p}(p_B) + \tfrac{2}{3}Q_{L,m}$$

with:

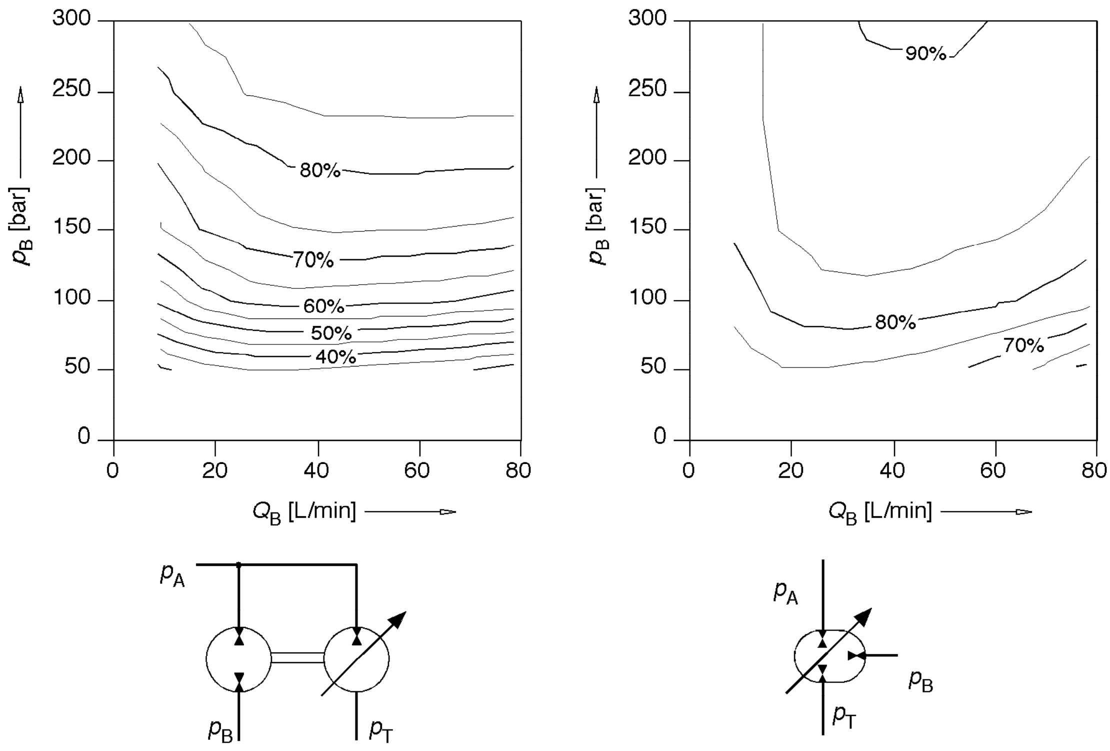
$Q_{L,IHT}(p_B)$	=	flow losses of the IHT due to leakage at a load pressure p_B
$Q_{L,p}(p_B)$	=	flow losses of a hydraulic pump due to leakage at a load pressure p_B
$Q_{L,m}$	=	flow losses of a hydraulic motor due to leakage at a load pressure of 300 bar

The reason for the term 2/3 is that the motor and pump sections of the IHT are pressurised for only 120° of rotation compared to 180° in a motor and a pump. In addition to the leakage losses the flow losses due to compressibility should be taken into account as far as it concerns the flow on the load side (i.e. the B-side) of the IHT.

As in a bent-axis motor or pump, the torque losses of an IHT can be divided in:
- shaft bearing drag,
- drag between the ball joint and the drive plate,
- drag between the piston and the bore,
- drag between the cylinder barrel and the port plate,
- pressure loss through the flow passages in the end cap,
- torque loss due to the transition from one port to another port
- and windage loss.

Similar to the higher losses due to leakage in an IHT compared to the leakage losses of its baseline unit, the torque losses in terms of the shaft bearing drag, the ball joint-drive plate drag, the piston-bore drag and the pressure loss through the flow passages in the end cap are also higher than that of the baseline unit. Under the condition that the cylinder barrel is well balanced, the cylinder barrel-port plate drag in an IHT is at least not greater than that in a baseline unit. The transition torque loss and the windage loss in an IHT are assumed to be the same as that in a baseline unit.

The comparison of the IHT with its baseline bent axis unit is however not very useful if it comes to the application of the transformer since it concerns two different machines. Figure 12 shows an efficiency comparison of the IHT with the classical combination of two hydraulic units. In both graphs the efficiency is shown as a function of the pressure and flow on the load side (B-port) of the transformer.



12a. Conventional hydraulic transformer

12b. Innas hydraulic transformer

Figure 12. Efficiency comparison between the IHT and a conventional hydraulic transformer (p_A = 300 bar)

It is obvious that the IHT-design has a much higher efficiency than the conventional hydraulic transformer, which has always the combined losses of two hydraulic machines of which one has to have a variable displacement. Over the entire field of operation the average efficiency improvement is 13%-points. At low loads the efficiency of the IHT can even be more than 30% higher than that of the traditional transformer design.

Figure 13 shows a direct comparison of the transformer efficiencies with throttle control. The graph is shown for an output flow at the load side of the transformer of 42.5 L/min. It is evident that throttle control is based on dissipating the pressure difference between the supply and the load side, and therefore gives a linear relationship between the efficiency and the Δp between load and supply side. All curves are based on calculations at a supply pressure of 300 bar.

The efficiency comparison shows that at high load pressures, close to the supply pressure, throttle control is better than control by means of a transformer. The transformer has its benefits for situations at low or medium loads. This is especially true for the IHT which has an efficiency that comes close to a load sensing valve [17]. Again the efficiency benefits of the conventional transformer are limited.

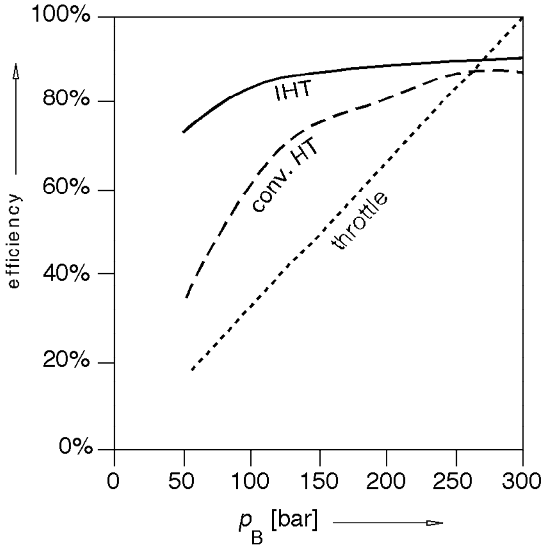
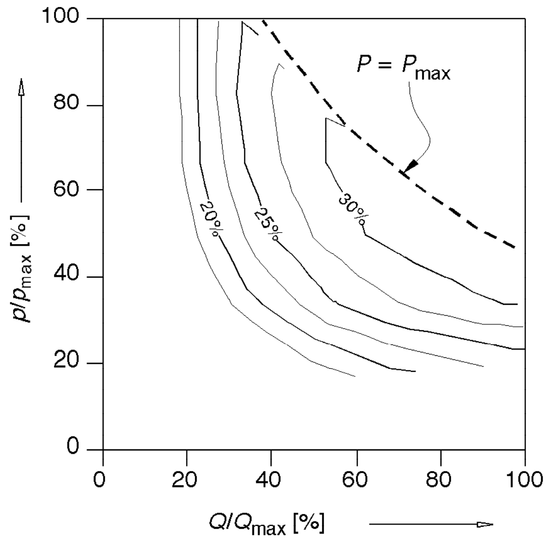
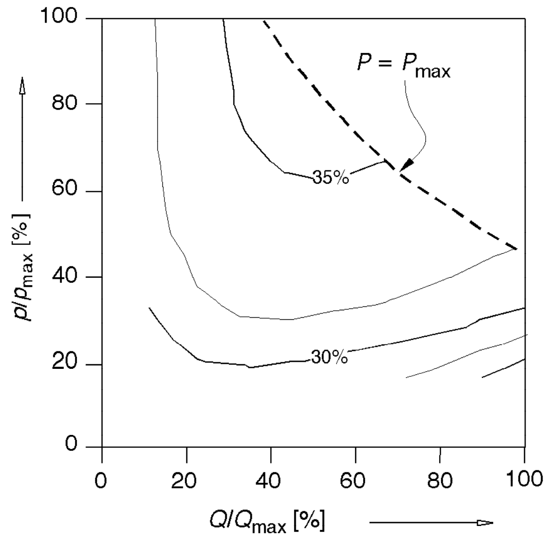


Figure 13. Efficiency comparison between the IHT, the conventional transformer and throttle control at a supply pressure of 300 bar. The efficiency curves for both transformers are calculated for an output flow of 42.5 L/min.

As mentioned before, a special application of the IHT is in combination with the diesel-hydraulic free piston engine which is developed by Innas. Because of the hydraulic energy output of the Innas Free Piston engine, the IFP engine and the IHT are a perfect match. This is also illustrated in the efficiency diagrams of figure 14. The contourplot of figure 14a shows the efficiency map of a classical combination of a modern diesel engine and a variable displacement hydraulic pump. Next to it figure 14b shows the same kind of diagram but now for the combination of an IFP engine and the IHT.



14a. Internal combustion engine + variable displacement hydraulic pump

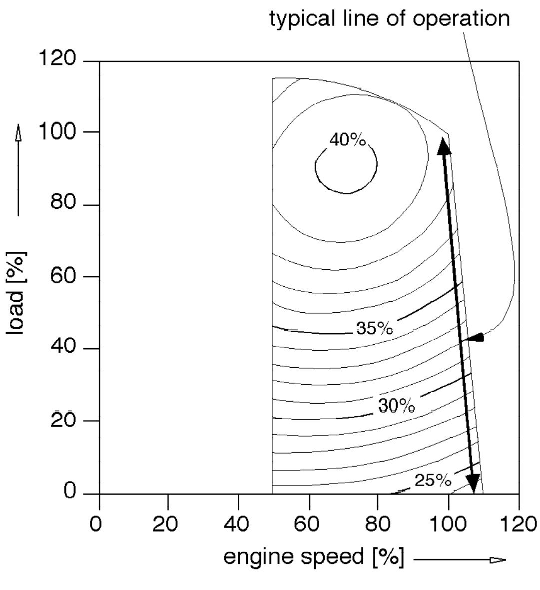


14b. Innas Free Piston engine + IHT

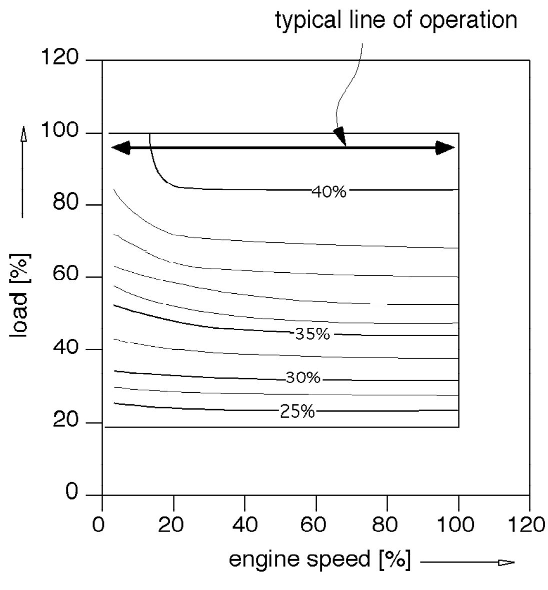
Figure 14. Total efficiency of converting the chemical bound energy of diesel fuel into hydraulic energy.

As can be seen the new combination of IFP engine and IHT has a modest but clear efficiency advantage. This is for a large part due to the differences in the way the two internal combustion engines are used. Figure 15 shows the fields of operation of the two engines including the iso-efficiency lines [18, 19]. Figure 15 also show two typical lines of operation. In combination with the IHT, the IFP engine will deliver its hydraulic energy at a constant pressure level: all load and flow variations at the load side of the transformer are ‘translated’ by the transformer in flow variations at the supply side of the transformer i.e. the load side of the IFP engine. Since the frequency –and with that the output flow– of the IFP engine can be changed instantaneously [11] the free piston engine matches very well with the characteristics of the hydraulic transformer. Also the complete frequency i.e. flow range in which the IFP engine can operate avoids the need of on-off-control or batch-control and with that avoids the need of large accumulators.

The current diesel engines neither have a fast response characteristic in the frequency domain nor can offer the complete speed range from zero to maximum speed. Most industrial diesel engines have a speed range of 1:2. As a consequence variable displacement pumps are needed to convert the variable pressure and flow demands in the supplied torque and speed of the engine, within the limited speed range of the engine. The resulting distribution of points of operation depends on the application. Figure 15a shows the typical line of operation of a hydraulic excavator [20]. Basically this is a constant speed control, which is completely opposite to the constant load control of the IFP engine.



15a. Conventional diesel engine



15b. Innas Free Piston engine

Figure 15. Comparison of the net effective efficiency of a conventional crankshaft diesel engine and a Innas Free Piston engine.

Figure 16 compares in the power domain the efficiencies of the two systems at the indicated lines of operation. The graph is based on real efficiency data of a 90 kW diesel engine and efficiency measurements on a 30 kW IFP engine. In order to realise a total output power of 90 kW, 3 IFP engines are put in parallel. Since the IFP engine allows individual cylinders to be turned off the efficiency stays at a high level and is almost constant at about 40% over the entire power i.e. flow range. Especially at part load this results in an efficiency advantage for the IFP engine.

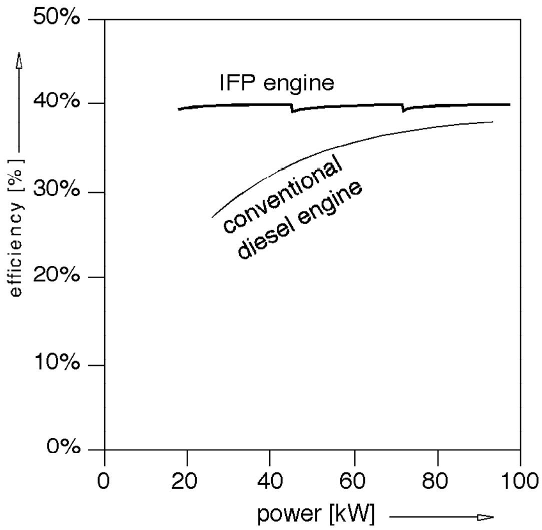
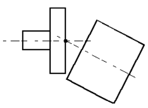
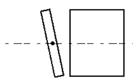
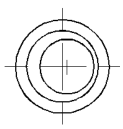
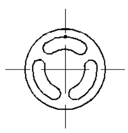


Figure 16. Efficiency of the IFP engine and a conventional diesel engine as a function of power output at the lines of operation shown in figure 15.

Table 2: Characteristics of variable hydraulic machines

Type	control displacement	control forces	theoretical control energy
	40 mm	2200 N	88 Nm
	20 mm	510 N	10 Nm
	7 mm	2000 N	14 Nm
	1 radial	< 5 Nm	< 5 Nm

In addition to this there are some other effects that directly will lead to a further reduction of specific fuel consumption. Probably most important is the capability of the combination of the IFP engine and the IHT to recuperate energy, even when applying implement systems. Also the IHT can be connected directly to a hydraulic cylinder or group of cylinders, thereby avoiding the losses of load sensing or other valves. Although this is in principle also possible by applying a separate variable displacement pump per hydraulic cylinder the lower cost of the IHT make this option much more feasible.

As the last point of the efficiency analysis the low load requirements for rotating the port plate should be mentioned. Table 2 shows the energy requirement for controlling the control angle of the port plate of the IHT and compares this to the theoretical control energy of today's hydraulic pumps and motors as given by Backé [21]. The lower energy needed to control the IHT will also increase the efficiency compared to current variable displacement motors and pumps. In addition it will make a direct electro-mechanical control more conceivable than with current motors and pumps.

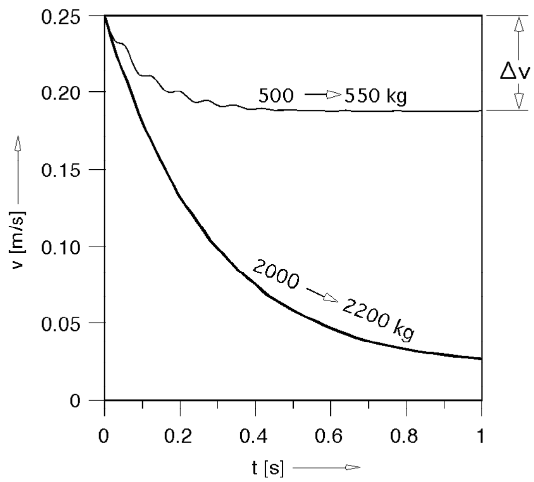
7 Dynamic control

Similar to the classical hydraulic transformer, the IHT is in principle a secondary controlled machine. As shown in the previous paragraphs the design of the IHT is radically different. These differences also have an effect on the dynamic response characteristics.

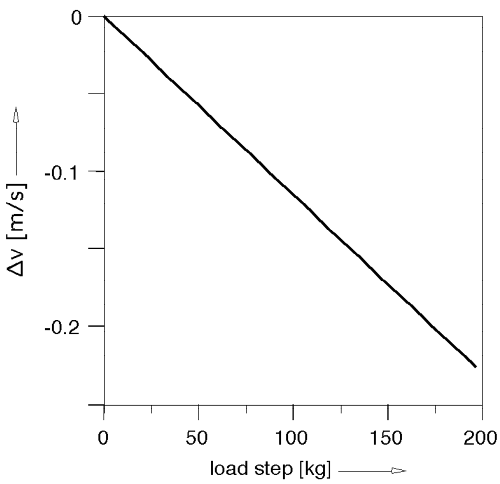
First of all the port plate of the IHT has a very low inertia and is well balanced. The torque that is needed to rotate the port plate is dominated by the friction between the cylinder barrel and the port plate plus the friction between the port plate and the end cap. This torque is very low compared to the forces needed to control variable displacement machines (see table 2). Consequently, the port plate can be turned very quickly without large effort i.e. without the need of large actuators. A variation of the port plate position immediately results in a change of the torque balance on the cylinder barrel of the transformer. Since this barrel has a rather low inertia, the change of torque results in a quick change of the rotational speed of the cylinder barrel i.e. of the flow delivered by the transformer. On the load side of the transformer the flow change will directly effect the load pressure. Since both the port plate and the cylinder barrel respond very fast the IHT has an inherent capability for a good responsiveness.

Unfortunately, if an implement system is controlled in open loop by means of an IHT, the resulting load disturbance rejection is poor. To illustrate this, figure 17a shows the speed response of the hydraulic cylinder to a 10 % change in load force, at a constant control angle δ. The figure was calculated for an IHT driving a small lift truck's lift cylinder, at typical operational parameters. The decrease of the equilibrium speed is linearly dependent on the load increase as can be seen in figure 17b. The sensitivity is evident. Mainly for operational reasons but in some applications also for safety reasons, a large sensitivity to disturbances in the load is not acceptable. Therefore, an IHT operated implement system requires some kind of feedback control.

Several options for adding a feedback loop to an IHT driven implement system, have been studied by Innas. The models were set up in MATLAB as linear state space models, routines from the MATLAB Control System Toolbox were used to evaluate them.



17a. Velocity hydraulic cylinder



17b. Stationary velocity drop

Figure 17. Load step response of an implement system with an uncontrolled IHT.

A very promising option for a simple hydromechanical feedback control was identified. A hydraulic scheme of this kind of control is shown in figure 18. The flow to the load passes a measuring orifice. A 4/3-valve is balanced between the resulting pressure drop and a spring force, representing the flow reference signal. If this balance is disturbed, the valve directs flow to the appropriate side of the port plate actuator's cylinder. This rotates the port plate. As a result the difference between reference and actual flow is eliminated. For reasons explained later the control contains a differential feed back of the port plate angle. This is realised by a series arrangement of a spring and a damper, acting between the valve and the control angle. The spring and damper can be integrated in the valve.

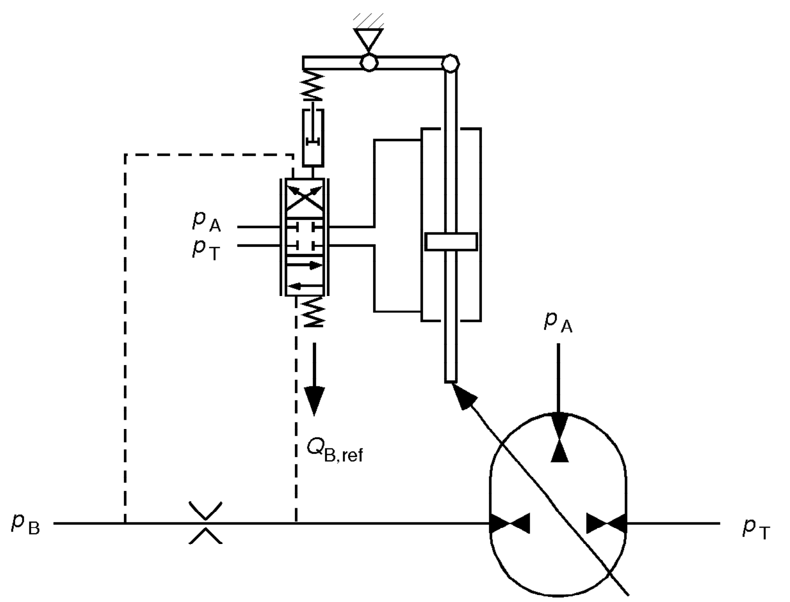


Figure 18. Example of a hydromechanical control for the port plate of the IHT

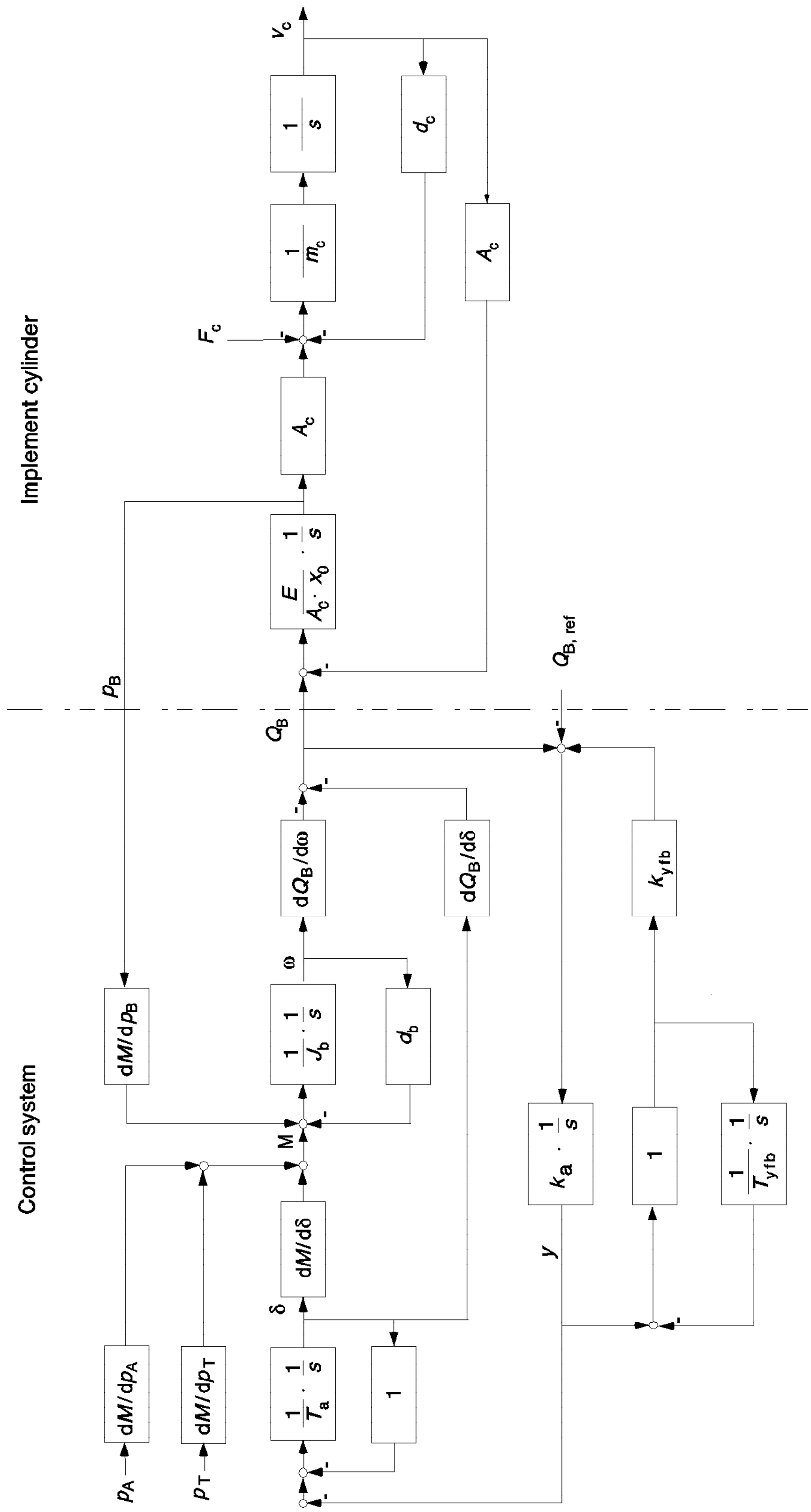


Figure 19. Block diagram of the IHT hydromechanical control as shown in figure 18

Figure 19 shows the block diagram of this control, applied to an IHT driving an implement cylinder. In this diagram the inertia of the transformer barrel (J_b), the mass (m_c) of the implement cylinder and the cylinder load F_c can easily be recognised. The integrator with integration constant $1/T_a$ and unity feed back, is not essential to the method of control. It serves as time lag element, representing the port plate actuators time lag. As described above the basic feed back loop is built around the flow to the load, which is directly proportional to the cylinder's speed.

Essential to this main feedback loop are the two integrators in series:

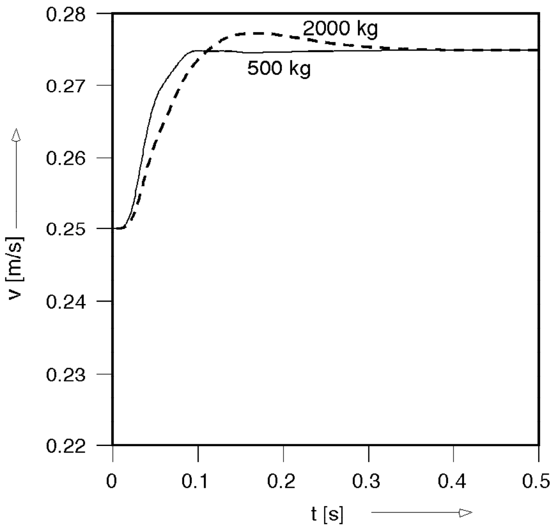
- The port plate actuator, a hydraulic cylinder which integrates flow to displacement (integration constant k_a).
- The inertia of the IHT's barrel, which integrates angular acceleration to speed (integration constant J_b).

This kind of series arrangement of two integrators is very common in secondary controlled systems. Without extra measures, it can lead to unstable behaviour of the controlled system. In [2] this has been demonstrated and several publications on secondary controlled systems describe the phenomenon (for instance [1],[5],[6] and [9]).

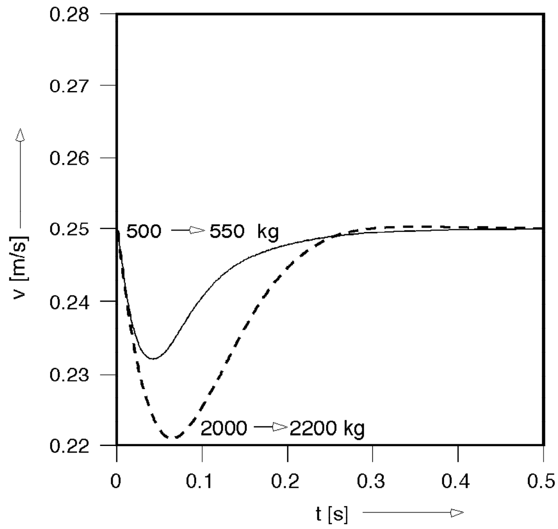
Simulations at Innas have shown that this would also happen in an IHT driven implement system. The stabilizing method chosen for this hydromechanical IHT control system, is to add an extra control angle feedback loop, which introduces extra damping in the system.
It is obvious that a proportional feedback of the control angle is not a good strategy. The control loop is designed to follow the speed command by adjusting the control angle around its equilibrium value. As this equilibrium value is mainly determined by the load pressure, proportional control angle feedback would introduce the same sensitivity to load disturbances that the feedback loop should prevent. The apparent solution is to use derivative feedback of the control angle. In the block diagram this derivative feedback is represented by the integrator with constant T_{yfb} and the gain with constant k_{yfb}.

For such a control system, used for the same lift truck's lift cylinder drive, the effects of varying the feedback parameters were studied by examining the associated paths of the root loci. A set of parameters could be identified, that results in a good responsiveness and adequate load disturbance rejection. For four different load situations, figure 20 gives the calculated responses to step inputs in the flow command and the load force. All responses were calculated at a speed of 0.25 m/s and with a constant rail pressure of 300 bar.

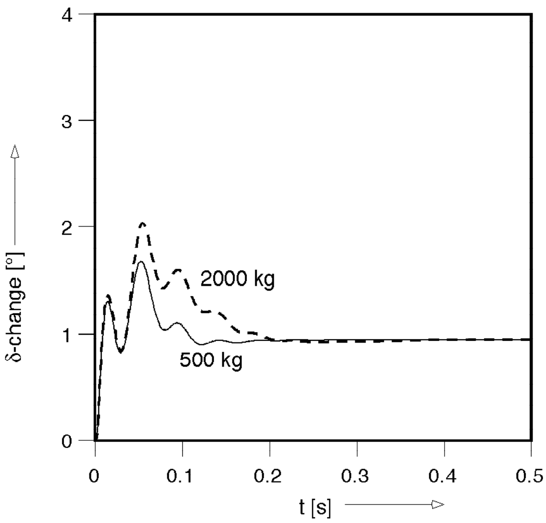
Although the effects of non linearities in the control system still have to be studied, it may be concluded that it should be possible to use a fairly simple, hydromechanical control system for an IHT driving an implement system.



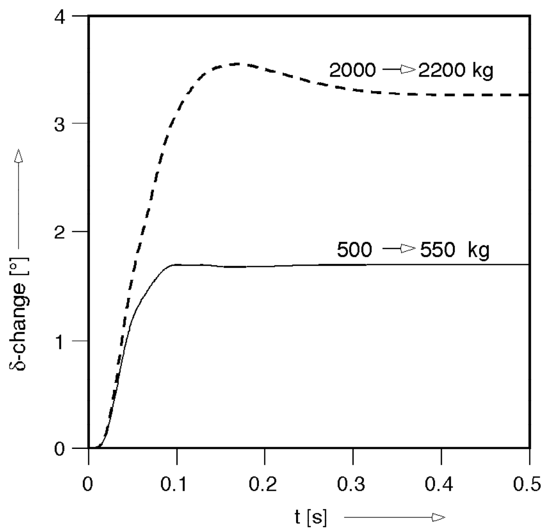
20a. v-respons to 10% step in speed command



20b. v-respons to 10% step in the load domain



20c. δ-respons to 10% step in speed command



20d. δ-respons to 10% step in the load domain

Figure 20. Calculated responsiveness and load disturbance rejection of an implement system with a hydromechanical controlled IHT

One of the major advantages of constant pressure rail systems, is the ease with which more functions can be connected to the same energy source. The only limit is given by the maximum power capability of this energy source. Some control strategy is necessary to handle a possible power shortage.
Conventional flow imposed systems need a similar power control strategy. In addition to this, they require extra, in most cases elaborate, circuitry to enable the parallel operation of more than one function, even when these do not exceed the power limit.

The hydromechanical IHT control solution described above, is a local one: without additional measures it cannot solve power conflicts originating from the simultaneous operation of different vehicle functions.

However, it can be easily applied in vehicles where the total power requirement of all implement functions, is smaller than the installed engine power. In such vehicles, if the implements are operated during hydrostatic driving and the combined power to the implements and the hystat drive motors exceeds the installed power, the natural thing to do is to reduce the drive power. In other words, every power conflict is solved by reducing the drive power.

When using an IFPE as the prime mover, implementing this strategy is easy. The total power delivered by the IFPE is known in the IFPE control unit at all times. When this exceeds the limit, the IFPE may signal the drive control to reduce the power to the wheel drive system, by reducing the control angle of the variable hystat drive motor(s) or the control angle of the IHT(s) driving the constant displacement hystat motor(s).

8 Future developments

The control system shown above is only an example how the IHT could be controlled. It illustrates clearly the aspects that are related to the control of the IHT. Meanwhile alternative controls have been developed for implement systems as well as for transformers that are going to be combined with constant displacement hydraulic motors for hydrostatic drives. In the latter application all four quadrants with respect to torque and speed can be realized by means of the port plate control.

Also, since the port plate only requires a very low torque, control systems are being designed with direct electromechanical actuation of the port plate. The possibility to control hydraulic power directly with electromechanical elements, would greatly improve circumstances for a breakthrough of electronic control in hydrostatic vehicles.

The developments are now shifting towards the application of the IHT, for a part in combination with the afore mentioned Innas Free Piston engine. Although hydraulic transformers are not new, only a small number have been applied. Due to their mechanical complexity they have only been applied in systems with large hydraulic cylinders [8]. The low cost solution of the IHT-concept, the high dynamic response capabilities and the high efficiency may however offer a break-through for secondary controlled systems in general and hydraulic transformers in particular.

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