VALVING LAND PHENOMENA OF THE INNAS HYDRAULIC TRANSFORMER

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Abstract

The design of a hydraulic pump or motor with a variable displacement could be much simpler, if a rotating valve plate could be used to vary the displacement. An essential aspect of the rotating valve plate is however that the passage of the cylinder ports from one kidney to the other often occurs while the pistons are moving. Experiments have revealed that this leads to high pressure spikes and cavitation. Because of this it is an unwritten rule that in hydraulic pumps and motors the passage of the so-called valving lands has to take place in the top and bottom dead centres.

Against this rule there is a new hydraulic transformer developed (the Innas Hydraulic Transformer or IHT) in which the rotating valve plate is introduced again. Instead of varying the displacement the position of the valve plate now defines the pressure ratio between the load and the supply port. As will be shown this has a crucial effect on the valving land phenomena. Instead of increasing pressure pulsations and cavitation the rotating valve plate can now help to reduce these effects.

Keywords: hydraulic transformer, valve plate, pressure pulsation, cavitation

1 Introduction

Generally speaking, scientists only write about their successes, not about their failures. Engineers and designers are not an exception to this rule. It could however be argued that we can learn more from failures and the causes of failure than of successes and that it is therefore more important to write about fiascos than about victories.

An example of such an unsuccessful project, about little to nothing has been written, is the attempt of a number of companies and institutes to design a new type of variable displacement hydraulic machine. Instead of using a variable drive plate or barrel angle, the angular position of the valve plate was used (see Fig. 1). The benefit of this concept is quit clear: the small, light and pressure balanced valve plate is much easier to control than the drive plate or the barrel. It is clear that the cost of such a pump or motor would be substantially lower than its conventional counterpart.

Despite its potential, the concept of a rotating valve plate has never been introduced. Aside from a number of patents (Smithson 1961, Budzich 1965, Stroze 1989) no written documentation can be found about the experiments or about the reasons of failure. The newly developed Innas Hydraulic Transformer (IHT) however is also based on the concept of a rotating valve plate (Achten et al 1997, Rotthäuser and Achten 1998, Vael and Achten 1998, Achten and Palmberg 1999, Werndin et al 1999a, Werndin 1999b). It is therefore important to find out the design aspects which lead to the failure of the concept of the rotating valve plate for hydraulic pumps and motors and see if these reasons also apply to the IHT.

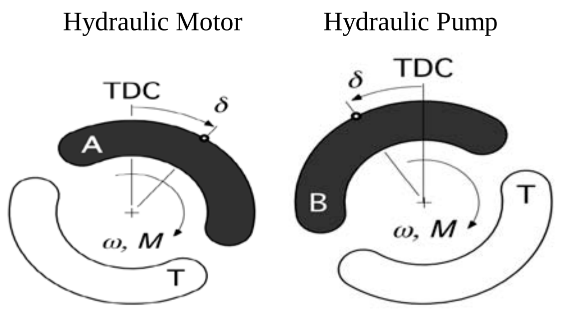

Fig. 1: *Variable displacement hydraulic machines by means of a rotating valve plate: TDC top dead centre, A high pressure kidney of the hydraulic motor, B high pressure kidney of the hydraulic pump, T low pressure kidney, δ control angle valve plate, ω rotational speed, M torque*

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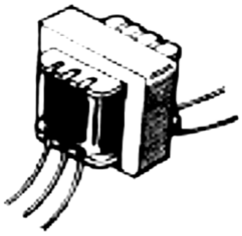
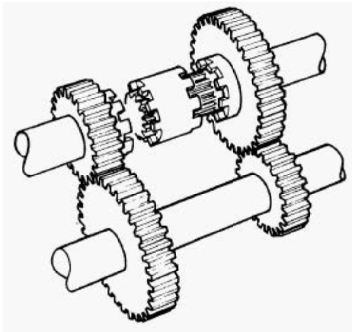
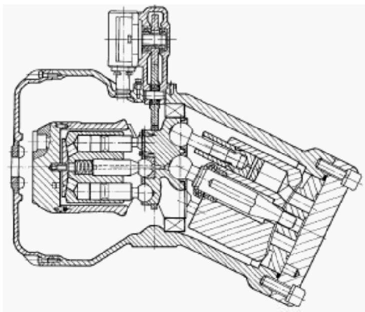


2a: electric transformer



2b: mechanical transformer



2c: hydraulic transformer

Fig. 2: *Transformers*

2 Needs for Transformers

Hydraulic transformers are not new. Some designs are known from patent literature (Tyler 1962, Kouns 1970, Reynolds 1974, Boehringer et al 1975, Leusch 1984, Mcgowan 1988, Dantlgraber 1997), others have really been built as a prototype. Yet only a few have been taken into production and operation (Kordak 1989, Kordak 1996). This is completely different in other fields of technology where transformers are very common. In mechanical systems, gear transmissions are the best known example of a transformer: they convert mechanical power into mechanical power, thereby changing the ratio between speed and torque. Aside from gear transmissions, which have a fixed transformation ratio, continuously variable transmissions or CVTs are examples of transformers in which the transformation ratio is variable.

In electrical systems, transformers are a key element. Most electric systems are built around a grid or common voltage rail: a power plant is supplying electric power to an electric power grid, which has in principle a constant voltage. At the load side the required voltage level will depend on the type of apparatus: some can do with the voltage offered by the grid, others, like radios and televisions need a different voltage level. If a lower voltage level is required, a resistor could realize the required voltage reduction. But this would lead to energy losses and heat. The transformer can do the same, without the dissipation and heat problems. And if a higher voltage level is required, a kind of transformer is obligatory, since a resistor can never create a higher voltage level.

If transformers are so widely used in mechanical and electric systems, why not so in hydraulic systems? It could be argued that there is not much need for a transformer since most hydraulic systems are currently flow or pump controlled. But the advantages of common pressure rail systems or secondary controlled systems are well known (Palmgren 1988, Kordak 1989, Dluzik 1989). There is more reason to believe that there are no common pressure rail systems on the market (aside from some exceptions), because of the lack of cost-effective and efficient secondary controlled components. This is especially true for the hydraulic transformer, which existed so far only as a combination of a secondary controlled unit and a constant displacement unit. Even if these two units were integrated into one housing (as shown in Fig. 2c), it is generally accepted that the end result would still be too expensive and the efficiency would be too low (Kordak 1989). The Innas Hydraulic Transformer or IHT has been developed to change this situation.

3 the Principle of Hydraulic Transformation

A hydraulic transformer is a component for controlling hydraulic power i.e. for converting hydraulic power into hydraulic power. In this respect it differs from most other hydraulic components in that it is not only a pressure or flow control component; it controls both at the same time.

Transformation is in principle a reversible process. This implies that the sum of all hydraulic power equals zero (flows toward the transformer are defined as having a positive sign):

$$\sum_{\mathrm{i}} p_{\mathrm{i}} \cdot Q_{\mathrm{i}} = 0 \tag{1}$$

With three hydraulic lines connected to the transformer, Eq. (1) results in:

$$p_1 \cdot Q_1 + p_2 \cdot Q_2 + p_3 \cdot Q_3 = 0 \tag{2}$$

If the third port would be connected to the reservoir of the hydraulic system (at a pressure $p_3 = 0$) this equation would be simplified to:

$$p_1 \cdot Q_1 = -p_2 \cdot Q_2 \tag{3}$$

The reason for the third flow Q_3 lies in the mass continuity equation:

$$\sum_{\mathrm{i}} m_{\mathrm{i}} = 0 \tag{4}$$

Assuming the compressibility of the oil can be neglected and the density of the fluid can be regarded as being constant this means that the total sum of all flows to and from the transformer has to be zero. With three lines connected to the transformer this results in:

$$Q_1 + Q_2 + Q_3 = 0 \tag{5}$$

If now for example:

$$p_1 > p_2 \tag{6}$$

then it follows from Eq. (3) that:

$$Q_1 < -Q_2 \tag{7}$$

In order to make up the difference between Q_1 and Q_2 a third flow Q_3 has to be added in order to fulfil the mass (flow) continuity equation. This implies that a hydraulic transformer will always be a hub with at least three connecting ports:

- a supply line
- a load line
- and a make-up line at a low pressure level

This is an important difference compared to throttling. A throttle is also a device that converts hydraulic energy into hydraulic energy. Only in the case of the throttle valve there are only two connections (one input and one output) and the output flow is equal to the input flow. Again compressibility effects are neglected. The pressure difference across the throttle valve is representing the amount of energy that is converted into heat: unlike transformation throttling is an irreversible process. Because of this irreversibility, oil in a throttle can only flow to a pressure level that is lower than at the input.

In a transformer the direction of the flows depend on the transformation or pressure ratio of the transformer. If the high-pressure supply-flow has a pressure level that is higher than at the output or load port the make-up flow will be directed towards the transformer. This means the make-up flow will have to go from the low-pressure level of the reservoir to the higher pressure level at the load port. This can only be realized by means of an active element that takes the oil molecules 'up stream' to the higher pressure level.

The most important design characteristics of a hydraulic transformer can therefore be defined as:

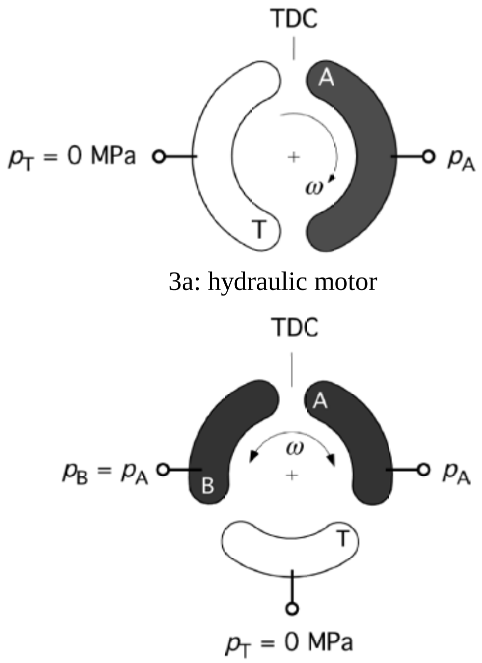
- there are (at least) three hydraulic lines connected to a transformer
- there is an active element needed to 'pump' oil from a relatively low pressure level to a higher pressure level

The conventional hydraulic transformer fulfils these requirements. The pump part can easily be recognized. And if we combine the two suction lines of the two hydraulic units then only three hydraulic connections remain: the supply line, the load line and the make-up line.

4 Theory of Operation of the IHT

The hydraulic transformer from Innas combines a number of well-known principles. Like in the conventional hydraulic transformer shown in Fig. 2c, the IHT has a pumping and a motoring part. This can in principle be realized by means of any type of displacement system. This article illustrates the principle of the IHT on the basis of an axial piston design, but it should be taken into consideration that the concept of the IHT could also be applied to radial piston designs, orbital, gerotor and geroler designs and various other displacement principles.

As in axial piston pumps and motors the IHT has a valve plate to switch the connection between the cylinders in the rotating barrel and the various pressure levels connected to the transformer. In a hydraulic pump or motor this valve plate only has two kidneys, one connected to the high and the other to the low-pressure side. But a transformer needs three hydraulic connections as shown before and therefore also needs three kidneys (Fig. 3b).



3b: hydraulic transformer

Fig. 3: *Valve plate configurations (TDC = top dead centre of the pistons)*

In case of the hydraulic motor (Fig. 3a) the balance of the barrel torque is created by a load torque at the shaft of the motor, which offsets the motoring torque, created at the A-kidney. In the transformer the torque balance is established by having a third kidney (the B-kidney). In the example given in Fig. 3b:

- the B-kidney has the same arc length as the A-kidney and
- these two kidneys are positioned exactly symmetrically around the top dead centre (TDC).

In case $p_B = p_A$ there is a complete torque balance and there is no remaining torque to change the rotational speed of the barrel. The rate of flows to and from the transformer would remain as they were and could in principle be in any direction. Since the T-kidney is positioned completely symmetrical around the bottom dead centre (BCD) there will be no effective flow going from the reservoir to the transformer. This is in correspondence with Eq. 3 and 5 and the condition that p_B is equal to p_A from which it follows:

$$Q_{\mathrm{T}} = Q_{\mathrm{A}}\left(\frac{p_{\mathrm{A}}}{p_{\mathrm{B}}} - 1\right) = Q_{\mathrm{A}}(1-1) = 0 \qquad (8)$$

In order to change the rotational speed of the barrel (i.e. the flow rates from the transformer) the torque balance has to be disturbed. In theory changing the dimensions of the A- and B-kidney could do this. If for instance the arc length of the A-kidney would be increased and the length of the B-kidney reduced, the motoring torque delivered by the A-kidney would become larger than the pumping torque asked by the B-kidney. The net effective torque that results from this difference would then accelerate the rotational speed of the barrel and increase the flow output to the B-kidney. But the dimensions of the kidneys can only be chosen once and are fixed as soon as the valve plate is produced. In order to control and vary the torque balance of the transformer barrel during operation a different solution has to be found.

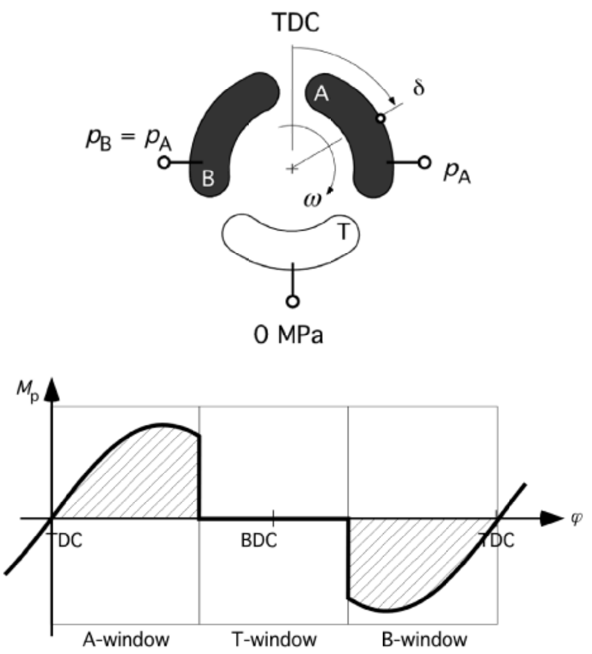


Fig. 4a: *Position of the valve plate of the IHT and torque curve for one piston at two different pressure levels at the load side (B-port) of the transformer: load pressure equal to supply pressure ($p_B = p_A$)*

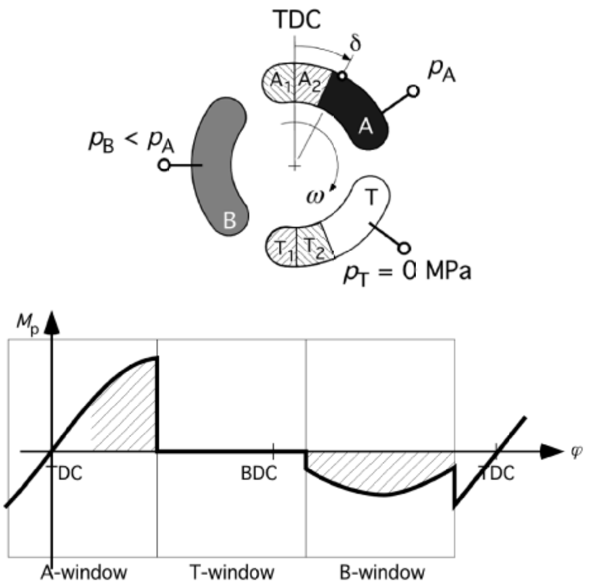


Fig. 4b: *Position of the valve plate of the IHT and torque curve for one piston at two different pressure levels at the load side (B-port) of the transformer: load pressure less than supply pressure ($p_B < p_A$)*

For this different designs are conceivable. The most favourable solution is to control the rotational position of the valve plate. The effect this has on the torque balance is illustrated in Fig. 4. The control angle δ is arbitrarily defined as the rotational position of the valve plate between the middle of the A-kidney on the valve plate and the TDC-point on the housing.

On the left side of Fig. 4 two different positions of the valve plate are shown. The first one is equal to the situation shown in Fig. 3b. The right side of Fig. 4 shows the torque generated by one piston during one revolution of the barrel. During this revolution the piston is connected to respectively the A-kidney the T-kidney and the B-kidney, as is represented in the diagram by the so-called ‘windows’. The hatched areas represent the energy that the pistons need or deliver during this barrel rotation. As can be seen the torque delivered by the T-kidney is zero since the pressure at that side is assumed to be zero. In Fig. 4a the torque curve in the A-window is equal to the torque curve in the B-window; only the sign is opposite. Because of this there is a balance between the energy delivered by the A-window and the energy supplied to the B-window.

Figure 4b shows the situation when the B-pressure is reduced. In order to restore the torque balance, the valve plate is now rotated to a smaller angle. As a result part of the A-kidney is positioned left of the TDC. Assuming the barrel is rotating clockwise, this part of the A-kidney (represented by area A_1 in Fig. 4b) is now on the pump side. To the right of the TDC an area A_2 is indicated which has the same arc length as A_1. The energy generated at area A_2 is in principle equal to the energy needed for the pump action of A_1. Since these two areas are compensated by each other the remaining black area of the A-kidney is the only effective part of the A-kidney. The end effect is that, whereas the pressure at the A-side is still high, the energy and torque generated by this part is reduced by the decreased angle over which the A-kidney is effective.

This is also shown in the torque diagram of Fig. 4b. The torque curve still follows the same line as in Fig. 4a but the effective area (the hatched area) is now reduced. The B-kidney is however still effective for the full arc length. But at that side the lower pressure results in a decrease of the torque curve and subsequently in a reduction of the energy area underneath this part of the curve. In the end again a balance is established as soon as the (effective) area in the A-window is equal to the energy area in the B-window.

With this principle any pressure or transformation ratio can be established. Assuming:

- a zero pressure at the T-side
- an arc length of zero degrees of the overlap between all successive kidneys
- no torque losses

the following equation can be derived for the pressure ratio:

$$\frac{p_B}{p_A} = \frac{\cos\left(\frac{1}{2}\alpha + \delta\right) - \cos\left(-\frac{1}{2}\alpha + \delta\right)}{\cos\left(-\frac{1}{2}\alpha + \delta - \beta\right) - \cos\left(-\frac{1}{2}\alpha + \delta\right)} \tag{9}$$

where:

p_A = pressure in the A-kidney (supply side of the transformer)
p_B = pressure in the B-kidney (load side of the transformer)
α = effective arc length of the A-kidney
β = effective arc length of the B-kidney
δ = control angle of the valve plate

This relation is also shown in Fig. 5 in which the three kidneys are assumed to have an equal effective arc length of 120° each. The control angle δ in this diagram is shown at the horizontal axis.

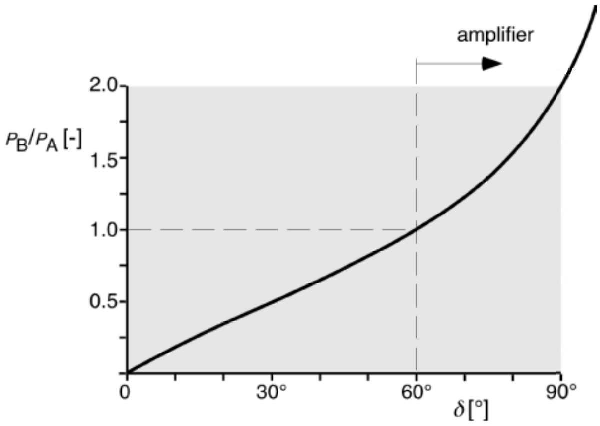


Fig. 5: *Transformation ratio p_B/p_A as a function of the control angle δ for an axial piston IHT with three equally sized kidneys (120°)*

As can be seen the pressure ratio becomes greater than 1, if the control angle δ is higher than 60°. In that case the transformer acts as a pressure amplifier. In theory the pressure ratio can become infinite if the control angle comes close to 120°. In reality the transformation ratio will be lower because of leakage and friction. Other transformation curves can be realized by means of changing the arc length of the various kidneys or by changing the pressure at the T-port.

5 Pumps and Motors with Rotating Valve Plates

The experiments with rotating valve plates date back to the sixties and seventies. In this period companies like Volvo Hydraulics and Parker have studied this concept. Discussions with engineers from these companies and from several institutes have revealed that detrimental pressure effects i.e. cavitation and high pressure peaks in the valve land between two successive kidneys were the most important reason why the concept did not become a success.

Whenever a port of a cylinder of the barrel is positioned between two kidneys the enclosed oil volume has no way out or in (thereby neglecting leakage). Every movement of the plunger will subsequently lead to a compression or expansion of the oil volume. The compression or expansion can be calculated by means of the formula:

$$\partial V = -\frac{V_0}{K}\partial p \tag{10}$$

in which:

V = volume (V_0 = starting volume at the beginning of the compression)
p = pressure
K = compressibility modulus

In addition the opening area between a cylinder port and a kidney will be reduced while the port is leaving or entering the kidney. If oil has to be displaced through this area the throttling effect of the reduced opening will influence the pressure in the cylinder. A high piston velocity will thereby result in a large throttling effect.

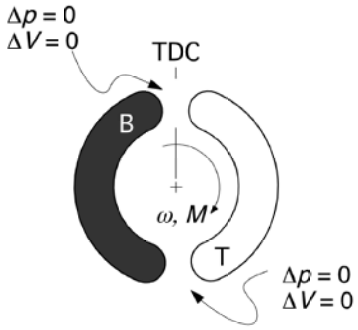


Fig. 6a: *Pressure build up and volume change in the valving land of variable displacement hydraulic pumps due to compressibility of the oil: conventional*

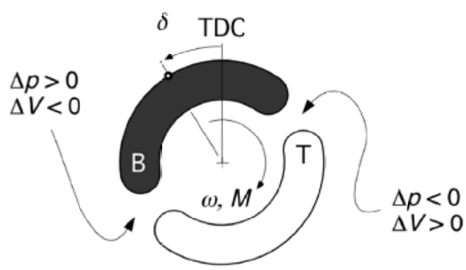


Fig. 6b: *Pressure build up and volume change in the valving land of variable displacement hydraulic pumps due to compressibility of the oil: with rotating valve plate*

Figure 6a shows the situation in a conventional hydraulic pump. As can be seen there is no effective change of volume in a conventional hydraulic pump (or motor) since the valve land occurs at the top and bottom dead centres of the plunger movement. Therefore the piston movement around the top and bottom dead centres cannot lead to cavitation or pressure spikes. Furthermore in the pump and motor configuration of Fig. 6a the piston velocity is close to zero while going from one kidney to the other and thus the throttling effects of port closing and opening are limited.

In the alternative concept (Fig. 6b) the crossing over from one kidney to the other however occurs while the pistons are moving. This results in a pressure change due to the compressibility of the oil, which is predominantly dependent on the dimensions of the pump and the control angle of the valve plate:

$$\Delta p_{\text{compr}} = K \cdot \ln\left(\frac{V_2}{V_1}\right) \tag{11}$$

where:

V_1 = enclosed volume at the start of the valve land
V_2 = enclosed volume at the end of the valve land

On the other hand the load pressure is in principle not dependent on the control angle of the valve plate. The position of the valve plate only controls the (effective) displacement volume. This is shown in Fig. 7, which shows the *p*-*Q*-diagram of a hydraulic pump or motor with a rotating valve plate. For a specific value of δ a complete range of load pressures can be applied.

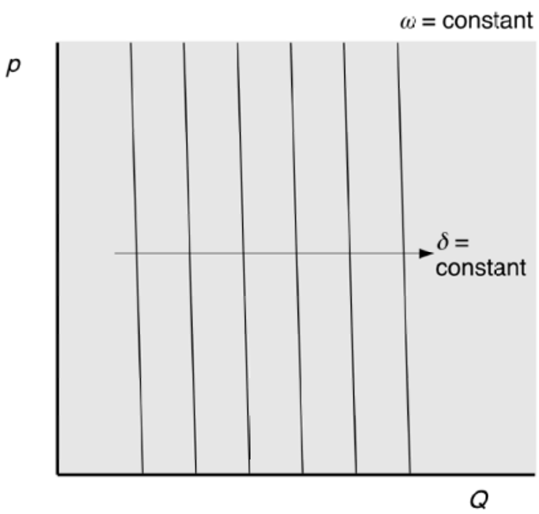


Fig. 7: *Lines of constant control angle δ in the p-Q-diagram at constant speed of a hydraulic unit with displacement control by means of rotating the valve plate of the unit*

In equations the problem can be described as follows:

$$p_{\text{kidney1}} - p_{\text{kidney2}} \neq \Delta p_{\text{compr}} \tag{12}$$

This mismatch can result in a high pressure build up as well as in cavitation. Both have a strong negative effect on the operation of pumps or motors.

6 IHT Valving Lands

How does the design of the IHT affect the pressure build up in the valve land areas? To answer this question it is important to understand that the control angle of the valve plate sets the pressure ratio in the IHT. This is illustrated in Fig. 8, which shows the diagram for a transformer build on the basis of a bent axis motor with a displacement of 45 cc/rev. The lines with a constant control angle δ are about horizontal in the IHT-diagram. This is completely different from the pumps and motors with a rotating valve plate where these lines are almost vertical in the *p*-*Q*-diagram (see Fig. 7).

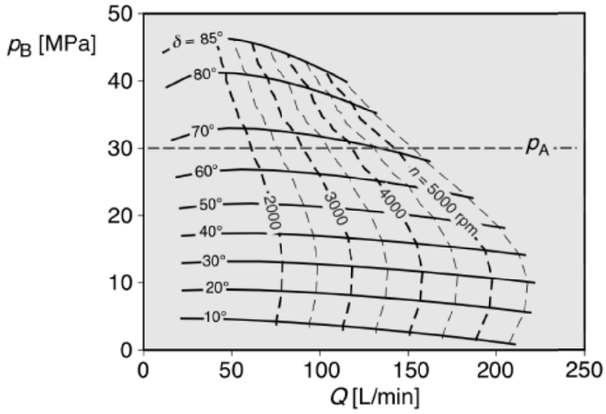


Fig. 8: *Diagram for an IHT based on the A2FM45 bent axis motor from Rexroth showing lines of constant control plate angle δ and lines of constant rotational speed n (p_A = 30 MPa)*

Aside from the pressure in the B-kidney the control angle δ also influences the position of the valving lands relative to the sinusoidal movement of the pistons. If for instance a valving land is located at the top or bottom dead centre positions, the piston movement is almost zero, whereas in the middle between the two top dead centres the volumetric changes will be at a maximum. This results in a pressure change in the cylinders of the barrel that will be dependent on the location of the valving land.

In the IHT it is possible to choose the design parameters as such that the effect of the δ-angle on the pressure difference between successive kidneys will be about equal to the pressure change due to compressibility. This is illustrated in Fig. 9, which shows two positions of the valve plate. In the left picture $\delta = 0°$ and subsequently the pressure in the B-kidney is equal to the pressure in the T-kidney. This means that between the B- and the T-kidney there is no pressure difference. At the same time the valving land between these two kidneys is at the BDC and there is no effective volumetric change i.e. no pressure change due to compressibility. This is completely different at the valving land between the B- and A-kidney. When passing this valving land the pistons move at a high speed towards the TDC-position, thereby compressing the oil. But the pressure increase, which is the result of this compression, is also welcome since at this position of the valve plate the A-pressure is much higher than the B-pressure.

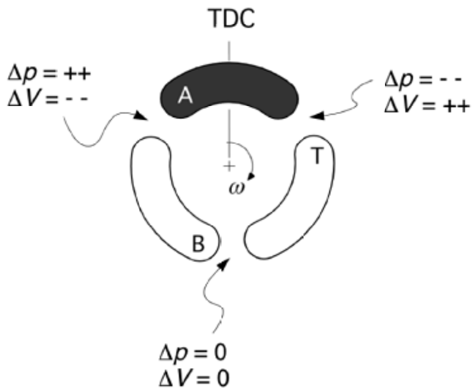


Fig. 9a: *Variation of cylinder volume and pressure during the passage of the valving lands in the IHT at two different positions of the valve plate. The three kidneys are equal in this example: control angle δ = 0°; $p_B = p_T$*

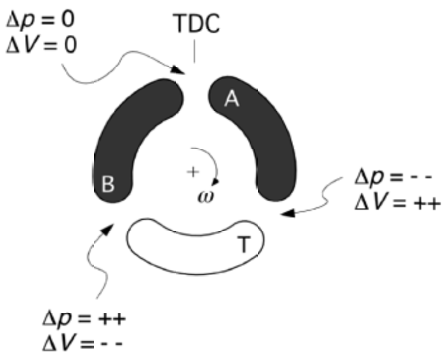


Fig. 9b: *Variation of cylinder volume and pressure during the passage of the valving lands in the IHT at two different positions of the valve plate. The three kidneys are equal in this example: control angle δ = 60°; $p_B = p_A$*

Fig. 10: *Prototype of the IHT showing the concurrent shapes of the cylinder ports and the kidneys*

When the valve plate is rotated 60° clockwise (as shown in Fig. 9b) the pressure in the B-kidney will rise due to the transformation inside the IHT. Subsequently the pressure difference between the B- and A-kidney will reduce until finally the B-pressure will be equal to the A-pressure. But at the same time the volumetric change in the cylinders will be reduced as well until finally it becomes effectively zero. As can be seen the pressure change due to compressibility effects in the valving lands corresponds with the pressure rise between the kidneys pairs.

The periods just before and after the valving lands are also of great importance. Whenever for instance a cylinder port leaves a kidney the opening area between the port and the kidney reduces gradually to zero (assuming a positive overlap). Especially during the end of this closing process the throttle effects increase exponentially. This not only leads to high energy losses (Werndin et al 1999a, Werndin 1999b) but also influences the pressure in the cylinders of the barrel during the passage of the valving land.

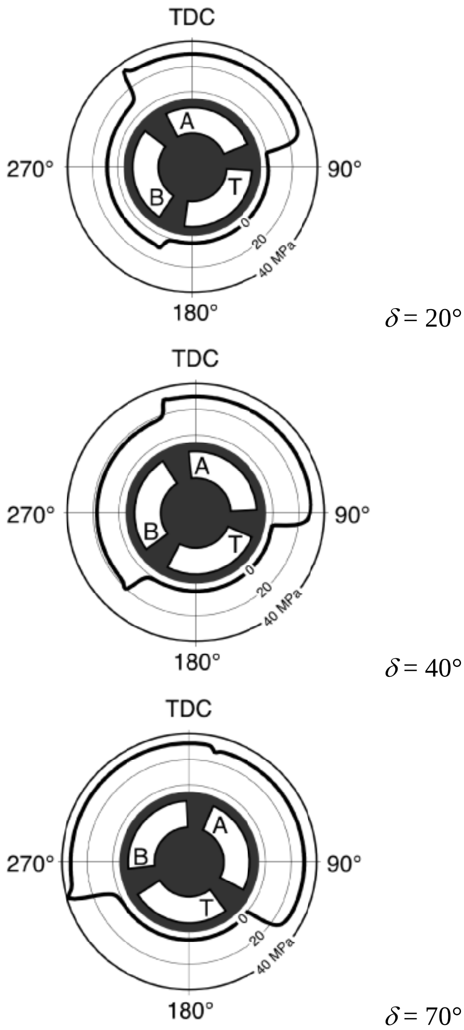


Fig. 11: *Calculated pressure in a single cylinder of the IHT-barrel during one revolution for three different control angles of the valve plate (45 cc/rev, 9 pistons, n = 3000 rpm)*

To minimize the port opening and closing effects it is important to make the leading and trailing edges of the cylinder ports concurrent with the leading and trailing edges of the kidneys. This can be realized by giving both the ports and the kidneys the shape of ring segments. This can be seen in Fig. 10, which portrays a photo of the barrel and the front face of the valve plate.

To optimize the design parameters a model has been developed to calculate the valving land phenomena in the IHT. One of the results of this model can be seen in Fig. 11, which shows the pressure trace of a barrel cylinder during one revolution for three different valve plate positions.

As can be seen the IHT can be designed to accommodate the pressure difference between the kidney pairs by creating a pre-compression or pre-expansion in the valving lands. The difficulty is that this should not only be realized for three kidneys simultaneously but also for all possible positions of the valve plate. The polar diagrams of Fig. 11 only show the pressure trace for a rotational speed of the barrel of 3000 rpm. It is however possible to design the IHT in such a way that similar good results can be found over the entire speed domain. The fit between the compression effects and the pressure difference across each of the kidney pairs is not perfect, but it is much better than in most pumps and motors.

7 Conclusions

In all hydraulic units with a rotating valve plate (including the IHT) the angular position of the valve plate controls the pressure change due to the compressibility of the oil in the valve land area's between the kidney pairs of the valve plate. In hydraulic pumps and motors that are built on the basis of this concept the pressure difference between two successive kidneys is independent of the control angle of the valve plate. This results in a mismatch between the required pressure change between two successive kidneys and the pressure difference due to compressibility. Cavitation and pressure peaks are the negative effects of this fundamental mismatch.

In the IHT not only the pressure effects due to compressibility but also the load pressure itself is a function of the control angle of the valve plate. Although not perfect, the dimensions of the IHT can be chosen as such that the pressure level in the enclosed cylinder volume is at the end of the valve land close to the pressure level in the following kidney. The match is better than in current pumps and motors where the pressure difference over the valve land equals the difference between the high and low pressure side of the unit.

Nomenclature

A	area	[m^2]
K	compressibility modulus	[Pa]
m	mass	[kg]
M	torque	[Nm]
p	pressure	[Pa]
Q	flow rate	[m^3/s]
V	volume	[m^3]
α	effective arc length supply kidney	[rad]
β	effective arc length load kidney	[rad]
δ	control angle valve plate	[rad]
φ	rotational position of the barrel	[rad]
ω	rotational speed	[rad/s]

Subscript

A	supply side
B	load side
compr	compression
i	index
p	Piston
T	tank or reservoir side

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