Dedicated design of the Hydraulic Transformer

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1 Introduction

The very first passenger cars looked like carriages without horses. In fact that's what they were: carriages driven by means of an engine. Likewise the first prototypes of the Innas Hydraulic Transformer (IHT) looked like a hydraulic motor without a load, or like a hydraulic pump without a drive connected to its shaft. And that is also what they were: converted hydrostatic machines, using as much parts of the original machines as possible /1/.

Although this was a fast and convenient way to prove the principle of the new IHT, it didn't give much information about the real potential of the transformer, whether this concerns dimensions and power density, costs, efficiency or noise. The requirements for the design of the IHT are essentially different from the design demands for pumps and motors. It is obvious that these differences will result in a design, which differs as much from the original hydrostatic machines as the passenger car differs from a carriage.

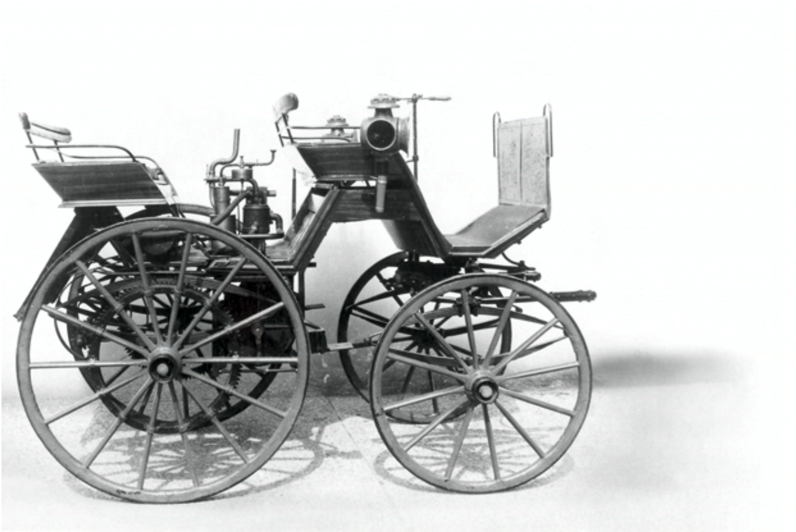


Figure 1: Daimler Motor Carriage 1886: a carriage without a horse

This paper describes the first concept of a dedicated design of the IHT, starting from a clean sheet of paper. Although the IHT can be applied for controlling hydrostatic motors as well as cylinders, the design that is described in this paper is especially focussed on the control of hydraulic cylinders. The transformer should be designed as such that it could be integrated in the base of a hydraulic cylinder. In the end this would result in a variable hydraulic cylinder with a high-pressure connection, a low-pressure connection and some wires to control the cylinder.

The final perspective of this development is that the variable cylinder will become a module with its own integrated controls. The control can be simple or elaborate, depending on market demands and the wishes of the customers. All of these cylinders can be applied in a common pressure rail (CPR) system, with the common pressure rail as a simple, well-defined backbone. System design, troubleshooting and maintenance will become much more simple than with current load sensing systems, and the flexibility of the hydraulic system will be much improved.

2 Design specifications

In the Innas Hydraulic Transformer (**figure 2**) the ratio between input pressure and output pressure as well as the ratio between supply flow and load flow, is controlled by setting the rotational position of the port plate.

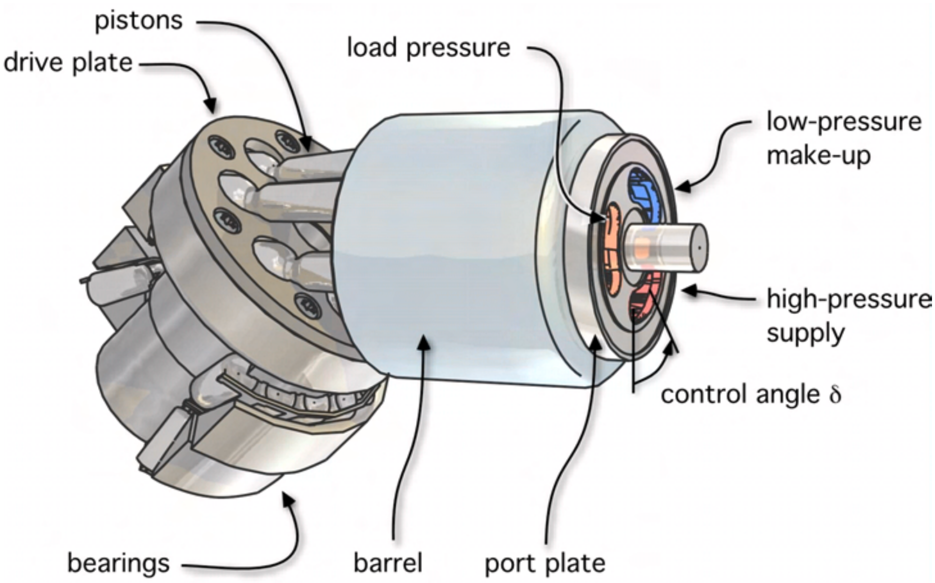


Figure 2: First prototype of the Innas Hydraulic Transformer, showing only the rotating parts and the port plate. Except for the port plate and the end-cap all the other parts originated from a bent axis axial piston motor

The advantage of this concept is that the motoring part and the pumping part of the transformation process are integrated into one single constant displacement machine. The downside of this concept is however that the port switching often occurs while the pistons have a relatively high speed. Without any measures taken this would result in:

- Energy losses, instability, reduced lifetime and increased noise levels due to pressure spikes in the displacement volumes during the passage from one port to the other.
- Start-up problems and difficulties in operating the transformer at low output flows due to variation of the torque as a function of the barrel position.
- Increased system noise and reduced life time due to flow variations.

By introducing the shuttles /2/ a simple and effective solution has been found for eliminating the pressure peaks. The shuttles also have a limited reducing effect on the torque variation since they take away the torque increase due to the pressure spikes. But the shuttles can't take away all torque variations and most certainly not the flow variations.

The most obvious way to reduce these variations is to increase the number of pistons. **Figure 3** shows the difference in torque and flow variations between a transformer with 7 pistons (as shown in figure 2) and a transformer with 18 pistons.

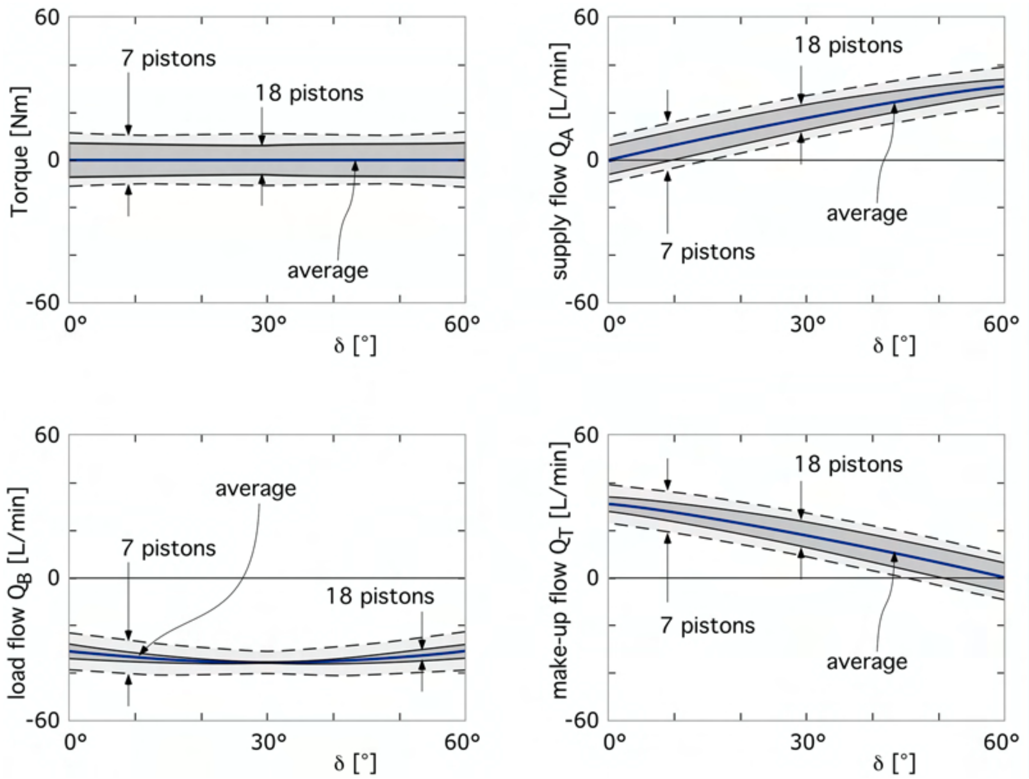


Figure 3: Effect of the number of pistons on the torque and flow variations of the IHT as a function of the control angle δ of the valve plate. The grey bands depict the minimum and maximum values of the variation.

This paper describes a concept of a dedicated design of the IHT for which the prime target is to reduce the torque and flow variations by means of increasing the number of pistons. Furthermore the new design will have a new bearing and sealing structure. Since the IHT only converts hydraulic energy, the shaft and it's sealing can be eliminated completely. We have also considered it to be important to reduce the structural load of the bearing and the housing of the IHT. Aside from increasing the volume and the costs of the bearing, a high bearing load will inevitably result in a high structural load on the housing. If the load varies, as is often the case in hydraulic machines, the vibrations will increase the noise level of the machine. Reducing the bearing load will therefore diminish an important source of noise.

The design of the new IHT will furthermore be optimized to give a low start-up torque. This will allow the transformer to start up even at high loads. A low start-up torque also reduces the torque that is necessary to rotate the port plate. If possible the port plate has to be controlled directly by means of an electro-mechanical actuator, without the need of a hydraulic interface. In order to achieve this, special attention must be given to the hydrostatic balancing of the barrel and the port plate.

Finally the new design will have to fulfil all demands with respect to costs, power density, efficiency, durability, reliability and robustness.

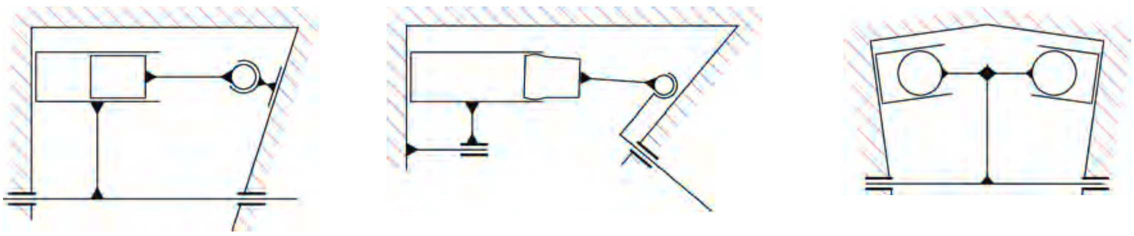
3 Basic principle

First the number of displacement volumes (for instance piston-cylinder combinations) has to be set. In principle this is a matter of diminishing returns: the largest reduction is realized by having 18 displacements per revolution. An additional increase would further reduce the torque and flow variations but the reductions would be limited and would probably not be justified by the extra parts, complexity and costs.

We have chosen to design the new IHT on the basis of the axial piston principle with 18 pistons. The axial piston principle offers advantages in terms of power density and efficiency. But axial piston pumps and motors are also relatively expensive. This is for a large part due to the kinematical principle, the bearing structure, and the high precision of the pistons. Increasing the number of these parts would, as it seems, bring about a further cost increase. This is both true for the bent axis principle as well as for the in-line slipper type machines. In order to increase the number of displacement volumes without increasing the costs, a different axial piston principle had to be found.

Figure 4.c shows the new principle, alongside with the familiar slipper-type and bent-axis principles. There are a number of similarities between the three principles:

- The displacement is realized by means of pistons moving up and down in cylinders;
- The cylinders are positioned in a barrel and the cylinder axis is approximately parallel to the axis from the barrel;
- The barrel rotates on top of a port plate and the pair of barrel and port plate acts as a valve similar to the way a commutator works in an electric motor.

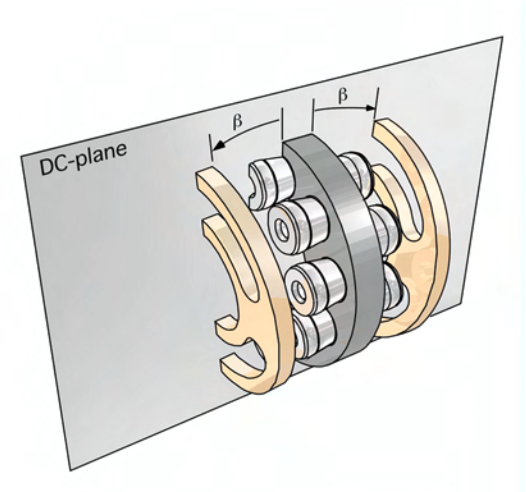


4a: slipper principle　　4b: bent-axis principle　　4c: new balanced principle

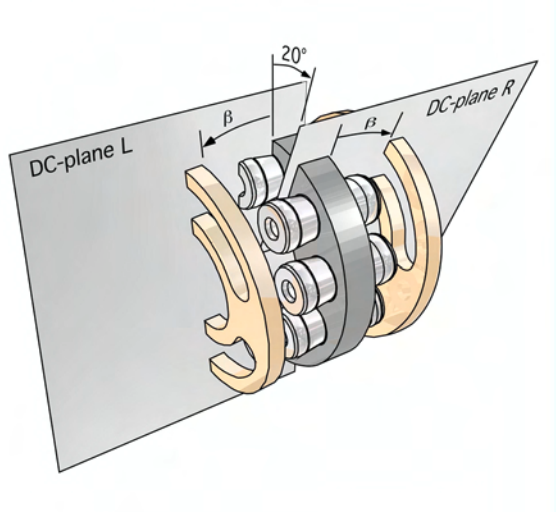
Figure 4: Axial piston principles

Instead of having one barrel the new principle has two barrels in a mirrored configuration. Now the large axial hydrostatic forces on both sides are completely balanced by each other.

With a total number of 18 displacement volumes, each barrel will have nine cylinders. However, if the displacement on the left side is exactly simultaneous with the displacement on the right side, the whole machine will act as if there were only 9 pistons. In order to establish a configuration of 18 pistons it is possible to dislocate the pistons on the left side relatively to the pistons on the right side. The pistons on the left side are then positioned 'in between' the pistons on the right side, having a phase shift of half the piston pitch (see **figure 5.a**). In the new design of the IHT we have conversely chosen to keep the pistons in line and to shift the angular plane of the valve plates (the principle is explained in **figure 5.b**). This brings about a rotation of the top and bottom dead centres, that is to say a phase shift of the displacement curves. In order to position the displacement curves from the right side exactly in between the displacement curves from the left side, the rotation of the plane where the dead centres occur (the DC-plane) has to be half the pitch between the pistons. For a configuration with on each side 9 pistons this means a rotation of 20°.



5a: Top and bottom dead centres in the same dead centre (DC-) plane, shifted piston positions



5b: Equal radial piston positions, phase shift between left and right DC-planes

Figure 5: Configurations with 18 out of phase displacement volumes. Only the rotor with the pistons and the valve plates (each with three ports) are shown.

In both cases the phase shift, either of the piston positions or of the dead-centre planes, will result in some axial forces on the rotor. However these forces are still much less than in case of a one-sided construction having only pistons on one side of the rotor. The advantage of having equal piston positions (figure 5.b) is that the piston pairs can be made from one piece, thereby only increasing the piston number from 7 to 9. Yet effectively the unit will operate with 18 displacement volumes.

4 Piston ring friction

In the new concept the pistons are rigidly attached to the rotor: the rotor and the pistons rotate as one piece and there is no linkage between the pistons and the rotor. Given the tilted position of the barrel (see **figure 6**) the circular movement of the pistons will have an elliptic path on the rotating plane of the barrel. Choosing the right dimensions can minimize the difference between the circle and the ellipse but, given a tilting angle between the barrel and the rotor of at least a few degrees, the elliptic deviation will never be zero.

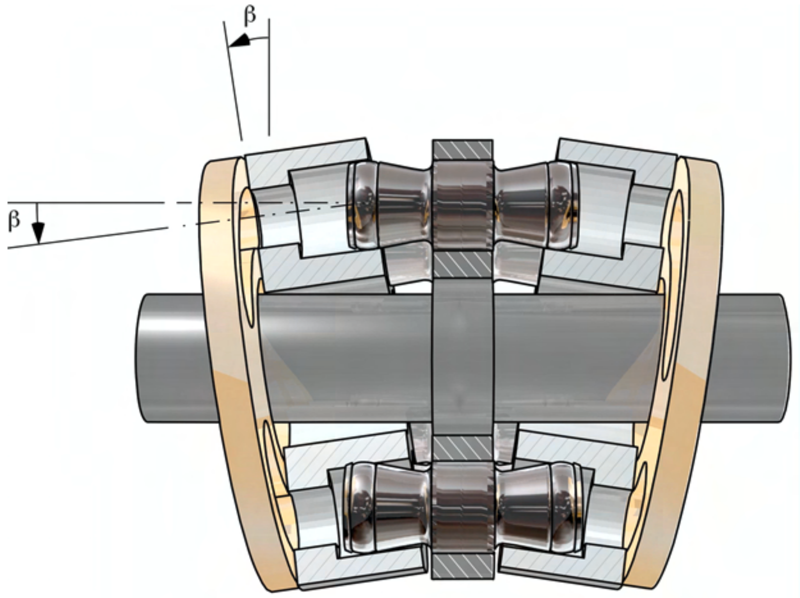


Figure 6: The angle between piston and cylinder equals the tilting angle of the barrel

If the tilting angle of the barrel is small, the elliptic difference, as well as other tolerances might be overcome by using piston rings. The gap that has to be closed by means of the piston rings will introduce an (extra) axial load on the rings, and the larger the gap the higher the structural loading of the rings. But there is a much bigger problem concerning the piston rings. Aside from the axial load, the piston rings will also introduce a radial force between the ring and the cylinder wall. The radial force created by the ring is even essential for the working principle since it is necessary to create the driving torque on the barrel. But if the torque is delivered via the barrel, the friction between the piston rings and the cylinder will be relatively high, especially during start-up.

The friction could be much higher than in current bent axis units. This is due to the relatively large angle between the pistons and the cylinder axis in the new transformer design. In bent axis motors and pumps the pistons can swivel at the joint with the drive plate. As a result the angle between the piston and the cylinder axis is mostly limited to about 3°. Yet, in the new design the pistons have a fixed position and the angle between the piston and the cylinder axis equals the angle between the rotor and the barrel (see figure 6). In order to increase the power density of the concept, this angle has to be at least 6° and preferably even more than 9°. At such large tilting angles the unbalanced design of the (conventional) piston rings would substantially increase the radial load between the piston rings and the cylinder wall.

Without any measures taken, the larger tilting angle of the pistons will result in an increase of the start-up torque. The start-up torque could even get to a level at which it would become impossible to start the transformer. Aside, the increased piston ring friction would substantially reduce the efficiency of the transformer.

The situation would change if the piston rings could be avoided. Then the hydraulic pressure in the cylinder would directly create a force on the piston and there would be hardly any friction between piston and cylinder. This situation is shown in **figure 7**. The line of contact between the spherical surface of the piston and the cylinder wall would always be in a plane, which stands perpendicular on the axis of the cylinder. This implies that the cylinder itself is always completely balanced and will not create a force upon the piston (aside from the friction between barrel and port plate as well as from inertia).

But even without the gap created by the elliptic projection, a piston ring is highly desirable. Especially since the transformer is meant to be applied for controlling hydraulic cylinders. It is well known that hydraulic cylinders are a major input for dirt particles /3/. Without the use of piston rings it would not be possible to compensate the radial gap between the piston and the cylinder for wear.

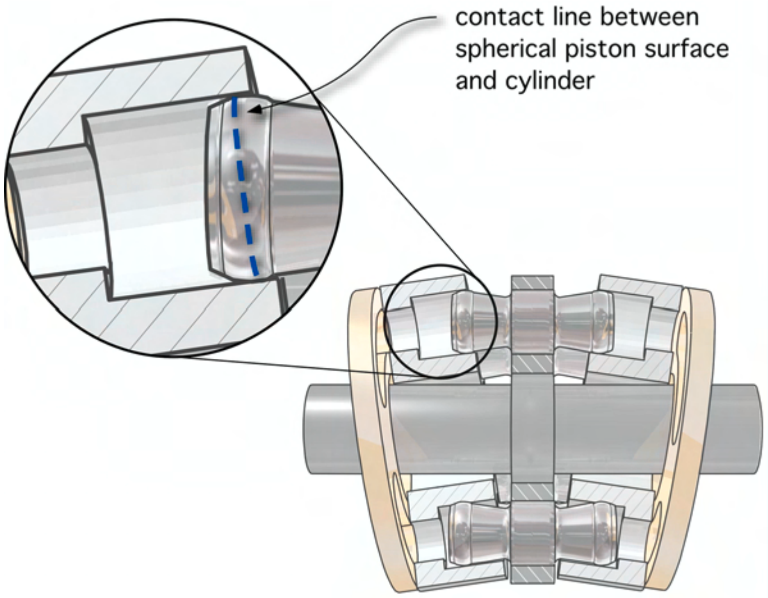


Figure 7: Line of contact between piston and cylinder wall (no piston rings applied).

5 Fixed pistons and moving cylinders

Giving each piston its own cuplike cylinder instead of having a collective cylinder block can solve the dilemma. The construction is illustrated in **figure 8**. The cylinder cups are free to move on the rotating barrel plane thereby compensating for the elliptic deviation between the rotor plane and the barrel plane.

In the axial direction the cylinder cups are slightly under balanced and the remaining force keeps the cups pushed against the barrel. The friction losses are minor because of the small relative movement and the light axial force between the cylinders and the barrel.

Essential for the new concept is the balancing of the piston rings. By cutting back the rings at the inner bottom part (as is shown in **figure 9**), the rings can be partially balanced. As a result only part of the ring will expand. Due to the tilting angle of the barrel and the cylinders this part is always pointed in the direction of the BDC-

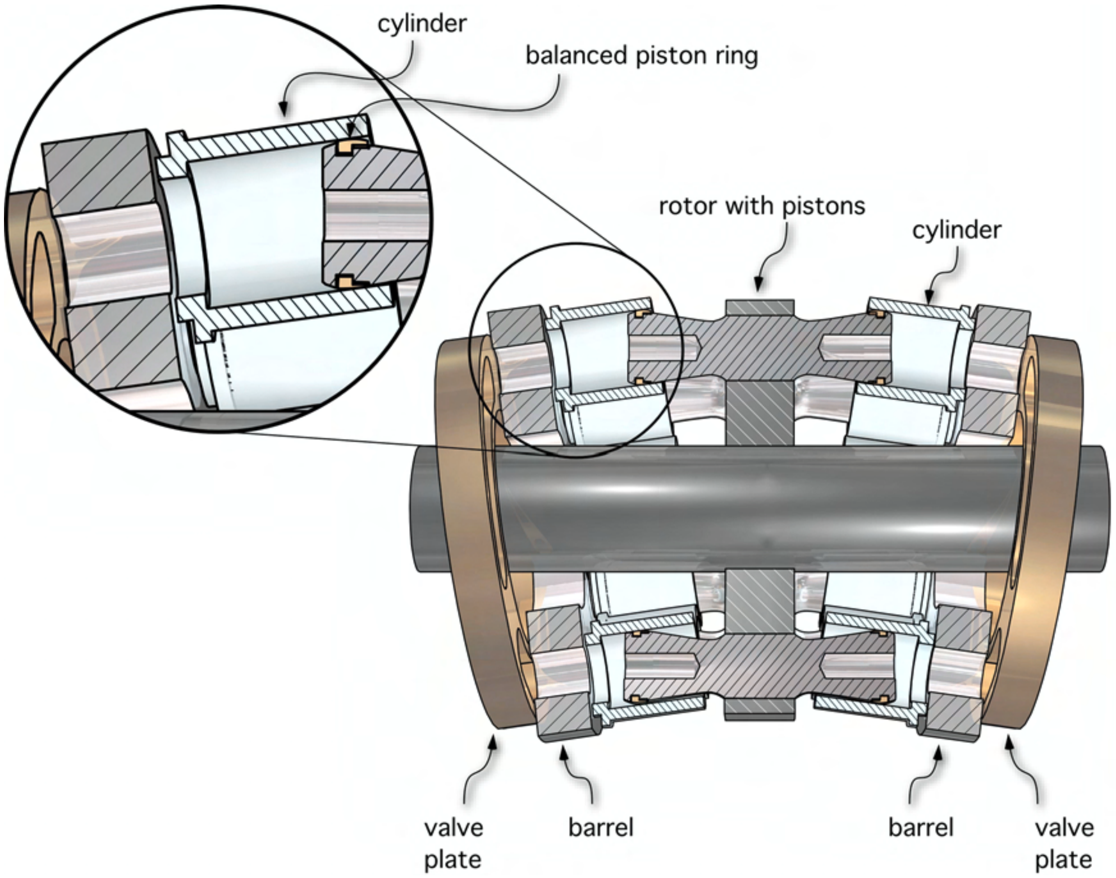


Figure 8: Axial piston principle with floating cylinders and balanced piston rings. The barrels only have to fulfil the sealing function for the valve plate. The bearing and mechanism for rotating the barrels is not shown.

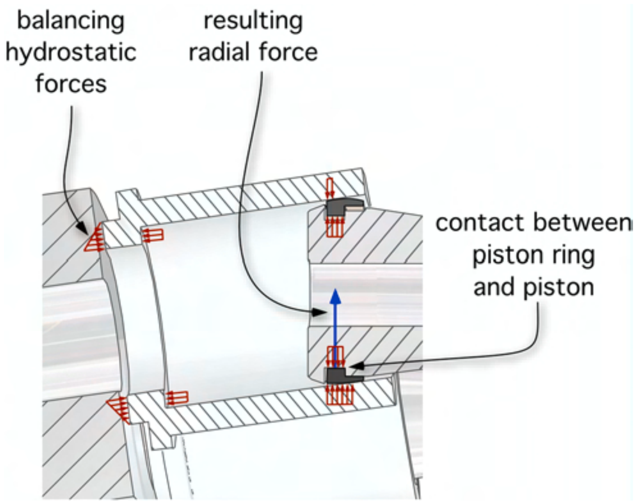


Figure 9: Detailed cross section of floating cylinder cups, fixed pistons and balanced piston rings.

position (straight up in figure 9). Since the piston rings and cylinders are free to move to some extend, the total package of ring and cylinder will move upward, thereby closing the gap at the other side of the piston ring. The movement will stop when the piston stops the ring-cylinder-package. This way the radial force created by the angle between piston and cylinder will directly be supplied to the piston, thereby to a large extend avoiding the loading of the cylinder wall that would otherwise be the case.

There are several ways to synchronize the rotation of the two barrels with the rotation of the rotor, the pistons and the cylinders. The most convenient way is to utilize the rotor shaft that is going through the barrels and to connect them to the shaft by means of for instance a spline coupling. The total concept is illustrated in **figure 10**. The torque needed to drive the barrels is small since it only involves the friction between the barrel and the valve plate and the inertia of the barrel. Because of the coupling between the barrels and the shaft even this torque is not transferred through the cylinders.

From a production point of view the construction with the cylinder cups avoids the stacking up of production tolerances. Because of the free positioning of the cups each pair of piston and cup is separated from the other pairs. The construction furthermore avoids the long pistons needed in slipper type machines. Also the structural loading of these pistons is avoided. Compared to the bent axis machines the large bearing is taken away. This is especially important for the design of the transformer since the total load from the supply and the load kidney can be much higher than in pumps and motors.

Despite the short stroke of the pistons the new transformer has a high power density (see size comparison in figure 10). This is especially due to the smaller bearings for the new IHT. At this time the new IHT is designed for a maximum speed of 8000 rpm, the same as for the original prototype, which was based on a 10 cc/rev bent axis motor. It is however feasible to extend the speed range of the new transformer, thereby increasing the power density of the new unit.

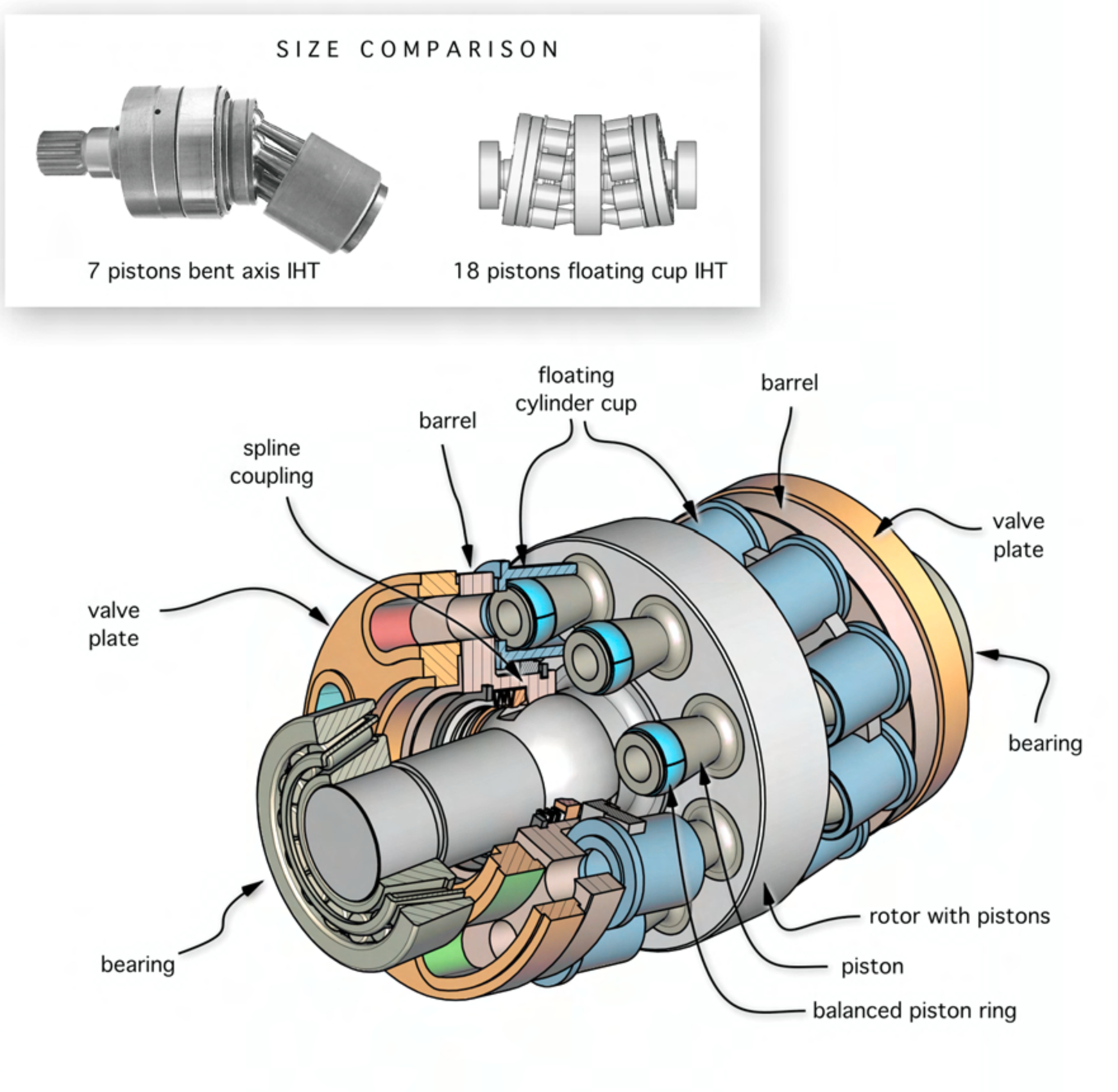


Figure 10: Total section of the most important parts of the new IHT design. On top a comparison of the rotating parts is shown. The capacity of the new IHT design is equal to the original bent axis machine.

An important advantage of the new design is the relatively large cylinder diameter and short stroke. Moreover the total flow is now divided over 18 cylinders. As a result the port opening losses are now strongly reduced. Ironically the port opening losses are now decreased to a point were shuttles are no longer a necessity. **Figure 11** shows the calculated pV-diagram both for the original bent axis IHT as for the new

18-piston-IHT. In the original bent axis design without the shuttles, pressure spikes up to 800 bar occur. In the new design (also without the shuttles) these pressure peaks are diminished to acceptable levels. It is however expected that the shuttles are again indispensable if the maximum rotational speed would be increased.

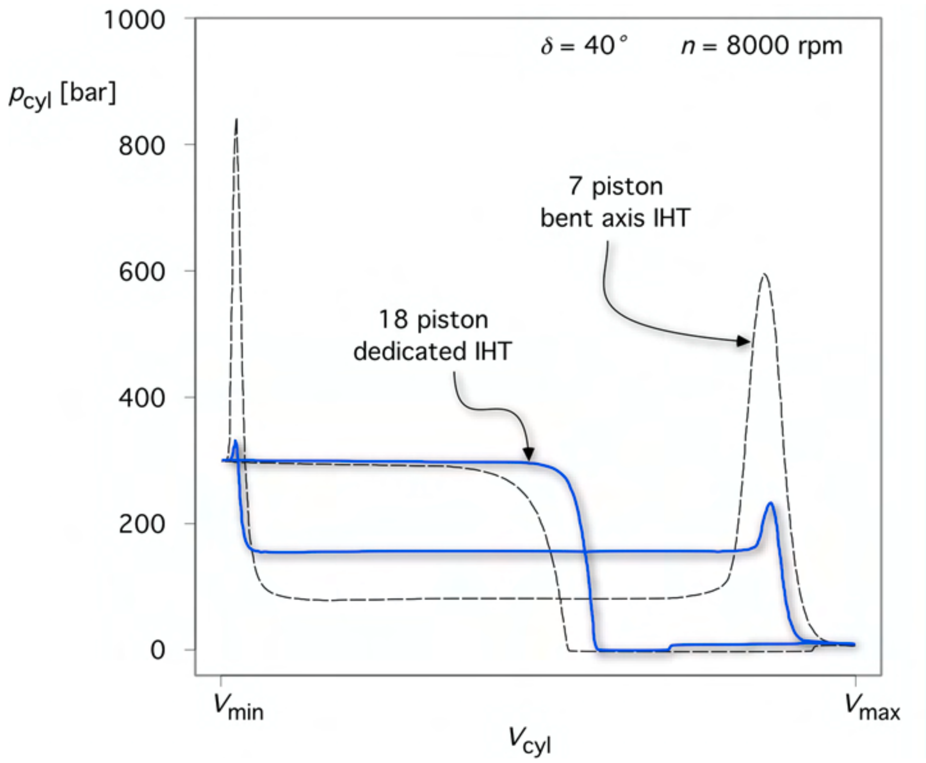


Figure 11: Calculated pV-diagram for an IHT with 7 pistons based on the bent axis motor as well as for the new 18-piston-IHT. In both cases there are no shuttles applied. (Control-angle port plate δ = 40°, rotational speed = 8000 rpm)

6 Evaluation of the new concept

Axial piston pumps and motors generally have 6 to 9 pistons. Although a larger number of pistons has advantages /4/ these are not offset by the cost increase. It is estimated that doubling the number of pistons increases the manufacturing costs by about 15%. It is also generally accepted that, in order to increase the power density, the swivel angle of the unit and thus the piston stroke should be as large as possible.

In this paper a new axial piston principle is introduced for the Innas Hydraulic Transformer in which none of these ‘rules’ is followed: the number of displacement volumes is increased to (at least) 18 and the swivel angle is reduced to about 9°.

Furthermore the pistons are rigidly connected to a rotor, thereby being arranged in double ring configuration having one ring of pistons on each side of the rotor. Essential for the new design is the introduction of free moving cuplike cylinders. Instead of having a single barrel with multiple cylinders, in the new concept each piston has its own cylinder. With these cups and the application of balanced piston rings the friction between the piston (ring) and the cylinder can be minimized. This is of great advantage, not only for reducing the friction losses but also for creating a low start-up torque.

The higher number of pistons i.e. of displacement volumes strongly reduces the flow and torque pulsations of the transformer. On the load side the flow pulsations are even reduced up to 90%. This is especially important since on the load side accumulators cannot be applied and the system stiffness is highest. The combination of the reduced torque pulsations and the reduced friction will strongly improve the start-up behaviour of the transformer. Furthermore it is expected that the reduction of the structural load as well as the reduction of the flow pulsations will lessen the noise produced by the transformer.

Although the new axial piston principle is designed for the hydraulic transformer the advantages of the new principle are also valid for pumps and motors and there are no reasons why this principle could not be applied also for designing other hydrostatic machines. It should even be possible to make the tilting angle of the barrel variable and thus change the displacement volume of the unit. Or it might be possible to change the flow rate by means of rotating valve plate, similar to the IHT.

It should be mentioned that the new principle is the theoretical outcome of a design study: it has not been tested so far. Although the torque and flow pulsations can be calculated with a high accuracy it is almost impossible to calculate the effects of the new design on the noise level. Furthermore it remains to be seen what the cost consequences of the new principle are. These and many more questions will have to be answered in the near future when a first model of the new transformer will be produced and tested.

7 References

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