

Efficiency by design: Piston pumps and motors with predefined tribological systems enable prediction and optimization of losses and efficiency

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Predefined tribological systems enable the designer of hydrostatic pumps and motors to leave the domain of mixed friction contacts and obtain a new way of design. The locations of force conduction and gap sealings evolve from incalculable subsystems into machine elements with predictable and computable properties.

The method is displayed and validated by the example of the RAC (radial piston machine) development and leads to a novel concept for an axial piston pump and motor that promises a favourable efficiency chart and excellent start-stop-properties (patent pending).

Keywords: Hydrostatic pump and motor, friction, tribology, energy losses, efficiency, hydrostatic bearing

Target audience: Manufacturer and designer of hydrostatic pumps and motors, innovation scouts

1 Introduction

In all types of hydrostatic pumps and motors efficiency is determined by the friction losses in the tribological systems (tribosystems) and by the volumetric losses through small gaps, and usually these gaps are a part of the tribosystems too.

In the usual piston machines, as swashplate, bent axis or radial piston type, the tribosystems are rather complicated. They combine Coulomb friction, Newtonian friction, the variation of viscosity by pressure and temperature and thus influence the gap between the sliding parts in a very confusing way. And the forces transferred by these contacts are complicated too, changing with pressure and speed and often superposed by lateral and rotating loads.

Hence, the calculation of gap height as an indicator for volumetric losses as well as for the type and amount of friction demands complex and laborious simulations that hardly can be done within the design process. And even in the optimization of established pump types it means a lot of sustained work in calculation and validation.

2 State of the art

2.1 Tribocontacts in swashplate pumps

Having a look at axial piston swashplate pumps, the hydrostatic force is conducted first from the pressure to the piston body, then to the piston slipper with its ball joint and further on to the swashplate. There the force is divided into the normal force and the lateral force, which tries to slide the slipper “downhill”.

As the lateral movement is only possible in accordance to the rotation of the cylinder barrel, the driving forces have to pass back to the piston and are transferred as lateral loads from the piston to the cylinder wall. The sum of cylinder side loads finally creates the torque, which is transferred to the pump or motor shaft.

This is rather complex and it causes high side loads to the cylinder barrel itself and affects the surface contact and sealing of the cylinder barrel on the port plate.

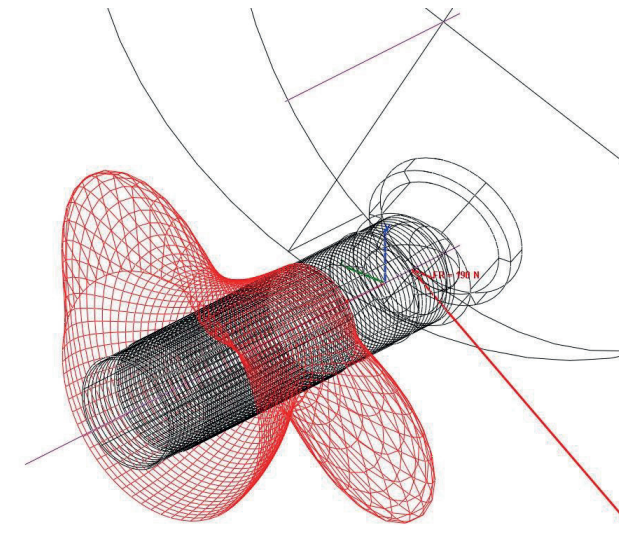


Figure 1: Lateral load on a swashplate type piston /2/

Gels /2/ has made investigations about the tribosystem between piston and cylinder and describes the complex correlations of speed, gap heights and the lateral forces, partly transferred by fluid pressure and partly by material contact. And he proves that a well-working piston-cylinder system is the result of the running-in wear of the cylinder bushing.

The tribosystems of the ball joint and the piston slipper are complicated in a similar way. The high forces that have to be conducted need hydrostatic discharge to a percentage over 90% of load. The remaining load is transferred as mixed friction in a combination of material contact and fluid lubrication with gap heights between 1 to 5 micrometers.

Under these circumstances, the design and optimization of axial piston swashplate machines either is a process with a lot of experience knowledge and virtuosity, or it has to depend on highly sophisticated three-dimensional simulations of several linked subsystems, working each under elasto-hydrodynamic conditions.

2.2 Description of tribocontacts

A first look at tribocontacts leads to the Stribeck-curve that shows the transition between boundary friction and hydrodynamic friction depending on the speed of motion, as it is known from typical radial bearings with oil lubrication (Fig. 2).

Hydrodynamic friction is generated by high velocity, high viscosity of lubricant and moderate load. It is promoted by convergent gaps that taper in the direction of movement. Some of these conditions are not typical for tribocontacts in hydrostatic machines.

The gap between the sliding partners is dependent on speed, viscosity and load too. In plane bearings of hydrostatic machines it hardly exceeds 6 micrometers.

At high pressure, you have also to consider elastic deformation of the bearing surfaces that may occur in a dimension comparable to the gap height. Sometimes this may help by creating local convergent gaps, but in other cases deformation just causes boundary contact and attrition.

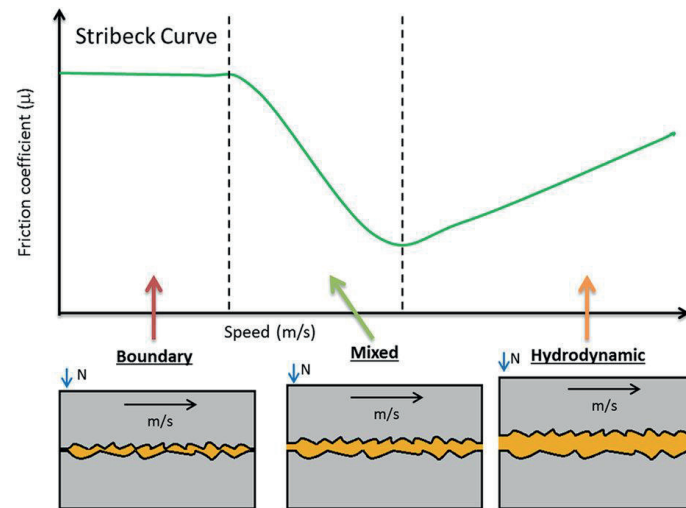


Figure 2: Stribeck curve and types of friction

2.3 Calculation of tribocontacts

In boundary and mixed friction situations the friction force F_{RC} is a function of normal force F_N and friction coefficient μ :

$$F_{RC} = F_N \cdot \mu \quad (1)$$

With complete hydrodynamic support, the friction force F_{RH} changes and becomes a function of viscosity η , velocity v , gap height h_s and contact area A :

$$F_{RH} = \frac{\eta \cdot v \cdot A}{h_s} \quad (2)$$

Please consider that hydrodynamic friction force is proportional to velocity and independent from the normal load F_N as long as the gap height h_s keeps constant. However, in normal mixed friction systems you will find a interdependence between normal force F_N and gap height h_s .

In both cases there may occur a leakage flow Q , caused by a pressure difference Δp along the gap length l . The gap is defined by its width b and the height h_s :

$$Q = \frac{\Delta p \cdot b \cdot h_s^3}{12 \cdot \eta \cdot l} \quad (3)$$

3 Predefinition of tribocontacts

A distinct way to manipulate tribocontacts is to ensure hydrodynamic conditions in all operating points and to control the gap height. To attain this we use the principles of hydrostatic bearings with upstream restrictors. The

most heard prejudice against hydrostatic bearings is their energy consumption. We try to clear that up in the following chapters with experiences we made in the development of the RAC machine.

3.1 Tilt piston design

The first condition to achieve predictable conditions in tribosystems is to find a design with clearly arranged forces, as we achieved in the draft of the RAC radial piston pump (Fig.3). It is based on the principle of simplicity in the conduction of forces and a self-supporting drivetrain with hydrostatic bearings.

The radial cylinders contain one-piece-pistons with a linear sealing orthogonal to the piston axis. The pistons can be tilt against the cylinder axis. By this, the unbalanced pressure fields on the cylinder wall directly generate the torque to the cylinder barrel and the shaft /1/.

The piston itself is not involved in the conduction of driving forces, it just has to balance the hydrostatic forces to the cam ring and provide the movement of the slipper with a minimum amount of losses.



Figure 3: RAC principle: Radial pump with axial portplates

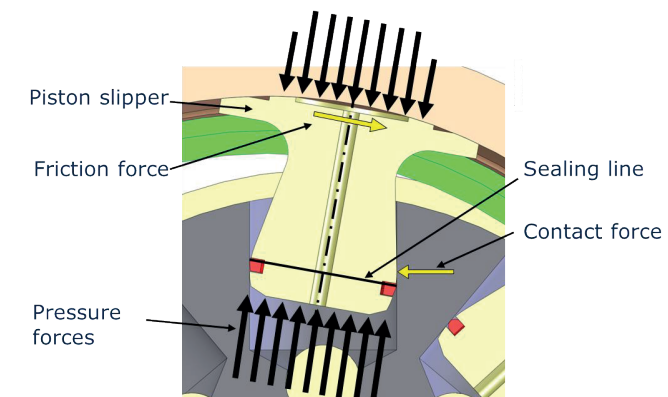


Figure 4: Tilt piston design

3.2 Piston slippers on tilt pistons

The design with orthogonal forces enables a peculiar design of the piston slipper. A laminar restrictor between the inner and outer end of the piston enables a design with a reduced pressure $p_2 < p_1$ which acts on an enlarged surface $A_2 > A_1$.

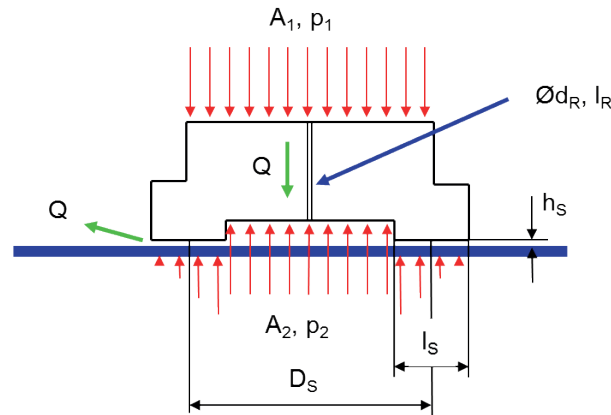


Figure 5: Tilt piston design, pressure fields and dimensions

$$h_S = \sqrt[3]{\left(\frac{A_2}{A_1} - 1\right) * \frac{3 * l_S * d_R^4}{32 * l_R * D_S}} \quad (4)$$

In comparison to the conventional hydrostatic unloaded slippers in swashplate machines, this slipper contains a real hydrostatic bearing with an upstream restrictor and creates a self-regulating gap height h_S .

It shows the typical characteristics of hydrostatic bearings: no material friction, high load stiffness and a gap height depending on the parameters of design (Equation 1). Please notice that the gap is a function of several design parameters but independent from the working pressure p_1 .

By tuning the gap height, both aspects of energy loss can be influenced:

- the volumetric loss Q is proportional to the pressure p_2 and the gap height $(h_S)^3$
- the friction loss (Newtonian friction force) is proportional to the relative speed, the contact area and inverse to the gap height

In our experimental work with the RAC design, we tuned the gap height by changing the restrictors diameter d_R or length l_R . Increasing the gap means increasing the volumetric loss Q , but also decreasing the Newtonian friction force especially at high rotational speeds.

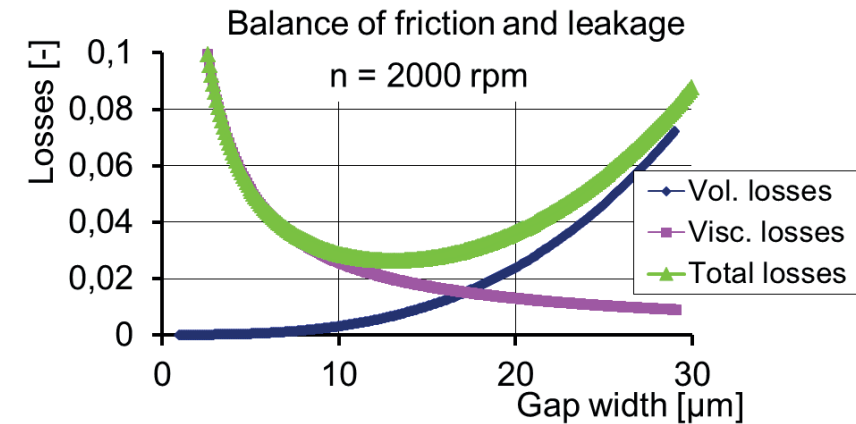


Figure 6: Energy consumption of piston slipper /1/

The sum of losses describes a graph with a distinct minimum. For most operation conditions, we found the minimum sum of losses (or the best efficiency) with gap heights between 10 to 18 micrometers.

A side benefit of the hydrostatic bearing is that the slipper is totally free from friction at low speed. So it is best fit to start-stop-cycles. And in case of lateral forces, it works self-stabilizing (Fig. 7) like a “hydrostatic weeble”.

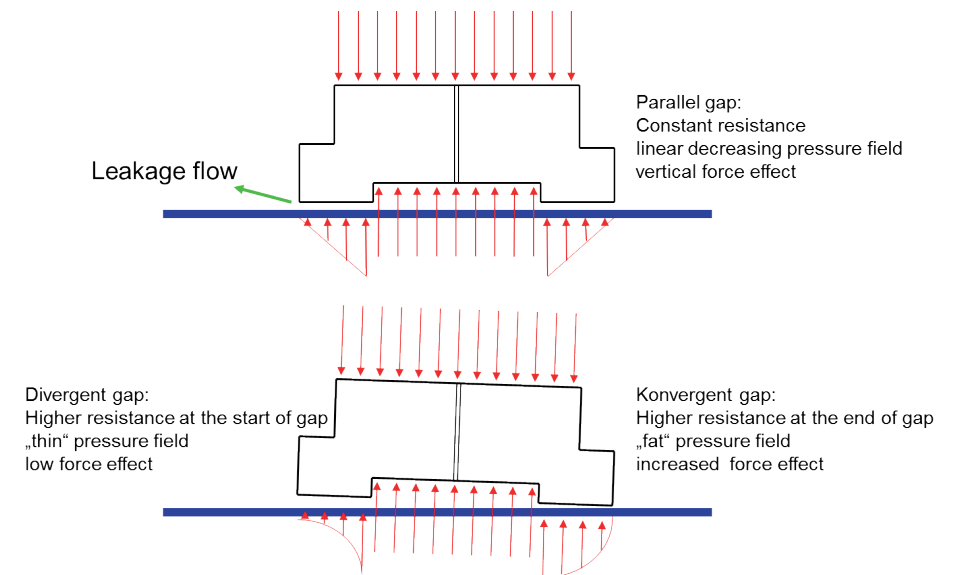


Figure 7: Self-stabilization of tilt piston slipper

In summary, the novel piston design enables us to manipulate the characteristics of the tribocontact to a system free of boundary friction and to optimize the gap height to a minimized sum of losses, all with the objective to optimize the efficiency of a hydraulic pump or motor in the main area of operating points.

3.3 Portplate design

A very similar way of modification is possible in the contact area between cylinder barrel and port plate. Achten /3/ describes a system of pockets and grooves that influence the gap, reduce friction and enhance efficiency.

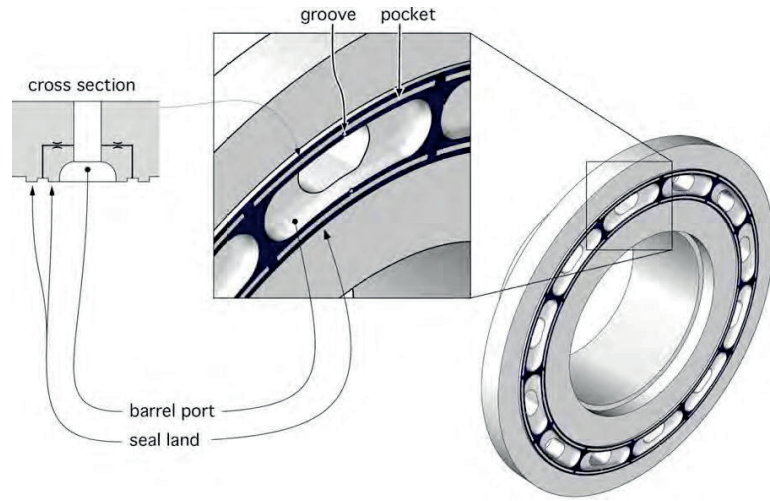


Figure 8: Port plate with pockets /3/

We tested this system in the RAC experiments and approved the reduction of friction at the port plates. And we found that a further stage of this idea enables the tuning of the port plate gap comparable to the method used in the slipper design.

4 Calculation of losses and efficiency

Starting from the premise, that all tribocontacts in a hydrostatic piston machine have the depicted characteristics and can be influenced in the gap height, the volumetric and hydromechanic losses of the whole machine can be computed by summarising the particular elements:

$$Q_{loss} = \sum_i \frac{\Delta p_i \cdot b_i \cdot h_{si}^3}{12 \cdot \eta \cdot l_i} \quad (4)$$

$$M_{RH} = \sum_i \frac{r \cdot v \cdot \eta \cdot A}{h_s} = \sum_i \frac{2 \cdot \pi \cdot r^2 \cdot n \cdot \eta \cdot A}{h_s} \quad (5)$$

From these equations for the volumetric loss and loss momentum you can derive the power equations

$$P_{loss,vol} = p \cdot Q_{loss} = p \cdot \sum_i \frac{\Delta p_i \cdot b_i \cdot h_{si}^3}{12 \cdot \eta \cdot l_i} \quad (6)$$

$$P_{loss,hm} = M_{RH} \cdot 2 \cdot \pi \cdot n = \sum_i \frac{4 \cdot \pi^2 \cdot r^2 \cdot n^2 \cdot \eta \cdot A}{h_s} \quad (7)$$

In the design, analysis and testing of the RAC we used these equations in a spreadsheet tool and found a good accordance to the test data. Of course there were some other elements that caused losses, like ball bearings, seal rings and piston seals, but these effects proved to be inferior to the depicted energy losses.

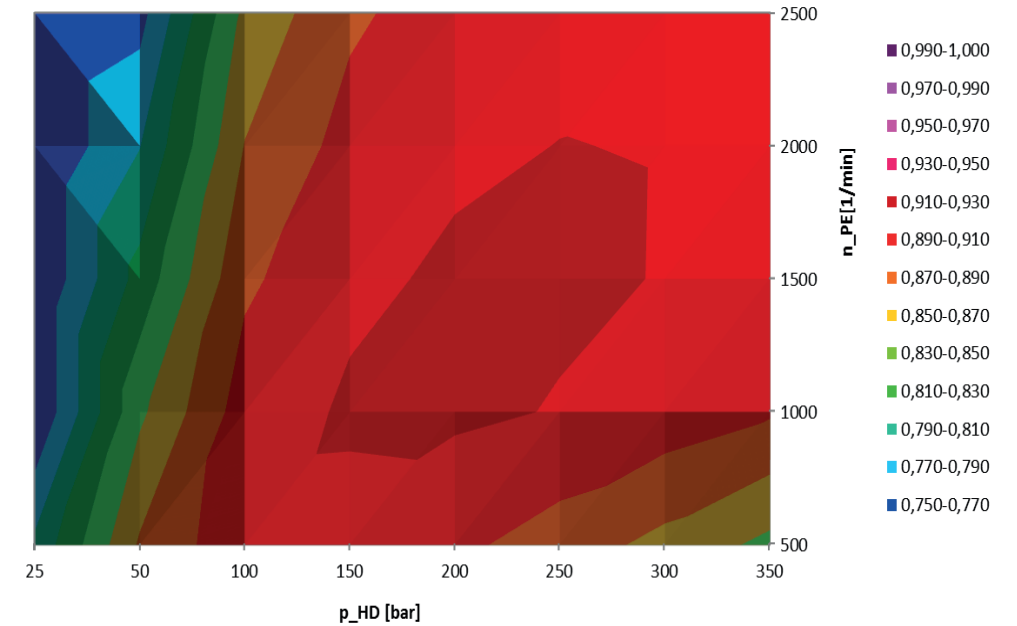


Figure 9: Calculated overall efficiency map of RAC pump (displacement 40 cm³)

Figure 9 shows a diagram of overall efficiency for the RAC that is calculated on the base of the equations (6) and (7). It includes the portplate losses, the slipper losses and small amounts of hydromechanic losses for the shaft bearings and sealing as well as for the linear piston sealings. It also includes terms for the pressure-induced change of gap heights as they occur by elastic deformation of sliding surfaces, e.g. at the port plate gap.

The comparison to the measured efficiency chart (Fig. 10) illustrates that the chosen way of computing leads to a rather good accordance with reality. In contrast to the simulation of tribocontacts or complete pumps the calculation allows a fast and simple evaluation of the effect of input parameters such as viscosity, geometric values or restrictor characteristics.

Thereby the designer is able to align the area of best efficiency to the main operation conditions and quote the effect of changes in temperature or viscosity. Additionally, he gets reliable information about the amount of lost energy for each spot of the machine and is thus able to make targeted modifications.

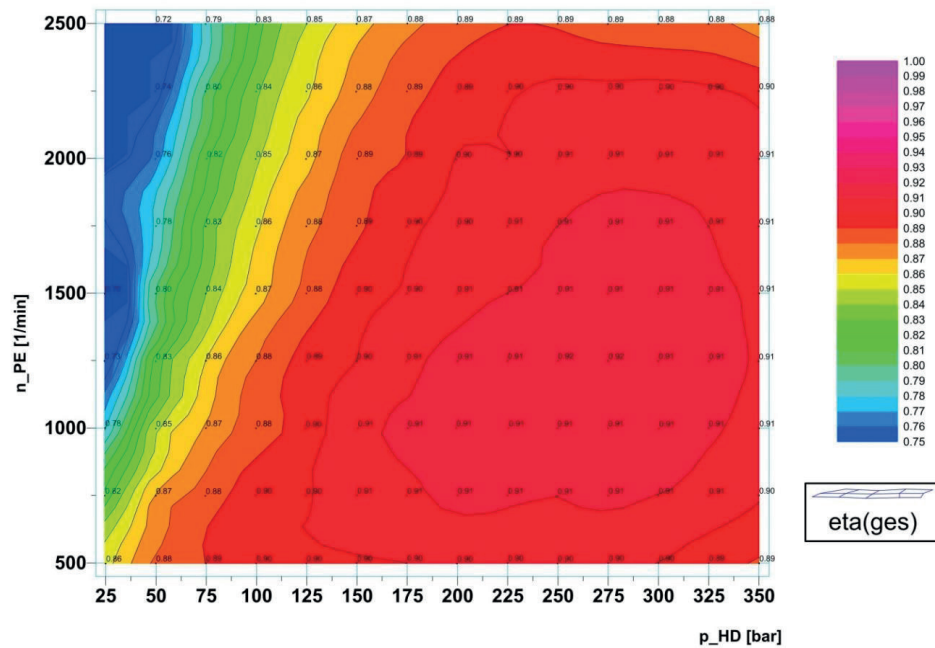


Figure 10: Measured overall efficiency map of RAC (displacement 40 cm³)

With this method, we got an intense review over the energy losses of the RAC pump and the untapped potential for the optimization of efficiency. And we could show that within a short period of development we could reach an overall efficiency equal-ranking to the best models of axial piston swashplate pumps.

5 Design of a novel axial piston machine

Once we had grasped the system of losses and efficiency, a new idea formed itself within the heads:

How about creating a new type of pump, that leaves out the difficult parts of the RAC design as the double portplate, the spherical surfaces and the rather complex axial compensation?

This hydrostatic pump or motor should use the ideas of simple conduction of forces and of hydrostatic bearing with controllable gap height in the same way as in the RAC concept, but with less parts and easy production technologies.

The novel machine type has a cylinder barrel, swashplate and portplate just as a normal axial swashplate machine (Fig. 11), but with a rather short cylinder barrel. The pistons are one-piece with the slippers and are tilted against the cylinder axis, fitting to the position of the swashplate.

The piston sealing ring provides a tilted limit line for the pressure field on the cylinder wall, thus creating unbalanced forces to the cylinder which are added to the driving momentum and conducted to the shaft. These characteristics have been well proven in the RAC development.

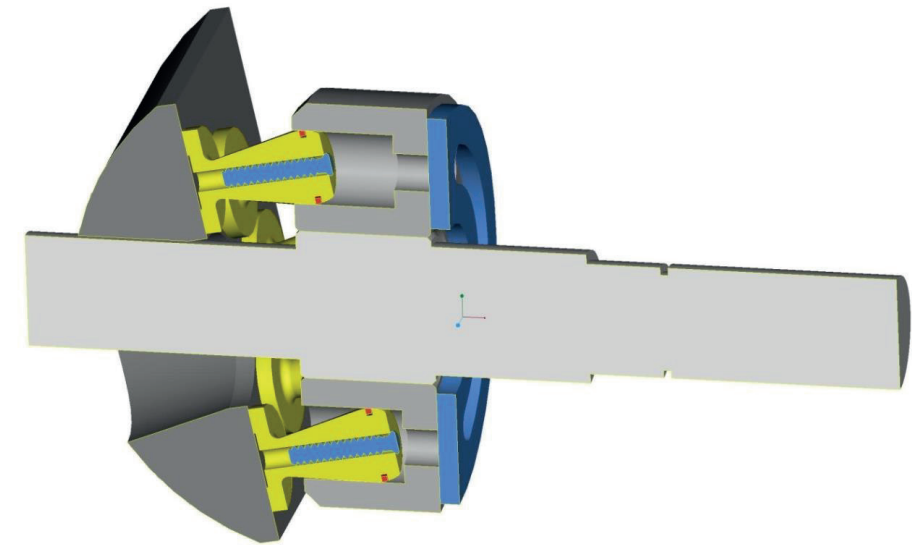


Figure 11: Principle of the axial tilt piston machine (patent pending)
For better understanding two cylinders are shown in the cross-section

The pistons are sliding on the swashplate with hydrostatic bearings of the type described in Fig. 5. The upstream resistor is accomplished by a pressed-in part with a helical channel and a filter screen. As the slipper movement has no boundary friction and the piston has no lateral forces to the cylinder wall, this machine is best fitted to start-and-stop conditions.

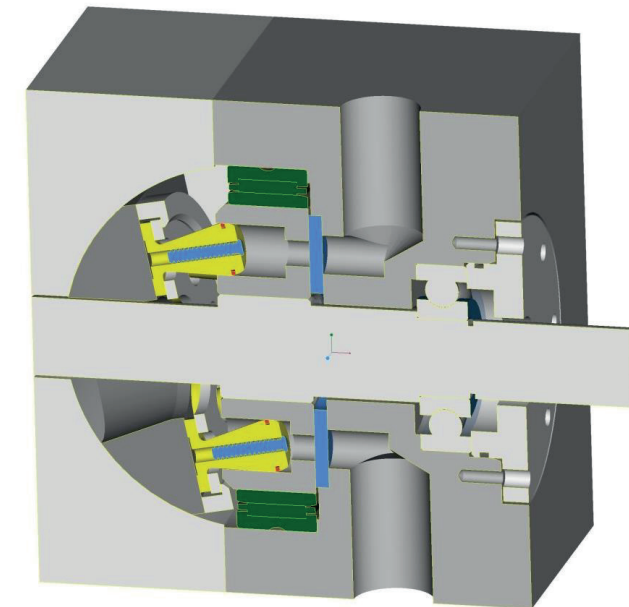


Figure 12: Axial tilt piston machine

The hydraulic side forces to the cylinder wall are centred in the barrel, so they can be supported by a roller bearing (Fig. 12). Thus, the forces do not cause any misalignment between the barrel and the portplate. The design is completed by a conventional downholder for the piston slippers and a preload-spring for the barrel-portplate-contact (not shown in Fig. 12).

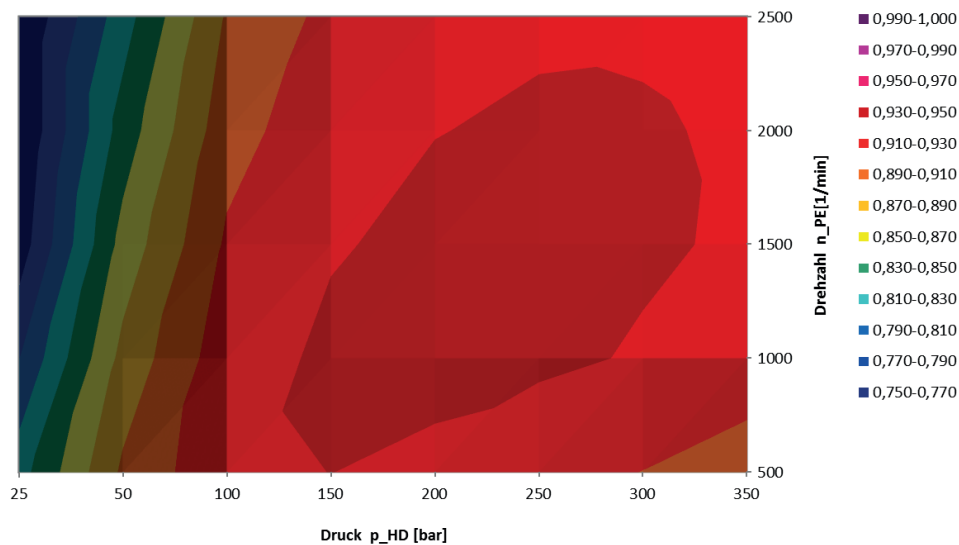


Figure 13: Efficiency pre-calculation for the axial tilt piston pump (displacement 40 cm³)

The pre-calculation of the losses and overall efficiency helps designing the tribocontacts and shows a very interesting efficiency-chart.

6 Summary and Conclusion

The presented concept of design avoids mixed friction tribocontacts in hydrostatic machines. By simplification of the force guidance it enables the use of hydrostatic bearings with upstream restrictors in the piston slippers and even in the portplate. Thus, the tribocontacts are free from boundary friction and wear, and they become computable. The calculation and summation of volumetric and hydromechanic losses for each tribocontact enables the depiction of a complete efficiency chart within seconds.

The possibility to evaluate the different loss elements permits the designer to make targeted modifications. With this method, he can gain an intense review over the energy losses of hydrostatic machines, their dependencies to geometric and fluid characteristics and the untapped potential for optimization.

As a result of the experiences with this method of design a novel type of an axial tilt piston pump or motor is presented.

Nomenclature

Variable	Description	Unit
A	Area	mm²
F	Force	[N]
μ	Friction coefficient	[-]
η	Dynamic viscosity	[Ns/m²]
v	Velocity	[m/s]
h _s	Gap height	[mm]
Δp	Pressure difference	[MPa]
b	Width of a gap	[mm]
l	Length of a gap or restrictor	[mm]
d, D	Diameter	[mm]
r	Radius	[mm]
M	Rotational moment	[Nm]
P	Power	[Nm/s]
Q	Volume flow	[m³/s]

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