Experimental Validation of a New Design Concept for the Increase of Efficiency of the Hydraulic Drive System in Mobile Working Machines

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Abstract. In mobile working machines like excavators, hydraulic actuators and drives are indispensable due to their advantages of a high power to weight ratio, robustness and the good power transmission over medium distances by the hydraulic fluid. Axial piston machines can deliver a large volume flow at a high operation pressure with high efficiency and are therefore widely used as system pump or drive motor. The three essential tribological contacts piston / bushing, sliding shoe / swash plate and valve plate / cylinder block strongly influence the machine’s efficiency and so the complete drive system. The paper on hand focuses on the experimental validation of a new design concept to increase the efficiency of the tribological contact between valve plate and cylinder block in an axial piston pump. Due to the different pressure forces at the high- and low-pressure side, the cylinder block tilts and holds this preferred position almost constant. Therefore, the gap height is not constant. In the area of minimum gap height, the danger of solid body contact increases. In addition, the heat dissipation is decreased, which can lead to local constant high temperature hot-spots. Both can destroy the surface structure or the coating. A new design concept has been developed to reduce these risks, thus increasing the efficiency, the lifetime and making it possible to dispense of leaded coating materials. In this design concept, pressure pockets are added in the area of minimum gap height. The simulation study shows a high potential of this concept. The experimental results of the prototype parts, which are measured on a special test rig and presented in this paper, confirm the simulations and validate the potential of the new design concept in this way.

1. Introduction

Mobile working machines are used worldwide in several areas, for example construction, mining, agriculture and forestry. Due to their advantages like a high power to weight ratio, robustness and the easy realization of linear actuators, hydraulic drives are the main drive technology in mobile working machines like excavators. The state of the art is that an internal combustion engine drives a hydraulic pump, which generates the hydraulic power and delivers it via the working fluid to the various actuators. Axial piston pumps are widely used because they can deliver high volume flows in high pressure operating points with an overall efficiency of about 90 % in modern units [1]. Figure 1 shows an axial piston pump with its main components. The losses of the three essential tribological contacts piston / bushing, sliding shoe / swash plate and valve plate / cylinder block [1] determine the efficiency of these pumps in a high degree. Research to improve these contacts is therefore of great importance for the
efficiency of mobile working machines in the context resource efficiency, climate protection and the emissions law.

Leaded alloys are used in the tribological contacts due to their emergency running properties and to compensate manufacturing errors. One part is therefore made of steel, while the other is coated with brass or bronze alloys. However, lead is a toxic heavy metal and its substitution is one of the main topics in hydraulics.

The topic of the here presented work is the tribological contact between the valve plate and the cylinder block. The drive shaft transmits the rotation movement to the cylinder block, which is pressed on the valve plate by the pressure forces and by springs to stabilize it and avoid lifting. The tribological system has two main tasks. The first is to minimize volumetric losses by sealing the high- and low-pressure side from each other and from the housing. The oil film in the gap fulfills the second task, to protect the contact surfaces against wear and to reduce the friction losses to a minimum. It lubricates the contact surfaces, dissipates heat and particles, and compensates the majority of the axial forces by hydrodynamic effects. However, this results in a conflict of objectives: The gap height needs to be small enough for low leakage but not too small, as this reduces heat and particle dissipation and increases wear. The perfect condition would be a small and even gap height over the complete contact area.

However, the wear pattern on the valve plate is not uniform. Early in the development of axial piston machines, it was assumed that the cylinder block tilts during its operation. This increases the load and thus the local wear. P. Achten investigates the tilting effect in [3]. He analyses and calculates the forces which are acting on the cylinder block in operation. By setting up and solving the equilibrium of these forces and torques he showed that they are unbalanced. On the one hand, the displacement chambers are filled with oil differently depending on the angle of rotation, so that the applied centrifugal forces are also uneven. On the other hand, the different compressive forces resulting from the narrowing of the cross-section are significantly higher. Here the difference is significant, especially in operating points with high load pressure, since a pressure of several hundred bar acts on the high pressure side, while the low pressure side is preloaded to reduce the tendency of cavitation. Because of this, a tilt torque to the high pressure side results. S. Wegner build up a special simulation model [4], which is briefly explained later in this paper, to investigate tribological contact of valve plate and cylinder block. In his simulation studies, he confirmed the results of the analytical investigation of P. Achten. The simulation results show that the cylinder block tilts rapidly to the high pressure side and consistently holds this preferred position steady. Figure 2 shows the position of minimum gap height when 4 or 5 pistons are pressurized with high pressure. Due to the earlier explained negative effects of a gap height that is too small, the danger of solid body contact and the danger of a local constant high temperature hot spot is increased in this area [5]. Both can destroy the surface condition or coating.
The new design concept with additional pressure pockets in the valve plate’s high pressure kidney, which is presented and validated experimentally in this paper, reduces the tilt angle of the cylinder block. This reduces the mechanical and thermal load significantly in the area of the minimum gap height. So, it could be the key for the implementation of new tribological optimized coating materials without lead.

2. State of the Art

For a long time, the development process focused mainly on the two problems, to avoid the lifting of the cylinder block and to improve its centering. In the context of the present study, different approaches have already been pursued. In [6], punctual pressure fields are positioned in the outer support ring and connected to the high-pressure kidney by means of holes. This supporting pressure field reduces the tilting of the cylinder block and thus increases the self-priming speed by 15%. The disadvantage of this concept is the high additional production-related expenditure. The connecting bores must be drilled from the outside into a round surface. In addition, a thread must be produced for sealing with a plug. The actual support bores would also be difficult to produce in the case of a spherical valve plat, but they also represent an additional expense in the case of flat ones. In addition, the sealing surfaces would have to be designed accordingly wide. A completely different concept is patented in [7]. Here, an additional device prevents the piston drum from tilting. Compensation pistons, which are connected to the flow channel through a bore, generate a moment against the tilting. A new approach with an additional relief field in the control mirror is patented in [8]. This relief field is located in the area of the high pressure side. In this approach, the pressure field is externally pressurized. The supporting torque can thus be controlled independently of the operating point. Apart from the additional manufacturing costs, the main disadvantage of this approach is the additional components of the control system. For stable operation, the tilting over the entire characteristic diagram must also be known. S. Haug extends this approach and implements a further pressure field on the low-pressure side [9][10]. In his investigations, he also applies this concept to other tribological contacts and develops an operating point-dependent control. By means of this control, tilting of the cylinder block can be prevented almost completely over the entire characteristic diagram of the pump under investigation. The disadvantages of this concept are the additional components and the complex control design.

3. Concept Development

The research at ifas combines the experimental and the simulation investigation of the contact between the cylinder block and the valve plate in an axial piston machine. By combining simulation and experimental results, the effect of the new design concepts of optimized additional pressure pockets can
be thoroughly analyzed. The simulation tool, as well as the test rig, were specifically designed for the measurement of the cylinder block movement.

3.1. Simulation Model
To analyze the tribological contact between the valve plate and the cylinder block, a simulation model was set up at ifas. The simulation model is based on the model for the piston/bushing contact described in [11]. Its development a detailed description can be found in [4]. It will only be presented briefly in this paper. The simulation tool was designed with the aim of comparing several geometries at different operating points within an acceptable computation time and a sufficient accuracy. To reach this goal, the simulation tool includes the following physical effects:

- Hydrostatic and hydrodynamic pressure build up using the Reynolds equation
- Force and torque equilibrium on the cylinder block
- Squeeze film effect, viscous friction, microscopic part movement resulting from the first two points
- Macroscopic part movement, analytically derived
- Simple contact and part deformation model

The finite volume method is used to discretize the Reynolds equation. To calculate the gap height and the cylinder block movement, a linear system of equations using mass and momentum balances is set up. The Reynolds equation is added to the linear set of equations. Using the system parameters from the boundary conditions, from the geometry and the high- and low-pressure areas, the unknown variables and the number of equations are balanced, and the system can be solved numerically.

Solid friction is implemented by utilizing the contact pressure: The contact pressure at each finite area is multiplied with a constant friction coefficient (0.06), leading to the share of friction torque for each finite area. Figure 3 shows a diagram of the basic steps of the simulation model.

![Simulation Model](image)

**Figure 3:** Simulation model [12]

3.2. Pre-study on additional forces to prevent cylinder block tilting
The pre-study had the purpose of determining the optimal force distribution to avoid the movement of the cylinder block into a preferred position. For this, four scenarios were simulated to show the effect of adding forces which create a torque that pushes the cylinder block out of the tilted position. The complete study can be found in [12]. The first option is to add one or more forces in z-direction. The
second is to add those forces in radial direction. The added forces can either be static or dynamic. This totals in four different scenarios: a static force in z-direction, a dynamic force in z-direction, a static radial force, or a dynamic radial force. In the pre-study the static force in z-direction showed the greatest potential. Thus, the results for an additional static force in z-direction will be discussed here. The added forces are described by their position through the radius, angle and size. Table 1 shows the geometric parameters. The values are limited by the geometry of the valve plate’s kidneys. The inner radius of the kidney is 29.45 mm, the outer radius is 44.55 mm and the medium radius is 37 mm as shown in Figure 4.

![Figure 4: Geometric Parameters](image)

| Parameter | Value   |
|-----------|---------|
| d<sub>cb</sub> | 97.2 mm |
| h<sub>cb</sub> | 80 mm   |
| r<sub>1</sub> | 29.45 mm |
| r<sub>2</sub> | 44.55 mm |
| r<sub>i</sub> | 23 mm   |
| r<sub>m</sub> | 37 mm   |
| r<sub>o</sub> | 48.6 mm |
| γ | 225° |
| δ | 120° |

Table 1: Geometric Parameters

For the simulation, the added forces varied from 500 N to 2500 N in steps of 500 N. The change of the tilting behavior was simulated for the position at the inner, outer, and medium radii and for the angles of 120°, 135° and 150°. To portray the change of the tilt behavior, the relative change of the tilt angle was chosen as a comparative value. The tilting can be calculated with the gap height difference and the cylinder block’s diameter by equation 1. The relative angle change is the ratio of the tilt angle with the additional forces and the tilt angle in the basic configuration.

\[
\alpha = \sin^{-1} \frac{\Delta h}{d_{cb}/2} \quad (1)
\]

The added force has two effects on the movement of the cylinder block movement. The first effect is a moment which reduces the tilt angle and the second is that the force lifts the cylinder block. Both effects reduce the gap height. The simulation results in Figure 5 show that the minimum gap height
rises with the value of the additional force. A larger radius increases this effect because the moment against the tilt angle increases.

3.3. Design Concept

To implement the concept of an additional force on the valve plate, a design concept is developed. Adding forces in hydraulic machines can be achieved by additional pressurized surfaces. For the concept presented here, pressure surfaces will be added to the sealing gaps of the valve plate’s high-pressure kidney. Figure 6 shows the geometry of the sealing gaps for the considered area as well as the additional pressure pockets as CAD and the prototype used in the experiments. The position of the pressure surfaces should be located around 150°. This position is chosen because it is the angle at which the minimum gap height was shown to be the smallest (Figure 2). The potential size of the pressure pocket results in 3.325 mm towards the shaft and 0.925 mm towards the housing, which results in maximum Surface of 145.72 mm². By assuming load pressure of 250 bar a force of 3.64 kN is added. For the prototype, the surface of 101.01 mm² was added, because a chamfer reduces the sealing width in the valve plate of the considered pump. In this way, the sealing gap width is reduced by 2 mm.

Figure 5: Change of relative tilt angle adding a static force in z-direction

Figure 6: Geometry of pressure pockets a) CAD, b) manufactured
3.4. Simulation Results

To evaluate the impact of the pressure pockets on the axial piston pump, the previously described simulation model was used. Figure 7 shows the simulation results for the minimum gap height, the contact pressure, the solid friction torque and the tilt angle. The change of the values depends on the speed and the pressure of the axial piston pump. Therefore, the simulation results are shown in a three-dimensional surface plot. The mesh grid is defined by the speed of the pump in rotations per minutes (rpm) and the pressure on the high-pressure side in bar. The simulations points of the pump speed are 500 rpm to 2500 rpm in steps of 500 rpm and the high-pressure goes from 50 bar to 250 bar in steps of 50 bar. Each simulation is run for the values with and without the pressure pockets. The values of the relative change graph were calculated with the ratio of the total values.

The minimum gap height increases when the pressure pockets are applied. Especially when the rotation speed is low, the relative change of the gap height is high. The minimum gap height increases up to 3.6 % with the added pressure pockets. The contact pressure and solid friction torque show a high decrease when adding the surfaces. Both decrease by up to 30%. The tilt angle also decreases. Especially in the area of low rotation speed and high pressure. When the rotation speed is low and the pressure high, the risk of solid friction is the highest. This is due to the lubricating fluid film between the valve plate and the cylinder block not being fully developed yet. According to the simulation results, the concept of added pressure pockets to the valve plate has great potential.

Figure 7: Simulation Results

4. Experimental Validation

4.1. Test Rig

For the experimental investigation of the tribological contact to validate the simulation model, a special test rig was designed at the institute. In Figure 8, a picture of the constructed test rig and a circuit sketch can be seen. The setup of the test rig is also described in [13] but is further explained in the following.
The test rig is based on the test pump (p1) which consists of the engine of a serial pump with a displacement of 140 ccm, a maximum pressure of 340 bar and a maximum rotational speed of 2400 rpm as well as a hydrostatic bearing, which is described in more detail later. A force sensor measures the frictional force in the contact between the piston drum and the control mirror. The gap height as well as the movement of the piston drum is measured by 6 eddy current sensors. The pump is driven by an electric motor with a nominal power of 160 kW and a maximum rotational speed of 2600 rpm. The motor is connected to the test pump via a coupling. At the drive shaft, the rotational speed, the torque, and the position (angle of rotation) is measured by the sensors s3, s4 and s5 respectively. A centrifugal pump (p2) with a maximum volume flow of 250 l/min supplies the test pump with fluid. In addition, the pump preloads the low-pressure side to 10 bar. This ensures the reproducibility of the measuring points and reduces the risk of cavitation. In addition, the oil is filtered before entering the test pump (f2). The sensors s1 and s2 measure temperature and pressure on the low-pressure side. The pump p4 supplies the hydrostatic bearing with oil. The pressure can be adjusted manually via the pressure relief valve A. The leakage volume flow of test pump and hydrostatic bearing is discharged into the tank by pump p3. On the high-pressure side, the temperature and the pressure are measured by sensors s8 and s9. The pressure relief valve B is set to a value of 350 bar to protect the components behind it. The output volume flow is measured by means of the volume flow meter s10. The load valve (LV) consists of three proportional pressure relief valves with a maximum pressure of 350 bar and a maximum volume flow of 80 l/min. The operating pressure is set and controlled via a PI controller, which is integrated in the test bench control system. The oil conditioning is realized in an environmental circuit consisting of a pump p5, the cooler c1 and the filter f2.

In Figure 9, the test pump (p1) with the hydrostatic bearing is shown. The angle of the swash plate is set manually using the fixation bolts. The drive mechanism of the pump, which consists of the cylinder block, the pistons and piston slippers, is connected to the drive shaft through a gearing in the piston drum. During operation, the valve plate lies frictionless in the hydrostatic bearing. The hydrostatic
bearing consists of the housing, the hydrostatically relieved internal rotor and the sensor holder. A force sensor and 6 eddy current sensors are integrated in the sensor holder.

4.2. Experimental Results

For the validation of the simulation results, the standard valve plate and the prototype are measured at the test bench to analyse the friction torque for the different configurations. It is measured by the force sensor (s6). The hydrostatic bearing is pressurized constantly with 92 bar. The measurements from the run-in-process showed that the inner runner is completely relieved even when the load pressure is more than 300 bar. The low-pressure side is pre-loaded with 8 bar constantly for all considered operation points. The measurement points are as follows, four speed points (250 rpm, 500 rpm, 750 rpm, 1000 rpm) are approached and then the pressure is increased from 100 bar to 250 bar. The oil, a standard HLP46, is regulated to have a constant temperature of 40°C, as it was also used in the simulations. The results of the temperature measurements show that the temperature on the low-pressure side is 46°C. This can be explained with the heat dissipation of the pump p2 and the filter f1. The temperature on the high-pressure side is higher but less than 1 °C and increases less than 2°C during operation. Figure 10 shows an exemplary measurement of the temperature for the measurement at 1000 rpm with the standard valve plate.

![Diagram](image_url)

**Figure 9:** Test pump and sensor location

**Figure 10:** Temperature during measurement at 1000 rpm

**Figure 11** shows the measurement results for both configurations. The friction torque increases almost linearly with the pressure increase. With the increase of the drive speed the friction torque also increases. These measurements are influenced by both the solid friction and shear friction, which increases with the drive speed.
The relative change of the friction torque when comparing the measurement results of both configurations is shown in Figure 12. A reduction of the friction torque can be noted for all operation points. The reduction is the highest at 250 bar and range from -6.3 % up to a maximum of -15.9 %. The results show that the effect of the pressure pockets is influenced by hydrodynamic effects and the increase of leakage. This can be seen in the measurement results of 250 rpm and 500 rpm. One advantage of the tilting towards the high pressure side is, that the decreased gap height increases the sealing, which is one of the main tasks of this tribological contact. The aim of the new design concept is to reduce the tilt angle, but this increases the leakage due to the higher gap height in this region. The experimental results show that the additional pressure pockets reduce the friction torque and support the simulation study even though the simulation showed a greater effect.
Due to the reduced sealing gap caused by the additional pressure pockets, leakage increases as expected. The relative change of the volumetric efficiency is shown in figure 13. On average it is less than 1 %. At low speed, leakage is comparatively high, especially at high load pressure. The measurements at 250 rpm show an increase in volumetric efficiency. However, this is probably due to the viscosity, which is strongly influenced even by small temperature differences.

![Figure 13: Relative change of volumetric efficiency](image)

5. Conclusion and Outlook

Based on a simulation study a new design concept with additional pressure pockets at high pressure side was developed to reduce the tilting of the cylinder block. The simulations of the configuration with an additional surface of 101.01 mm² show great potential. The implementation of these pressure pockets reduces the tilting and the solid friction in all considered operation points. In this work it is shown that the simulation results could also be validate in experimental investigation of the new design concept. The friction torque for the configurations with a standard and a prototype valve plate is measured for several operation points. The experimental results show that the friction torque increases with the load pressure as well as the rotational speed. In all investigated operation points the friction torque is reduced for the prototype configuration. However, the reduction is at maximum 15.9 % and therefore less than predicted by the simulations. Additionally, the increasing effect of the pressure pockets with an increasing load pressure could not confirmed with the experimental results. One possible cause can be fluid friction, which is increased by the changed geometry. Basically, the experimental results show that the new design concept has the potential to reduce the friction in the considered contact.

The next steps in the ongoing research project are the extension of the experimental investigation for this design concept as well as the design concept presented in [14]. The effect of both design concepts to the tilting has to be measured with the explained single contact test rig. After, the results can be used by hydraulic displacement machine manufacturers to increase the overall efficiency of the components.

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**Nomenclature**

\( c_1 \) Cooler 1

**DOR** Direction of rotation

\( d_{cb} \) Outer diameter of the cylinder block

\( f_1...2 \) Filter 1...2

\( h \) Gap height

\( h_{cb} \) Height of the cylinder block

\( \Delta h \) Gap height difference

**ifas** Institute for fluid power drives and systems

\( l \) Gap length

**LV** Load valve

\( p_1...5 \) Pump 1...5

**PI** PI controller

\( r_1 \) Inner radius of high pressure kidney

\( r_2 \) Outer radius of high pressure kidney

\( r_i \) Inner radius of valve plate

\( r_m \) Medium radius of high pressure kidney

\( r_o \) Outer radius of valve plate

\( s_1...10 \) Sensor 1...10

**T_{LP}** Temperature low pressure

**T_{HP}** Temperature high pressure

\( \alpha \) Tilt angle

\( \gamma \) Start angle of high pressure kidney

\( \delta \) End angle of high pressure kidney

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