Optimization of Axial Piston Units Based on Demand-driven Relief of Tribological Contacts

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Abstract

Markets show a clear trend towards an ever more extensive electronic networking in mobile and stationary applications. This requires a certain degree of electronic integration of hydraulic components such as axial piston pumps. Beside some well-known approaches, the transmission of axial piston units still is relatively unexplored regarding electronification. Nonetheless there is a quite high potential to be optimized by electronic. In view of this fact, the present paper deals with the tribological contacts of pumps based on a demand driven hydrostatic relief. The contact areas at cylinder - distributor plate, cradle bearing and slipper - swash plate will be investigated in detail and it will be shown how the pump behavior can be improved considerably through a higher level of relief and a central remaining force ratio. The potential of optimization is to improve the efficiency, especially in partial loaded operation, power range, also for multi quadrant operation, precision and stability. A stable lubricating film for slow-speed running and for very high speeds at different pressures is ensured as well.

KEYWORDS: Electronification; axial piston pump; demand-driven relief; efficiency

1. Introduction

Highly integrative and ever more networked systems, which are also referred to as electronification, are a clear trend in hydraulic applications. This development is due to higher demands on energy efficiency, precision, usability and low-maintenance systems. This leads to the increased use of electronically controlled hydraulic units in stationary applications. In addition, there is a high interest in networking the different system components. /1/

This trend is similar for mobile applications. Bus and sensor systems are indispensable in modern tractors. Research projects, dedicated to extensive system networking or electronification, are well-known from agricultural and construction machines. /2/ /3/ /4/

Electronic pump control is the main approach used for hydrostatic displacement units like axial piston units. Furthermore, electronic-based research deals with condition monitoring subjects and with different approaches to optimize pressure transition processes. /5/ /6/

Early failures are mostly due to units' highly stressed tribological contacts. This lack of reliability is nearly always caused by insufficient thickness of lubricating films.

For geometric-design reasons, the power range of axial piston units is limited to a certain range of operation. The most relevant factor to be taken into account is the hydrostatic force of relief i.e. the resulting force because of the pressurized field, which is built up in the lubrication gap. The relief is to largely compensate the contact forces that are applied to the relatively moving parts. Due to the geometric dependence of the level of relief (κ) as a quotient of the relief force and contact force, the latter can only be designed with a certain pressure dependence for the operating range of axial piston units. Since the system, however, is operated at different drive speeds and swivel angles, shear and centrifugal forces as well as hydrodynamic effects all influence the equilibrium of forces and moments in addition. The level of relief hence is only a compromise enabling operativeness over the entire operating range. Since the hydrodynamic and hydrostatic effects interfere with each other, the issues above become evident during slow-speed running and at very high drive speeds. But the biggest problems are caused by the multi-quadrant operation, as a pump or a motor. This leads to an inversion of direction of the frictional forces between piston and cylinder barrel. The relation between stress and relief forces changes.

In spite of an increasing cost pressure, the efficiency-specific optimization of hydrostats still is highly relevant. Mobile machines are required to meet ever stricter environmental and energy efficiency criteria. Those are prescribed by the EU emission standards /7/. In the case of stationary applications, there is a continuous trend towards fully-electric driven production machines /8/, due to maintenance and energy efficiency advantages. To be able to compete, the efficiency of the displacement units must be increased.

An evaluation of diverse load spectra of agricultural, forestry and construction machines and of a tractor shown in particular, revealed that mobile applications are often operated in the partial load range.

The specifics of the tractors' operating conditions make it very difficult to derive generalized load spectra. The DLG (Deutsche Landwirtschafts-Gesellschaft), for example, offers the possibility to evaluate different vehicles on basis of customers individual needs /9/. According to /10/, a significant trend can be derived (**Figure 1**, left).

In the case only for working hydraulics, one can differentiate between heavy pulling tasks, power take-off, hydraulic work and transportation (see Figure 1, right).

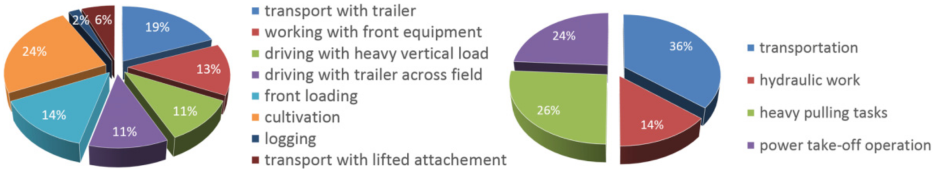

Figure 1 load spectrum of a tractor

Considering the load of the hydrostatic pump (**figure 2)**, which supplies the working hydraulics, one finds that particularly the increases in partial load and standby conditions must be considered in spite of the lower total power.

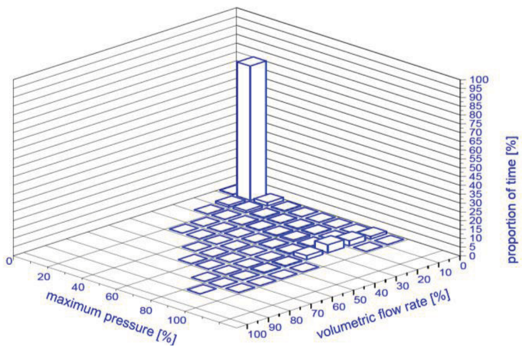

Figure 2 characteristic operation points of a tractor

The efficiency of the axial piston pump is defined essentially by its tribological contacts. In the corner power range, modern axial piston pumps achieve efficiencies of more than 90 %, which, however, are much lower in the partial-load range. This is due to the fact that the losses do not only depend on the power and that the relief, which is essentially responsible for friction and volumetric losses, is designed for corner power for reasons of robustness /11/.

As is evident of the individual losses according to /12/, the tribological contacts slipper pad - swash plate, cylinder barrel - distributor plate and the cradle bearing, are the main points of loss beside the cylinder barrel – piston contact of the axial piston unit.

2. Demand-driven Relief

Electronification intends to combine the advantages of both technologies: The robustness and power density of hydraulics and the energy efficiency and networkability of electronics.

One approach to solve the problems referred to above consists in the demand-driven relief of the tribological contacts. For that purpose, the relief forces, which counteract the contact pressure in the plain bearings, are controlled. Control is intended to keep the residual contact pressure as low as possible and locate the position vector of the latter as centrally as possible in order to realize a continuous and at the same time minimum gap. The resulting additional benefit is a minimum friction loss, which also minimizes unnecessary wear. Moreover, the volumetric losses depend on the gap height very much.

2.1. Cradle Bearing Approach

The cradle bearing is among the three contact areas dealt with above. In axial piston pumps, cradle bearings are mostly of the plain-bearing type and are relieved hydrostatically to minimize the losses. The swash plate obviously deforms due to positioning forces, transmission forces and the resulting bending moments **(Figure 3)**. This creates increased gaps, especially on the outer side of the cradle bearing. In contrast to that, the inner side gap is zero. This is responsible for high volumetric losses and losses of friction at the same time.

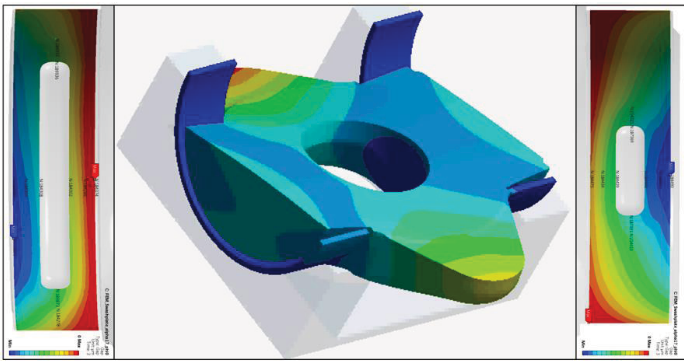

Figure 3 deformation of a swash plate axial piston unit

The compromise to be found between the relief of the cradle bearing and the resulting friction is between dynamics and stability of the control system. This influence has been regarded in a sensitivity analysis.

By means of a suitable control system, demand-driven relief aims to adjust the pressures of the fields of relief to positively influence both the losses and the dynamics of the control system.

In terms of pressure the curve of the level of relief **(Figure 4)** reveals the potential of relief in accordance with the operating point due to the irregular characteristic of the remaining force. On the one hand, it is obvious that, due to spring preload, the level of relief decreases considerably in the case of low pressures. Moreover the cylinder barrel position, which defines how many pistons are pressurized, has a high relevance. This leads to temporary over relief and hence to sealing gap.

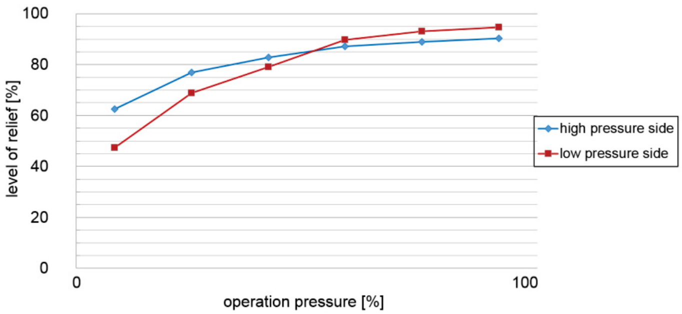

Figure 4 swash plate bearing relief of a swash plate piston pump

Based on the demand-driven relief, the level of relief can be adjusted optimally. In addition, demand-driven relief provides the considerable additional benefit of providing relief only when there is a need for swiveling. This considerably reduces the losses and vibrations.

The functional principle shown in **Figure 5**, is currently applied to validate the theoretical additional value.

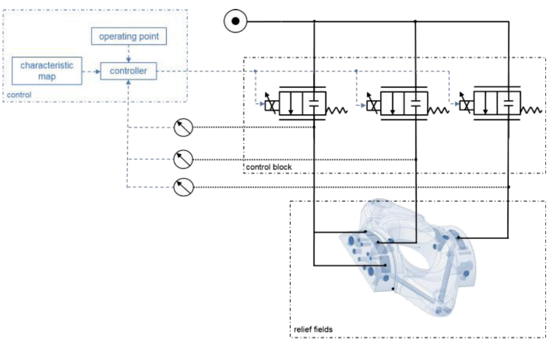

Figure 5 circuit diagram of valve controlled relief fields of the swash plate

The control of the fields contains information about the current operating point and compares it to data in a specific characteristic map. In doing so, the pressures in the individual pockets are controlled and adapted to the operating points

The additional benefit of this approach, on the one hand, is evaluated considering the swivel dynamics and transmission stability out of the swivel angle signal and, at the same time, through gap measurement devices and volume flow sensors in the high pressure fields.

Especially for applications e.g. mobile applications that are not swiveled over long periods of time the losses can be reduced considerable.

2.2. Cylinder Barrel – Distributor Plate Approach

The tribological pair cylinder barrel – distributor plate is another contact area whose losses have a significant impact on the efficiency. /12/

Also here, the basic problem is that a geometry-determined level of relief occurs at different operating points. The cylinder barrel is pressed onto the distributor plate mainly through action of the compressive force. Besides, there are centrifugal forces, shear forces at the pistons of transmission and speed-dependent hydrodynamic effects. The frictional forces whose direction, as explained above, depend on the operating mode, mainly occur at the pressurized pistons of the transmission unit. During pump operation, the oncoming pistons are subjected to high pressure, which creates a compressive force. During motor operation, this force is reversed and acts as an additional relieving force. The precise characteristics of the frictional force can be calculated by means of the simulation tool SiKoBu /13/.

Another problem consists in the above-mentioned centrifugal forces and shear forces. These affect the center of gravity of the residual compression of the two contact partners in addition. The decentralized position results in cylinder tilting and, hence, inconsistent dimension of the lubrication gap, which causes increased friction and volumetric losses.

The resultant relief field can be determined by the two dimensional Reynolds equation. /14/ Expressed in polar coordinates reveals the resulting formula:

$$\frac{\partial}{\partial r}\left(rh^3 \cdot \frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \vartheta}\left(h^3 \cdot \frac{\partial p}{\partial \vartheta}\right) = 6\eta \cdot \left(2r\frac{\partial h}{\partial t} + \frac{\partial}{\partial r}\left(h(u_r^1 + u_r^0)\right) + \frac{\partial}{\partial \vartheta}\left(h\left(u_\vartheta^1 + u_\vartheta^0\right)\right)\right) \qquad (1)$$

The equation (1) can be simplified by insertion the boundary conditions of velocity (2).

$$u_r^1 = u_r^0 = u_\vartheta^0 = 0 \;\&\; u_\vartheta^1 = \omega r \qquad (2)$$

$$\frac{\partial}{\partial r}\left(rh^3 \cdot \frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \vartheta}\left(h^3 \cdot \frac{\partial p}{\partial \vartheta}\right) = 6\eta \cdot \left(2r\frac{\partial h}{\partial t} + \omega r\frac{\partial h}{\partial \vartheta}\right) \tag{3}$$

Since the intended parallel gap is the aim of the optimization, the right side of the Reynolds equation, with the “wedge” and “squeeze effect”, can be neglected:

$$\frac{\partial}{\partial r}\left(rh^3 \cdot \frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \vartheta}\left(h^3 \cdot \frac{\partial p}{\partial \vartheta}\right) = 0 \tag{4}$$

The unit’s possible operating range depends on the extent of the residual pressure and the behavior of the relevant position vector. If the level of relief increases or the characteristics of the residual force decentralizes too much, the lubrication gap widens. This multiple widening is referred to as transmission lifting. If this case occurs, the unit does no longer operate correctly.

Figure 6 shows the relief field (left) at nominal speed and maximum pressure for a swiveled-out condition. Whereas the yellow (bright) area is pressurized, the blue (dark) one represents the surrounding or suction pressure. In the graphic on the right, the center of the loading force is shown in blue (dark grey), whereas the relief is represented in green (bright grey) and the residual force is marked red (grey).

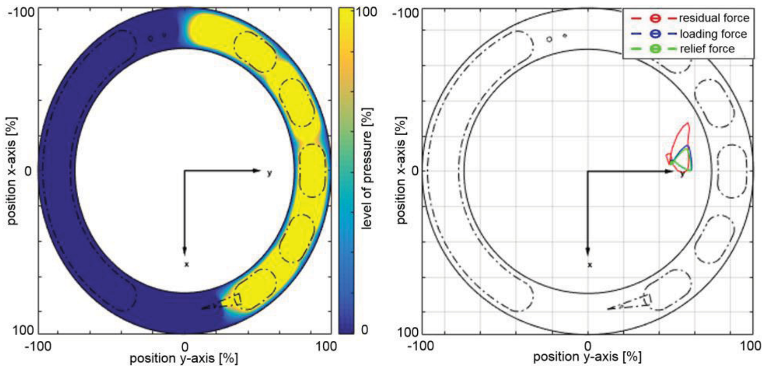

Figure 6 series relief and center of force curves of the distributor

By means of demand-driven relief, both the position and the amount of the residual contact pressure can be influenced. The approach applied is a reduced basic relief, which about corresponds to that of a motor unit, as well as additional relief fields that can be demand-triggered.

The valve actuation concept shown in **Figure 7** depicts a three-field implementation. Equivalently to the cradle bearing, the operating point is measured and the relevant pressure applications to the fields are read out from a characteristic map.

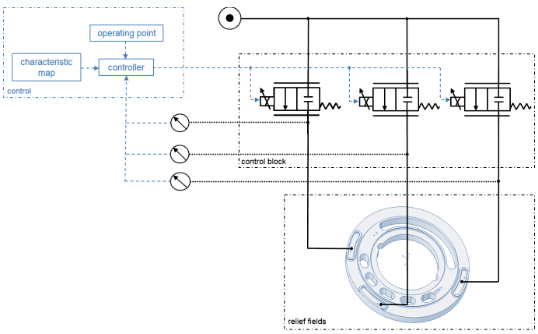

Figure 7 circuit diagram of valve controlled relief fields of the distributor

As shown in **Figure 8**, simulation can prove that the level of relief can be optimized towards full relief while, at the same time, the position of the vector of the residual force orients itself more in the direction of the center.

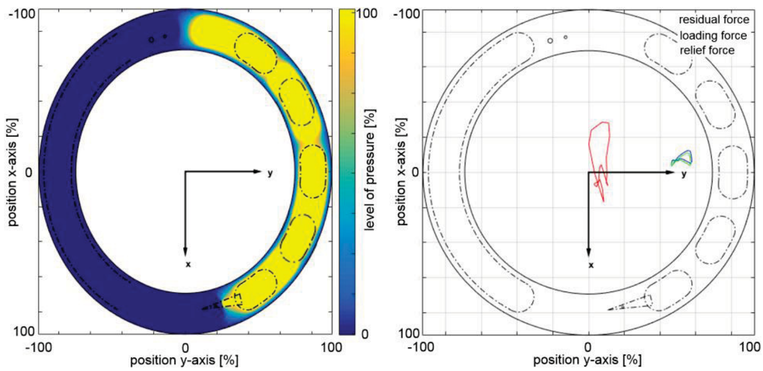

Figure 8 optimized relief and center of force curves between of the distributor

Comparing the calculated additional value of the optimized unit during pump operation with a unit designed especially for pumping, it becomes clear **(Figure 9)** that the level of relief is increased during partial-load operation and stays constant at high speeds. The lifting problem can be shifted to speeds more than double as high as those of current series products.

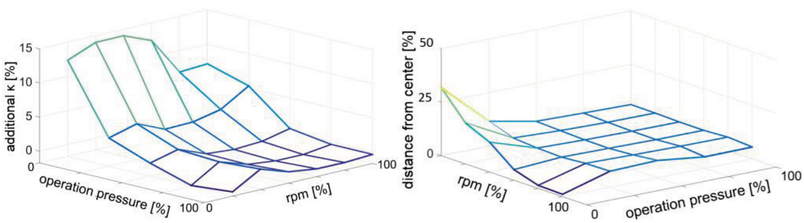

Figure 9 κ difference and center of residual force in pump mode

Figure 10 shows that also during motor operation, compared to the original system, the level of relief is more uniform in the case of a more centralized characteristic of the residual force and, hence, lower volumetric losses.

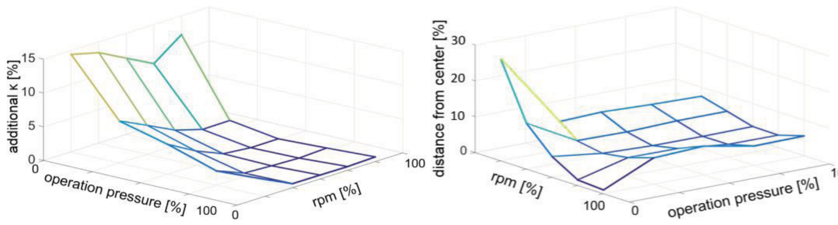

Figure 10 κ difference and center of residual force in motor mode

2.3. Sliding-disk Approach

The losses at the tribological contact slipper pad – swash plate have approximately the same level than those at the distributor plate.

This tribocontact is also relieved depending on the pressure and based on its geometric design. However, as already explained in /15/, also other pressure-independent forces, which cannot be compensated merely geometrically, have an influence.

The centrifugal force plays a particular role because the slipper pad experiences a torque that leads to tilting due to the rotation around the drive shaft and the fact that the center of mass is not in its own pivot point. At the same time, hydrodynamic effects act on the slipper pad, causing additional tilting in the direction of rotation. In addition, this has an influence on the relief force as a function of the speed. The operation either as a pump or motor also affects the extent of the contact pressure.

As in the case of the above contacts, these problems can be approached through demand-driven relief. Based on a sliding disk, which is designed rotationally symmetric to the drive shaft, the hydrostatic field can be controlled, lifting problems on the suction side can be solved and centrifugal forces and hydrodynamic effects can be reduced considerably.

By controlling the relief, the system can be relieved uniformly both in the motor and pump modes.

The functional principle shown in **Figure 11** left, takes account into a rotatory relative movement between the swash plate and the sliding disk, which rotates along with the pistons of transmission. The radial movement, which is required due to the elliptic circular path of the swiveled-out swash plate units, is realized separately between the sliding disk and the individual pistons of transmission.

Due to the above, the resulting contact pair sliding disk – swash plate is relatively equivalent to the contact pair cylinder barrel – distributor and can be relieved accordingly through demand-driven relief of kidney-shaped pressure fields (Figure 11 right).

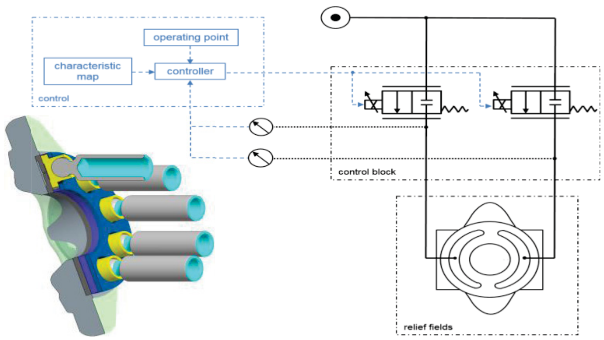

Figure 11 design and circuit diagram of the relief fields of the sliding disk

In turn, the relevant relief field can be adjusted such that the relief force counteracts the contact pressure in the best possible way and the eccentricity and size of the residual contact pressure are kept as low as possible.

For the swiveled-out condition, **Figure 12** shows the relief field at nominal speed and maximum pressure. Whereas the blue curve (dark grey) represents the loading force, green (bright grey) represents the relief and red (grey) stands for the residual force. The pressurized area is marked yellow (bright) and the surrounding or suction pressure is marked blue (dark)

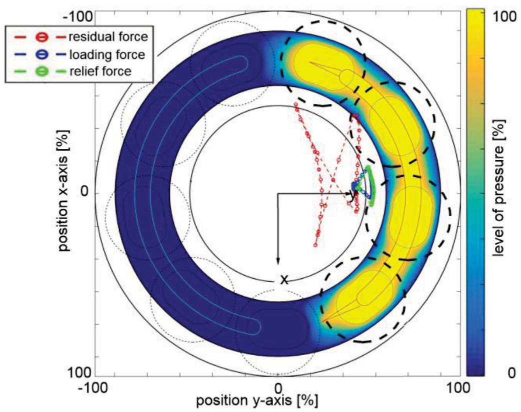

Figure 12 relief and center of force curves between of the sliding disk

3. Summary and Outlook

The results discussed hereunder provide an approach to obtaining an optimized design and control of the critical lubrication gaps in an axial piston unit. By way of simulation, a considerable additional benefit was derived, which apart from enabling an increase in efficiency over the entire operating range allows an increase in stability and an extension of the power range. In that way, axial piston units can be operated easily in the multi-quadrant mode and, based on electronification, can solve diverse challenges which will arise in the future for hydraulic displacement units for mobile and stationary applications. To be able to prove the real additional value, the approach is currently realized by a prototype and the theoretical thesis will be validated.

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5. Nomenclature

κ	level of relief	$-$
h	gap height	mm
η	dynamic viscosity	$Pa \cdot s$
r	radial coordination of the gap	mm
p	pressure	MPa
ϑ	circumferential coordinates	rad
t	time	s
u_r	radial surface velocity	$\frac{mm}{s}$
u_ϑ	surface velocity in circumferential direction	$\frac{mm}{s}$
ω	angular velocity	$\frac{rad}{s}$