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MEASUREMENT OF FRICTION FORCES BETWEEN PISTON AND BUSHING OF AN AXIAL PISTON DISPLACEMENT UNIT

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Abstract

In industrial applications mineral oil based fluids are used for power transmission and lubrication, enriched with additives for additional functions. These fluids are not fast biologically degradable and often used additives are even environmentally toxic. In case of leakage and disposal into the environment these fluids cause bad damage or high costs for damage prevention. Within the Collaborative Research Centre 442 „Environmentally Friendly Tribosystems by Suitable Coatings and Fluids with Respect to the Machine Tool“ at RWTH Aachen the aim is to replace mineral oil based fluids for power transmission and lubrication by fast biologically degradable fluids, which are based on native esters. To compensate the loss of functions in consequence for avoiding the usage of toxic additives in each tribological system, one of two sliding partners of a tribological system gets a PVD-coating. The subproject "Tribological Systems in Hydrostatic Displacement Units" at IFAS strives to modify a hydrostatic axial piston machine in a way, that it can be run with fast biologically degradable fluids without any disadvantages compared with today commonly used units performing with mineral oil based fluids. The axial piston pump is chosen, because its tribological systems include different geometries, loads and moving behaviour. The most critical point in this machine is the tribological system piston-bushing. The contact zone between both bodies is characterised by areas with high pressure, especially in case of complete evacuation of lubrication fluid from the gap and direct contact of the metallic bodies. To ensure the lifelong performance of coated pistons this load has to be reduced. The strategy for this reduction is realised by changing the contour of piston and bushing. In this paper it will be demonstrated, that the friction behaviour of piston and bushing, which is an indicator for the contact pressure, can be changed by the manipulation of the piston geometry. For the friction measurement a new test facility was designed and built up. A detailed introduction to the design and performance of this facility is given in this paper.

Keywords: hydrostatic displacement unit, fast biologically degradable fluid, biofluids, PVD, coating, piston, friction, piezosensors

1 Introduction

Most of the world-wide used hydraulic systems in a wide variety of applications work with hydraulic fluids based on mineral oil. Important characteristics like reduction of friction and wear are improved by additives. Solid and fluid materials are brought to a high effective combination (Fessenbecker, 1996 and Mang, 2001). So a change of the basic fluid material will effect important characteristics and might cause the system to fail. The use of fluids based on native resources which are fast biologically degradable leads to increasing wear and friction, because additives, which are highly toxic in most cases, have to be avoided. But they provide the system with important functions like reduction of friction and wear. So these tribological functions have to be transferred to the solid parts of hydrostatic units. A very useful way is the coating of tribological contact surfaces with ZrC.

Within the Collaborative Research Centre 442 the focus is directed towards PVD-coatings with an increasing fraction of carbon between 0% and 25%. This structure shows a variation in hardness with the maximum in the middle between the bottom and top layer (Fig. 1). At this point the coating elements exist in stoichiometric composition, which means that a pure Me-C compound exists. With advanced wear the resistance against wear increases and after a certain running-in period a balance is achieved between charge and resistance forces (Fig. 2). In this case the coating systems adapts to the applied charge (van Bebber, 2002 (2)). This running in behaviour of self adjusting geometry but without

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changes of the material hardness is well known for mainly used brass-steal combinations in tribological systems. Similar to brass the carbon is chosen to be an element of the coating system, because it is a solid lubricant and helps to reduce the friction coefficient of the new combination PVD-coating with steal.

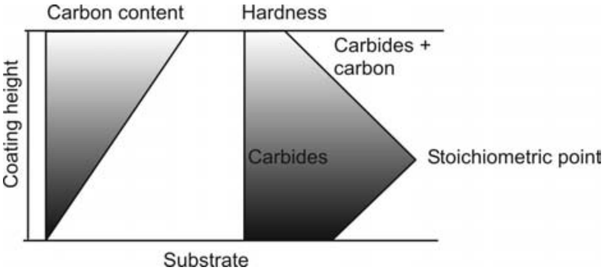


Fig. 1: *Structure of the ZrC_g-coating film*

The choice of the PVD coating process is based on the fact, that the low process temperature does not effect a change of the structure of the substrate material. The PVD process allows to realise material films of only a few µm. This means that a well prepared substrate surface is mapped identical. So coated surfaces are quite easy to handle in tribolocical systems with high surface qualities and very narrow gap heights (between 5µm and 10µm).

The characteristics of the developed coating (van Bebber, 2003) within a tribological system has to be adapted to the real tribological systems of the hydrostatic displacement unit. The piston-bushing contact represents a high challenge because of changed boundary conditions, like complex geometric conditions, high loads and discontinuously movement.

It has been demonstrated that PVD-coated pistons can be used in hydrostatic displacement units. WC/C and W+C:H^2 coated pistons exhibit much less traces of wear than a brass-steal combination in a pump test bench. A fluid contaminated with particles does not effect visible wear on the pistons (van Bebber, 2003 (1), 2003 (2)).

Both coatings have a more or less discrete design and they do not yet follow the strategy of graded coatings with a continuously change of hardness, which is the main idea within the Collaborative Research Centre 442.

To apply a new coating system to the piston, it needs detailed research on this subject. To accelerate the process of wear in experiment, measurements will be done at low speed. Once a fitting coating is developed, investigation will include measurements at high speed, too. A limit of n_{max}=5 1/min is chosen, which is near the speed, that hydrodynamic lubrication is established, so below friction and wear have their maximum. A new test bench concept is set up to realise measurements with low speed and high pressure. Investigations with pressure levels of 40 MPa will be published later.

2 Characteristics of the piston's movement inside the bushing

Due to the pressure and the movement of the piston a specific balance of forces is set up to the piston, which is shown in Fig. 2, reduced to the quasi static situation neglecting dynamic loads. This simplification is possible for low engine speed.

Figure 2 shows the piston in a tilted position and the reaction forces. F_1 and F_2 represent the resulting forces of the stress distribution σ, shown on the right side of Fig. 2. F_1 and F_2 effect the frictrion force F_R=$F_{R,1}$+$F_{R,2}$. The friction coefficient μ depends on the condition of lubrication, the surface roughness, materials, etc.. The total friction F_R is calculated as:

$$F_R = \frac{F_p}{1 + \frac{a-b}{(a+b)\cdot\mu\cdot\tan\alpha}} \quad (1)$$

For the motor mode the direction of $F_{R,1}$ and $F_{R,2}$ changes and F_R decreases for the same friction coefficient μ and pressure p.

Viscose friction is not yet regarded. In relation to F_R viscose friction, consisting of the Couette and Poiseuille part, is negligible low compared to the high friction forces shown in chapter 5.

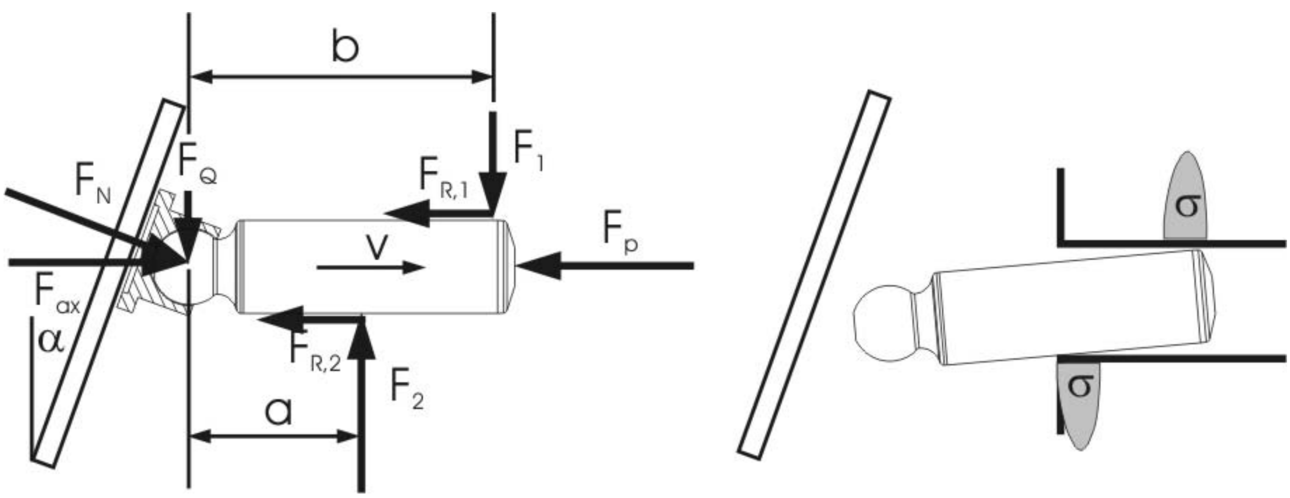


Fig. 2: *Static balance of forces at the inclined piston in pump mode*

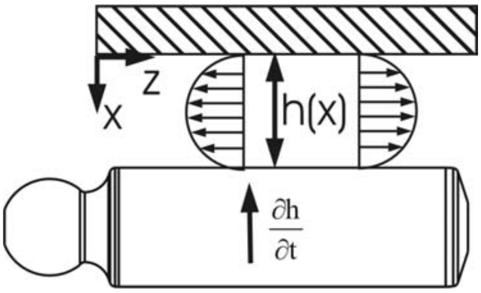


Fig. 3: *Squeeze effect*

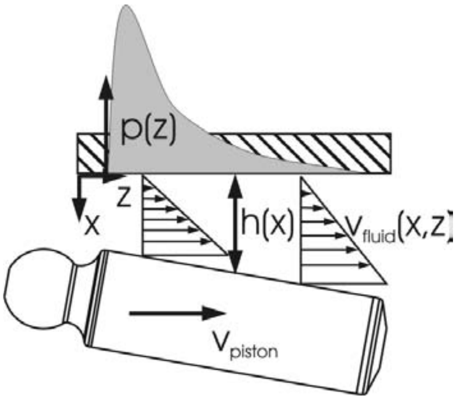


Fig. 4: *Creation of hydrodynamic pressure (Couette)*

Charged with low pressure the piston remains in a concentric position within the bushing. It is tilted in radial direction by F_Q, which increases rapidly when the pressure changes to the high level. The fluid between piston and bushing is pressed out of the gap between both (Fig. 3) and the fluid pressure rises. The radial movement is slowed down by this pressure force and the contact of piston and bushing is either retarded or avoided, which depends on the engine speed. Within the Reynolds's equation (Eq. 2) $2 \cdot \frac{\partial h}{\partial t}$ shows this effect, which is the squeeze effect. Equation 2 is reduced to the 1-dimensional form, which is sufficient to show the main influences to the creation of pressure within the lubrication gap.

$$\frac{\partial}{\partial z}\left(\frac{\partial p}{\partial z}\frac{h^3}{\mu}\right) = 6 \cdot \left(v_{\text{piston}} \cdot \frac{\partial h}{\partial z} + 2 \cdot \frac{\partial h}{\partial t}\right) \tag{2}$$

The hydrodynamic pressure is built up within the gap for increasing velocity of the moving piston (Fig. 4). The fluid is forced through the diminishing gap, that means, that the volume gets smaller and the pressure rises, which is represented in Eq. 2 by the term $v_{\text{piston}} \cdot \frac{\partial h}{\partial z}$.

For low engine speed this part gets small in relation to further aspects.

Except of the squeeze effect further phenomenon are worth to respect in terms of analysing reasons of friction. The later on described measurements support the idea, that the direction of the pistons movement is quite important to the friction. During displacement under pressure either the edge of the piston slides over the surface of the bushing or vice versa. For the pump performance the angle in moving direction between piston and bushing is large (Fig. 5, case A). This geometry does not deliver any benefits to the lubrication film between both parts. For the motor the angle is near to zero (Fig. 5, case B) and that does support the setup of a separating lubrication film immensely. Otherwise the effect could be explained by elastic deformations in very small dimensions, which produce a geometry, that does influence the direction of the normal contact force in that way, that it features a force which points against the direction of the piston movement. In case A it is more distinct than in case B.

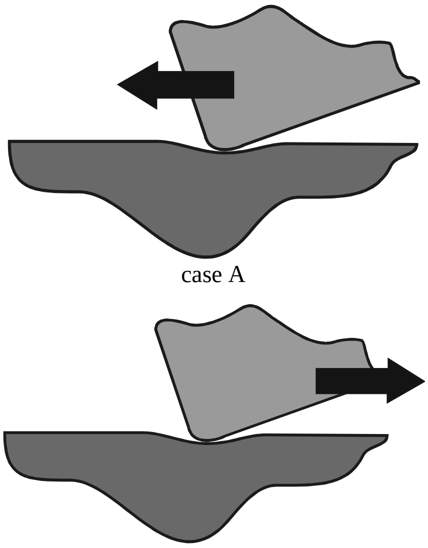


Fig. 5: *Sliding of an edge on a surface*

The moving direction also touches the ability to build up a lubrication film between piston and bushing (Fig. 6). Except of the moving direction the length of the bushing's surface is essential. Is the end of the piston, which should be the side without ball joint and slipper, inside the bushing, the provision of the gaps between piston and bushing with fluid is sufficient, if the piston is pushed out of the bushing, because on both sides the gap becomes shorter and oil is not easily brought into it. In case of a pump both gaps become longer and the provision with lubricant is not sufficient, because on one side it cannot be pressed into the gap fast enough and is slowed down by its viscosity in the very thin gap and on the other side the gap's volume grows without any high pressure provision.

Is the end of the piston outside the bushing, in both kinds of performance, pump and motor, there is one gap, which is provided sufficiently by fluid, that is torn into the gap by the piston itself, and the other one, which is not provided sufficiently, because the piston tears the oil out of the gap.

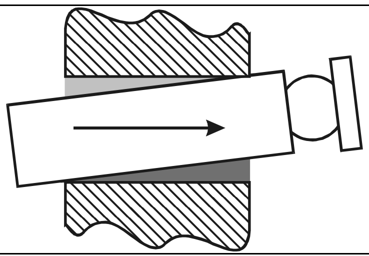
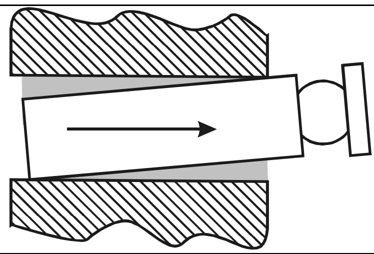
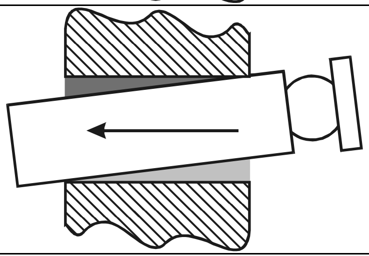
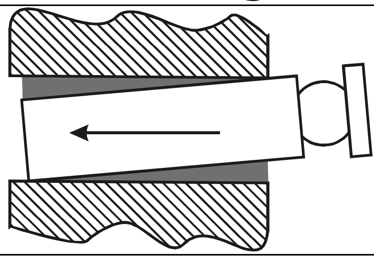
sufficient oil provision not sufficient oil provision	short bushing	long bushing
motor		
pump		

Fig. 6: *Effect of oil provision in the gap (Renius, 1974)*

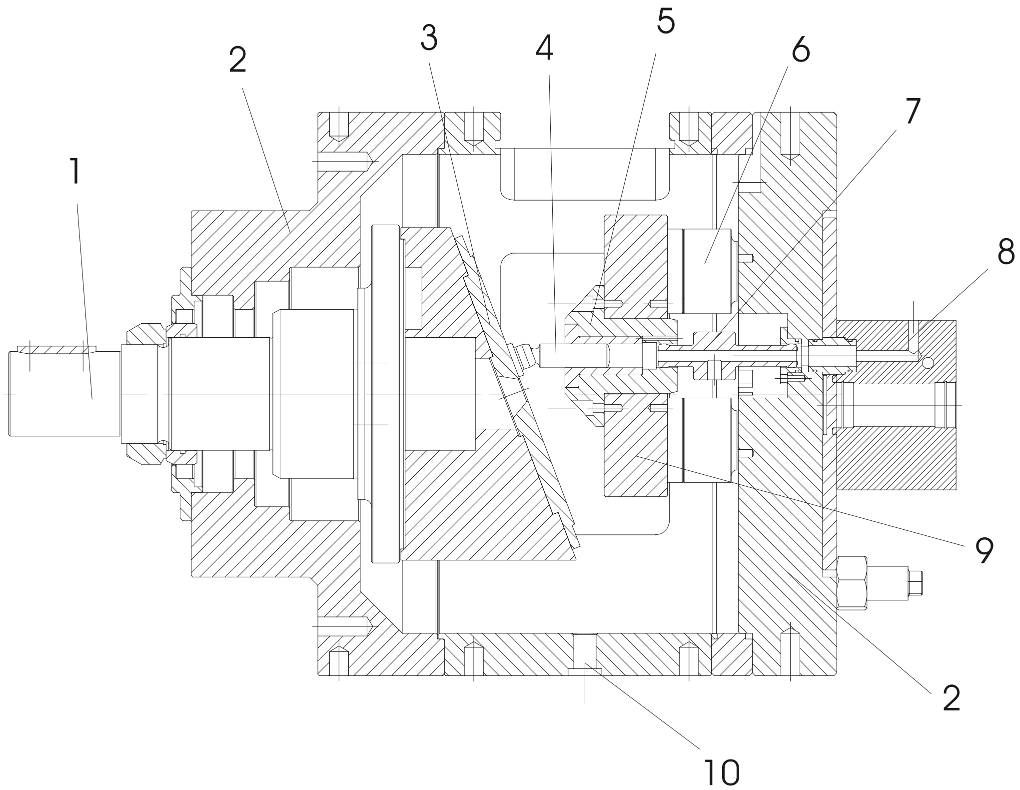


Fig. 7: *Cross section of the test setup; 1: drive shaft, 2: housing, 3: swash plate in wobble design, 4: piston, 5: bushing (with singleserving weared inner part), 6: 4 piezo sensors, 7: compensation piston, 8: position for adapting a valveplate alternative, 9: plate, 10: exit for leackage*

3 Research at the contact piston-bushing

The highest load within the tribological systems of a hydrostatic displacement unit occurs in the contact between piston and bushing. The zones of high pressure are very small and are restricted to areas of solid contact of piston and bushing. The distribution of this high pressure over a larger contact zone will reduce the local load remarkably and support the disposition of a coating, which is very sensitive to high load gradients. A better distribution of the load can be reached by manipulating the surfaces of the contact bodies. A continuous reduction of the piston diameter at both ends can make the contact zone between piston and bushing larger. It also will reinforce the squeeze effect, that delays the moment of contact between piston and bushing. That has to be approved by experimental results.

The contact of piston and bushing has been subject of several scientific investigations.

Renius (1973, 1974) built up a test bench with rotating swashplate and one piston. The bushing was placed in a hydrostatic bearing, to separate friction forces and applied forces. He made measurements of axial and tangential friction forces at high rotation speed (n>= 250 1/min) and low pressure (p=< 14 MPa).

Manring (1999) gives a analytical model for cylindrical shaped pistons and presented measurements at low speed and pressure for verification. Stress distribution can not be calculated as well as not cylindrical pistons or bushings.

Deeken (2001) and Wieczorek (2002) present numerical methods to calculate pressure distribution within lubrication gaps and between not separated solid bodies. These numerical tools are restricted to high revolution numbers and do not include material wear and specific elastic behaviour of the simulated bodies.

Lasaar (2003, 2004) presents measurements and simulation results of pistons with reduced diameters at both sides of the piston to influence hydrodynamics positively and reduce viscose friction forces. The measurements and calculations are made at high engine speed. Wear at piston and bushing is not covered in this work.

To quantify the load between piston and bushing as well as wear and wear resistance of both components at low speed and pressures, including high pressure levels, experiments have to be conducted in a different way than they have been made before. For this reason a new test bench has been designed (Fig. 7) and used to conduct the experiments, which are described later on.

4 Description of the test setup

The friction force between piston and bushing is measured in axial and tangential direction to the piston symmetry axis. To simplify the positioning of the force sensors the swashplate is rotating, analogous to an axial piston pump in wobble design. The number of pistons is reduced to one. So far the test facility is built up following the idea of Renius (1974). Centrifugal forces are neglected, which are very small at low number of revolutions ($n_{\max} = 5$ 1/min) and do not cause inexact results.

There are no centrifugal forces to the slipper, which do exist in real axial piston pumps with rotating cylinder block. These forces have the effect of an inclination of the slipper radial to the rotating axis. In the shown test device this difference of the arrangement does not effect the friction between piston and bushing compared to real units as long as it is used at low engine speed. The rotation of the piston around its own axis, like it takes place in real axial piston pumps (Lasaar, 2004), can be observed in the test stand.

The bushing is placed on a plate, which is connected with the housing by four piezoelectric force sensors. Even charged by high forces piezos are almost not compressed or expanded, what makes the facility very stiff. The arrangement avoids the application of a bushing placed in a hydrostatic bearing. These bearings demand a highly accurate production to realise well dimensioned gaps and inherits a lot of unknown parameters, so that it is difficult to handle.

Between plate and housing an additional part is placed, which functions as a compensation piston (Fig. 8). Via this compensation piston the piston working area is charged with high or low pressure without transferring any forces between plate and housing. The compensation piston has the same diameter as the piston. Thus the pressure force F_p is the same at the piston and the compensation piston. The ball joints gurrantee the self-adapting placement in the bushing and housing. The compensation piston does, except of negligible low viscous friction forces due to leakage between compensation piston and bushing, not transfer friction forces to the plate. By this way the piezosensors are only charged by the friction forces between bushing and piston and not by the pressure forces in the working area of the piston. These pressure forces are transmitted without losses by the compensation piston to the housing.

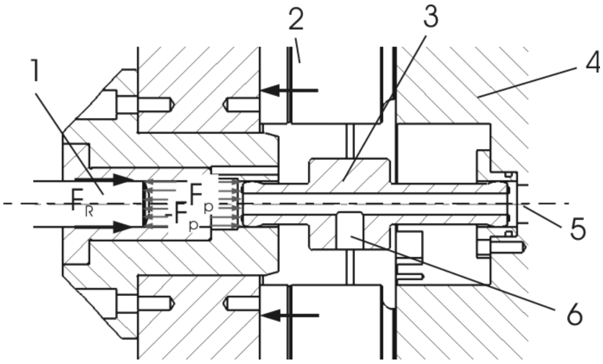


Fig. 8: *1: end of piston, 2: piezo sensor, 3: compensation piston, 4: housing, 5: position for adapting a valve-plate alternative, 6: access for pressure sensor, F_p: pressure force, F_f: friction force*

The ball joints at both sides of the compensation piston allow an adjustment, in case that the axes of the assembled parts are not identical. The sealing is realised via a line contact, which requires a high demand to a very exact production of all parts.

In case of leakage at high pressures hot oil could touch the plate and cause a temperature drift of the piezo signal. This will be treated later in detail. This problem seems to be of less influence for the design of the compensation piston by Renius (Renius, 1974). The sealing between compensation piston and plate in this case is a surface contact, what is easier to handle. The connection to the housing near point 5 is designed as a hydrostatic axial slipper bearing, which has a special formed surface to control the gap height. This option will be evaluated in experiments as well.

In a first step the valve plate of the original hydrostatic displacement unit is replaced by a servo valve, so that pressure changing functions for different valve plate designs can be imitated. A further step will be to install a mechanical control unit, that works similar to the original valve plate.

The arrangement of the piezosensors can be taken from Fig. 9 in the x-y-plane. The z-axis is orthogonal to x and y. The charge signals are summed up for each direction before passing through the amplifier. By this method the signal parts generated by the radial force on the piston are eliminated, because the sum of their portions is zero for the x-axis and the y-axis.

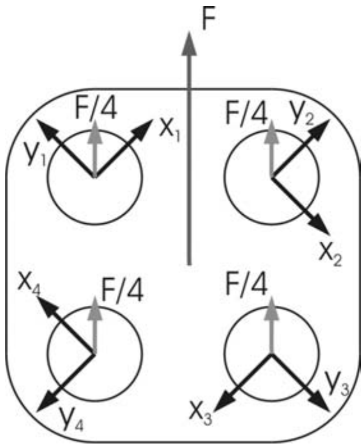


Fig. 9: *Assembly of the piezo sensors in the x-y-plane*

To measure the friction between piston and bushing only the x- and z-axis are of major interest. They provide an image of the resulting friction forces in the tribological system.

A characteristic of piezosensors lies in the reduction of charge, especially for quasi static measurements over long cycle times. In this case a drifting signal has also a second reason. If the plate is sprinkled by hot leakage oil, it is expanding faster than the much bigger and massive housing because of its smaller mass. Within the x-y-plane the plate is statically overdetermined with its four fixed contact points to the piezosensors. So it bows in z-direction and the sensors put out additional charge. In relation to the chosen cycle times the drifting zero point draws a linear line over time. The addition of a linear function helps to reconstruct the pure force signal without being not exact.

5 General aspects of the movement of the piston inside the bushing

Loads applied to piston and bushing depend in a high degree on the kind of unit operation. Is the machine used as a motor, they are much different to a machine used as a pump.

At the Top Dead Centre (TDC) the piston is deeper inside the bushing than during any other moment within one turn. Working in pump mode (Fig. 10) the piston is then pulled out of the bushing by the downholder, in the described test stand it is pushed out by the low pressure of ≈ 1 MPa. The pressure load is small and the piston has a concentric position in the bushing. At the Bottom Dead Centre (BDC) the load at the piston changes to high pressure and strong radial forces push the piston to the side. The piston inclines and is pressed to the bushing at both ends. The squeeze effect delays the displacement of the piston and is important to reduce friction and avoid direct contact of the solid parts of the tribological system. The latter especially at a high number of revolutions. But also a pure delay of metallic contact means to reduce the maximum of possible friction between piston and bushing, because with a certain delay the contact conditions, which are the position of the surfaces against each other, improve steadily.

With increasing velocity the piston is pushed into the bushing and the creation and reinforcement of a separating lubricant film is supported. The forces, that hold against the torque, which results from the radial force to the slipper, are further away from each other thus decreasing frictional forces.

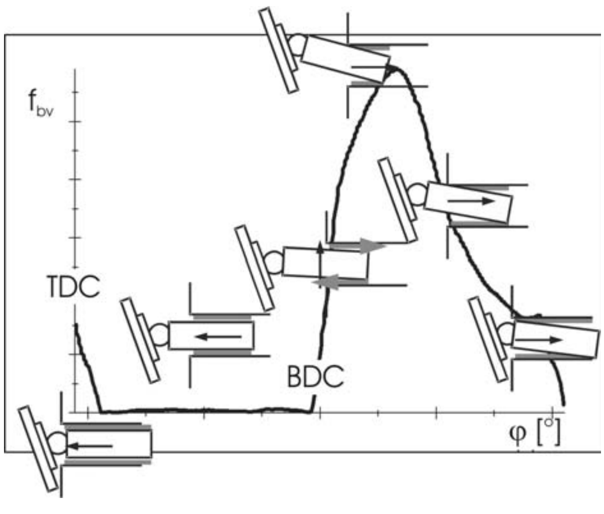


Fig. 10: *Dislocation of the piston while moving in pump mode*

In a motor (Fig. 11) the pressure changes at the top dead centre from low to high. The piston is inside the bushing and the parts of the piston at both ends of it, where the main forces hold the balance with the radial force at the slipper, have the maximum distance. The pressure between piston and bushing is low and friction increases with the distance, that it is pushed out of the bushing by the high pressure. The distance of the balance holding contact forces decreases and metallic contact can occur. The conditions for a maintaining lubricant film are much better than in a pump, which will be demonstrated further on. In the BDC pressure changes to low and the piston is pushed back into the bushing. Without radial displacement it holds a concentric position.

The pressure force, that pushes the piston against the swashplate, excites the radial force, that inclines the piston and causes high friction. In motor mode the friction forces are oriented against the pressure forces during high pressure and so by the friction forces the radial force is reduced, what can easily deduced by Fig. 2 and Eq. 1. Thus in motor mode the piston is less charged than in pump mode. In addition viscous friction between piston and bushing has the same effect, as leakage flow between piston and bushing is always directed along the pressure forces.

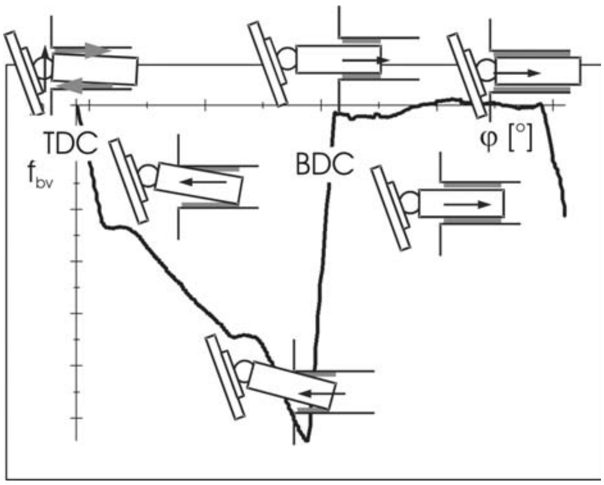


Fig. 11: *Dislocation of the piston while moving in motor mode*

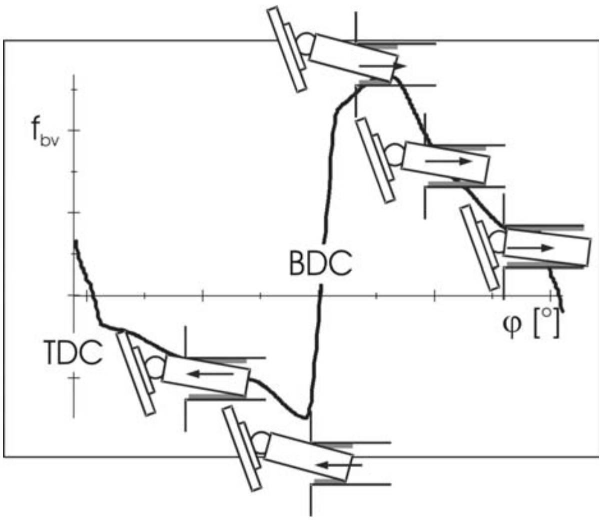


Fig. 12: *Dislocation of the piston while moving in pump-motor mode*

A further kind of performance for a hydrostatic displacement unit is to hold the pressure continiously on a high level (Fig. 12). In this case the piston is permanently forced to a rotated position in the bushing. In the dead center the movement from the concentric position to the linked position does not take place and as a consequence the squeeze effect is not part of the process. This kind of performance is just of scientific interest, because it is useful to quantify the impact of the squeeze effect to the friction.

6 Measurements

Two pairs of piston bushings have been investigated on the new test stand. Both pistons are made of steal and the bushings of brass. One piston has an industrial cylindrical surface, the second has the same macro-geometric sizes, but at both ends a reduction of the diameter has been realised with a high-performance-CNC-grinding-machine. To visualise the difference, only the axial part of the friction between piston and bushing was looked at. To simplify the comparison of measurements for different pressure levels the friction force R_{axial} is based on the pressure force F_p, that effects the end of the piston $\frac{F_R}{F_p} = \frac{F_R}{p \cdot \pi \cdot \frac{d^2}{4}}$. That also provides a statement about the impact of the pressure to the adhesion factor.

The friction values are shown from 360° to 720° of the angle position of the swashplate to demonstrate, that measurements are made of a steady process.

The legend is to read as:
example: n2_100_2_PU

- n2: number of revolutions n = 2 1/min (the range is from n01: n = 0,1 1/min to n5: n = 5 1/min)
- 100: pressure p = 10 MPa
- 2: the second of 3 measured cycles
- PU: pump performance (MO: motor performance; PM: pump-motor performance, continuously high pressure)

7 Industrial piston

Diameter amounts to d = 17 mm and the length of the surface l = 48 mm.

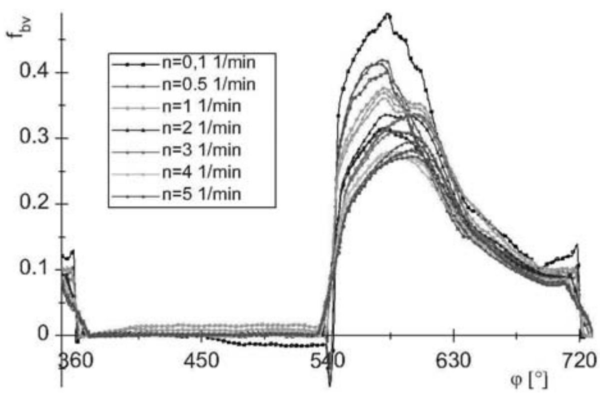


Fig. 13: *Friction forces of the cylindrical piston in pump mode at p = 10 MPa*

During pump operation (Fig. 13) the standardised friction value $f_{bv} = \dfrac{F_R}{F_p}$ decreases with increasing number of revolutions. At the BDC f_{bv} grows rapidly due to the raising radial force at the slipper. Then it changes visibly to a lower gradient. In both sections the gradient depends linearly of the engine speed. At the end of this section the maximum value is taken, that depends massively on the number of revolutions. With faster slide movement f_{bv} falls until the piston reaches the TDC. A rising value at the end of the movement might be increased due to the accumulation of worn off material at the end of the bushing, at the position of 3 mm in Fig. 14 b, against which the piston is pressed (Fig. 14 a,b). Also the worn surface of the bushing does not support a separating lubrication film.

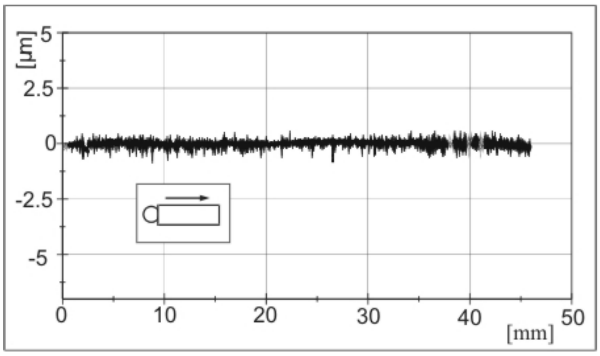


Fig. 14a: *Profil of the industrial piston*

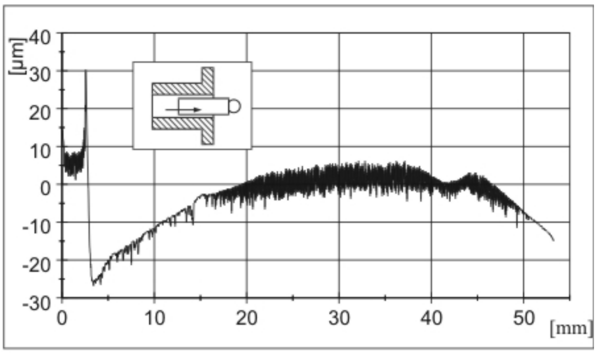


Fig. 14b: *Profil of the bushing*

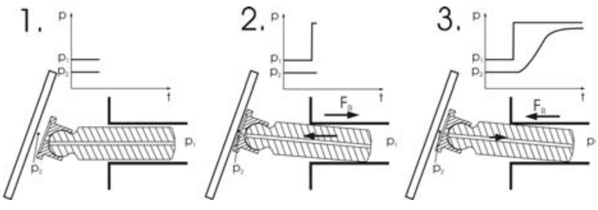


Fig.15: *Change of pressure under the slipper*

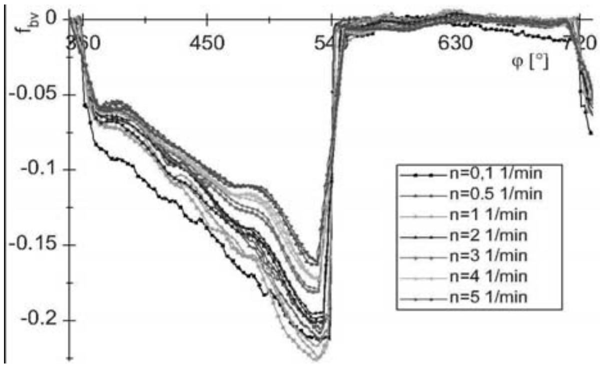


Fig. 16: *Friction forces of the cylindrical piston in motor mode at p = 10 MPa*

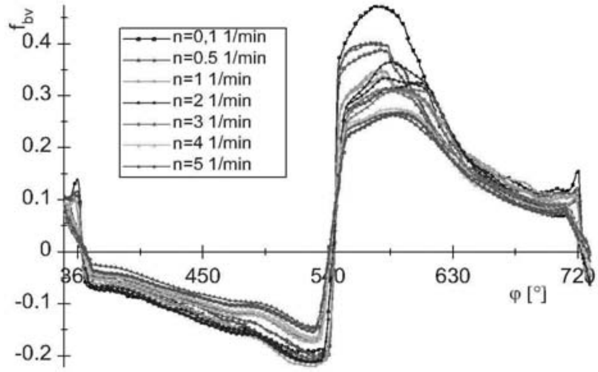


Fig. 17: *Friction forces of the cylindrical piston in motor pump-mode at p = 10 MPa*

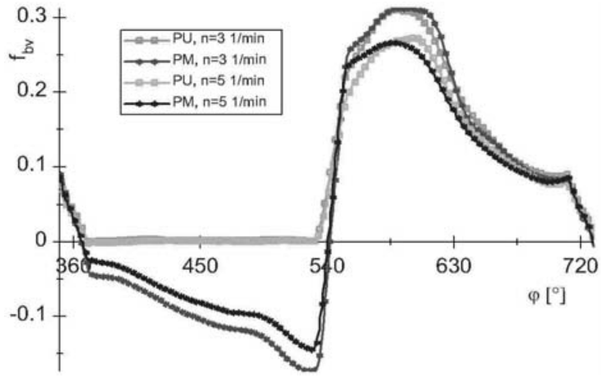


Fig. 18: *Comparison of f_{bv} with and without squeeze effect at different numbers of revolution for p=10 MPa*

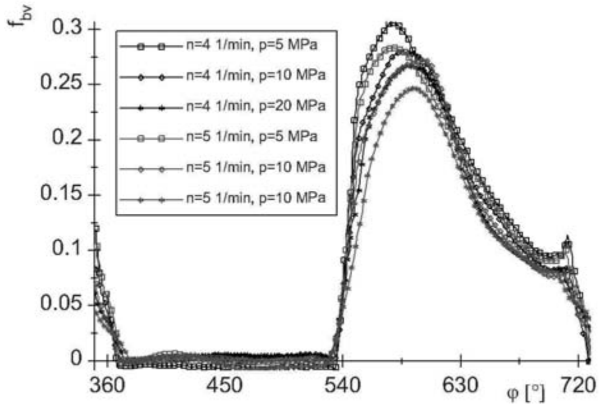


Fig. 19: *Impact of pressure on the friction*

At low engine speed at the angle $\varphi = 540°$ f_{bv} falls below zero. Friction due to a flow inside the gap, which is the Poiseuille's part of flow within the gap, can be calculated and lays beyond 5 N, which could not be seen in the graph (Fig. 13). This kind of friction also should not change with the engine speed and according to Lasaar (2004) it should appear at the end of the high pressure load in a similar way like at the beginning, which it does not in this case. The following explanation, which is illustrated in Fig. 15, seems to be more probable. The piston is pushed out of the bushing with low pressure. That pressure is sufficient to follow the backwards movement of the swash plate but not to press the slipper against it and being much bigger than the pressure force between slipper and swash plate. Is high pressure applied to the piston, the gap between slipper and swash plate becomes smaller, until a new balance of forces is set up. The piston moves out of the bushing for the distance, the gap becomes smaller, respected the trigonometric functions between both. During this movement the friction force exhibits a negative value.

In motor operation (Fig. 16) the piston is much less loaded in the contact areas to the bushing. The maximum as well as the gradient of f_{bv} depend on rotating speed. The maximum shows up at the end of the load cycle at the BDC, because it depends on the contact forces between piston and bushing, that have the smallest distances at this position. The level of friction is lower compared to pump operation. The reason is the positive effect of the sliding edge of piston or bushing in the surface of the counterbody and the better lubricant provision in the gaps between the long bushings and the piston.

In the academic mode (Fig. 17) of pump and motor the piston is tilted by high pressure load and the resulting radial force at the slipper inside the bushing over the whole cycle. Without the radial movement at the BDC by the applied radial force from the concentric to the tilted position the squeeze effect is suppressed. The first steep ramp of f_{bv} during the load's change goes up almost to the maximum value. The change of the load to the section of a lower gradient is later and smaller. The reduction of friction due to the squeeze effect does not take place. For the motor no essential differences exist.

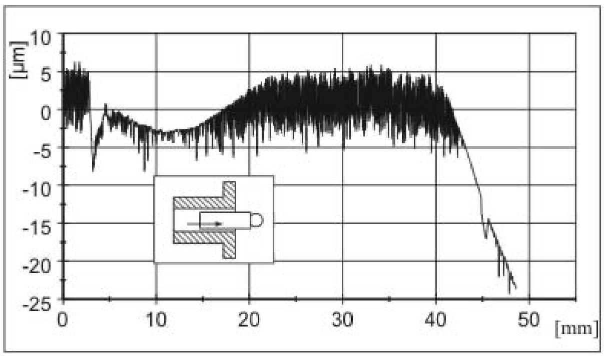


Fig. 20a: *Profil of the piston with reduced diameter*

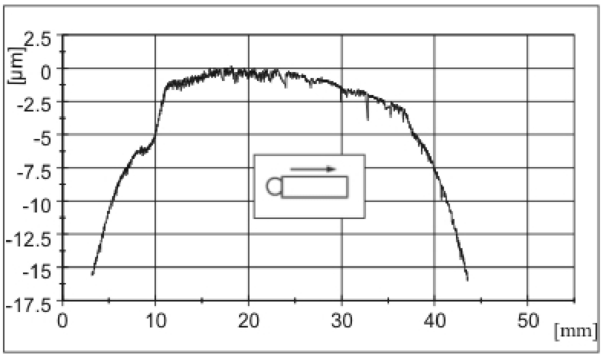


Fig. 20b: *Profil of the bushing*

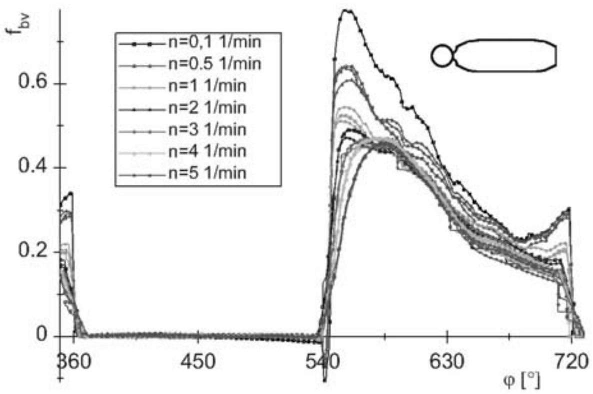


Fig. 21: *Friction of a piston with reduced diameter in pump mode at p = 10 MPa*

A comparison of measurements with and without squeeze effect (Fig. 18) shows, that without it the first increase in friction is much steeper and the change to a lower gradient occurs at a higher value. With this effect the gradient of the first step is less steep and the change to a less steep gradient in the second section is on a lower friction level. The impact also seems to grow with engine speed. Then deflection is faster and the oil has even less time to evade the shrinking gap volume. So the oil compression is higher and leads to a better separation of piston and bushing.

For the pump mode f_{bv} decreases with increasing pressure level despite of a higher friction level. That means that the influence of the pressure force on f_{bv} is reduced (Fig. 19).

8 Piston with diameter reduction at both ends

Originally the piston is from the same production lot like the first one, so the basic dimensions are the same. But at both ends the geometry is formed like a cone (Fig. 20a). This has been achieved by a grinding process.

The contour of the piston is based on geometries used for numerical calculations of van Bebber (2003). The reduction of the diameter has been increased to simplify the grinding process.

The effect of this geometry change is shown by the graph (Fig. 21). In pump mode the form of the rising ramp is different. With the cylindrical piston the friction shows two clearly separated sections between the ignition of friction and the maximum, one with a very steep gradient and a second with a lower gradient. With the changed piston design there is only one section with a linear increasing f_{bv} between the initiation and the maximum. For low numbers of revolutions this ramp goes straight up to the maximum value with a steep gradient, almost steeper than for the cylindrical piston. With growing engine speed the gradient decreases quickly. For $n \geq 3$ 1/min the gradient is more even than for the cylindrical piston. So the squeeze effect has for raised numbers of revolutions a much stronger impact to the progression of f_{bv}. It is based upon the changed geometry of the piston. Also the wear of the bushing (Fig. 20) is much less than for the cylindrical piston, regarding that the original profile is even over the whole length of the bushing.

The maximum values of the special designed piston's friction are much higher than the industrial piston's one. Other effects, that still need to be identified, worsen the performance of piston and bushing concerning the friction.

The manipulation of the piston geometry changed the friction behaviour of both parts. This means, that a defined reduction of the load between piston and bushing was obtained by this way. This reduced load effects less wear at the bushing.

9 Outlook

It was shown that the friction between piston and bushing can be measured with the presented test stand, using piezoelectric sensors.

Friction of piston and bushing is influenced by changes of the geometry of the piston in that way that the squeeze effect seems to be increased. Wear is reduced significantly at the end of the bushing, so the maximum pressure seems to be diminished.

In further research a piston and bushing geometry or its design rules need to be found for different operation parameters, that support the livelong use of coated pistons without wearing the coating off. The ZrC_g-coating is characterised by an initial running in wear to adapt its geometry. After the adapting phase the wear must stop and the generated geometry is supposed to last for the whole life cycle of the displacement unit reducing friction permanently. The ongoing work will include measurements of the distribution of radial tension around the bushing surface with a piezofoil. This foil will be placed inside the bushing near the contact surface and provides signals corresponding to the tensions in normal direction to the foil. These topics will be addressed in a future paper.

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Nomenclature

F_R	friction force in direction of the z-axis, which is identical with the symmetry axis of the piston and the direction of the piston's translation	[N]
F_p	pressure force	[N]
F_N	normal force to the slipper	[N]
F_Q	radial force to the slipper/piston	[N]
F_{ax}	axial force to the slipper/piston	[N]
α	angle of the swashplate	[°]
a,b	distance	[mm]
μ	friction coefficient	[/]
σ	stress	[N/mm^2]
$h(x)$	gap height	[µm]
v	velocity	[mm/s]
p	pressure	[MPa]
x_i	axis of piezosensor in tangential direction to the pistons perimeter	
y_i	axis of piezosensor in radial direction to the pistons perimeter	
φ	angle position of the drive shaft	[°]
f_{bv}	based value for the friction ($= \frac{F_R}{F_p}$)	[/]

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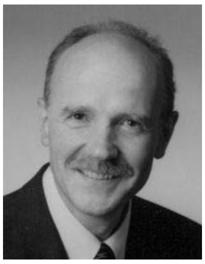

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