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WEAR AND FRICTION OF ZRC$_G$-COATED PISTONS OF AXIAL PISTON PUMPS

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Abstract

Within the Collaborative Research Center 442 "Environmental friendly tribological systems" components of different tribological systems are applied with PVD-coatings to realize the performance of these systems with biologically fast degradable fluids without increased friction and wear. With the usage of biologically fast degradable fluids without environmentally toxic additives some essential functions, that had been given by the former fluid, have to be substituted, what is realized by the provision of sliding parts with coatings. By that way wear and friction are decreased significantly. In this paper it is shown, that coated pistons of an axial piston machine reveal limited wear within the first hours of operation and very low friction between piston and bushing. The experiments are made on an one piston test bench.

Keywords: carbon, coatings, PVD, hydrostatic displacement unit, wear, friction

1 Introduction

Within the Collaborative Research Center 442 at RWTH Aachen University tribological systems are provided with coated parts to take over functions of mineral oil based fluids (Bebber 2003). This way the performance of these tribological systems with biologically fast degradable fluids based on native esters is made possible without increasing wear and friction (Bebber 2003).

The usage of fluids based on alternative resources than mineral oil gains more and more attention due to legislation and resource availability. The range of functions necessary for the performance of tribilogical systems is not given by biologically degradable fluids, because environmentally toxic additives, which are the bright majority of known additives, can not be used (Mang and Dresel 2001). To replace the functions of these chemical components PVD-coatings are an useful possibility due to the low disposition temperature (<200 °C) during the coating process. Applied within tribological systems caotings with increasing content of carbon as a solid lubricant can reduce wear and friction significantly, as shown in this publication. The tribological system described in this paper is the contact between piston and bushing of an axial piston pump. The experiments are made on an one piston test bench (Scharf and Murrenhoff 2005).

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In previous works different designs of one piston test benches have been used. Renius (Renius 1974, Renius 1973) and Breuer (see below) measured the friction between piston and bushing with a moving swashplate and a fixed bushing. Renius mounted the bushing with a hydrostatic bearing to separate the friction forces from the torques on piston and bushing and he realized a compensation piston to separate the friction forces from the pressure forces on the piston. He focused on high turning speeds and already used piezoelectric sensors for the axial friction forces and the tangential friction forces. He introduced the Guembel-Hersey-number to research works in the field of hydraulics. It is a very powerful tool to compare the friction behaviour of different tribologic systems including temperature, load and velocity.

Breuer took over the moving swashplate and the compensation piston, that he modified slightly, but replaced the hydrostatic bearing by a plate that was fixed to the housing by four piezoelectric sensors. He measured with very low turning speeds to see the characteristic behaviour of the tribologic system piston/bushing in the means of static and mixed friction. His works have been made in cooperation with the Linde AG and will be published soon.

Lasaar (Lasaar 2003) used also a hydrosatic bearing for the bushing, but he took a static swashplate and a moving cylinder block to measure also the friction due to the dynamic loads on the piston. He compared the measurement with numeric simulations, which are much closer to reality for high turning speeds due to the very small influence of mixed friction. He showed, that pistons with modified geometry may have less friction than pistons with a cylindric body.

Manring (Manring 1999) showed, that measurements of friction at low speeds and low pressure fit well with calculations based on the analytical analysis of the contact conditions between piston and bushing.

Van Bebber (Bebber 2003) demonstrated in his research work within the Collaborative Research Center 442 (http://www.sfb442.rwth-aachen.de/) at the RWTH Aachen University that different carbon containing PVD-coatings reduce friction and wear in tribological systems, especially in combination with not or less additivated biological fast degradable fluids that are based on native esters. In a first approach a Hafnium-carbide coating with a graded content of carbon (HfC_g) over the height of the coating showed a significant reduction in friction and wear. In a further step, due to the high cost of Hafnium, a Zirconium Carbide coating with graded carbon content, (ZrC_g), was developed and showed quite the same results. Both coatings are designed of the Surface Engineering Institute (IOT) at the RWTH Aachen University (Prof. Bobzin, http://www.iot.rwth-aachen.de/), which is a main partner within the Collaborative Research Center 442. Quick shots with ZrC_g-coated pistons in a complete hydrostatic displacement unit, which is in this case an axial piston pump, showed, that due to its complexity further sophisticated research on the tribological system piston/bushing with coated pistons is necessary and therefore a one piston test bench was designed and built up, that will be presented within the following chapters.

2 Test Bench

To measure the friction between a piston and bushing a one piston test bench (Fig. 1) was designed and built up at the IFAS following the works of Renius and Breuer. A detailed description is given in (Scharf and Murrenhoff 2005). A change has been made at the compensation piston, that formerly has been designed as Breuer did and now has a design like that one used by Renius (Renius 1973) as shown in Fig. 2. Between the bushing and the compensation piston a cylindric gap allows a minimized leakage to realize a maximum reduced friction between both bodies. At the other side the compensation piston is supported by the housing over a axial hydrostatic bearing. The surface of the compensation piston is designed such that a separation from the housing is suppressed (Renius 1973).

The aim of the experiments is to measure friction and the wear that corresponds to the specific load. To accelerate emerging wear the turning speed is choosen to be low (0.5 rpm - 5 rpm).

The test bench functions as follows: One piston moves inside a bushing, that is fixed over a plate to the housing by four piezoelectric force sensors. The compensation piston provides the bushing with pressure and transmits the pressure forces to the housing. By negligible small friction forces between compensation piston and plate the piezoelectric force sensors transmit exclusively the friction force, that is initiated between piston and bushing. The swashplate is moved by a hydrostatic unit with turning speeds between 0 and 10 rpm. The piston is moved out of the bushing by a low pressure of around 1MPa. The valveplate is replaced by a servovalve, that can be programed to simulate different pressure changing functions.

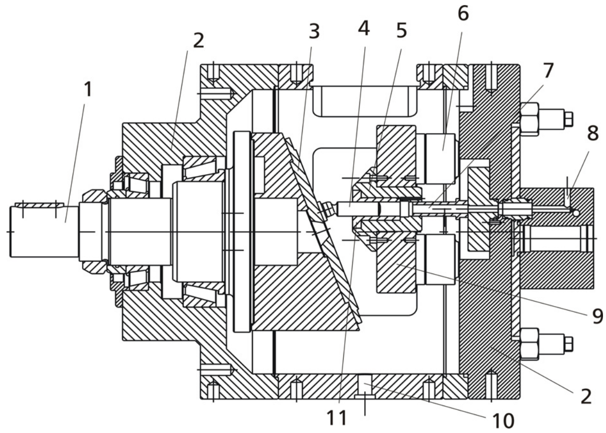

Fig. 1: *Cross section of the test setup; 1: drive shaft, 2: housing, 3: swash plate in wobble design, 4: piston, 5: bushing (with singleserving weared inner part), 6: 4 piezo sensors, 7: compensation piston, 8: position for adapting a valveplate alternative, 9: plate, 10: exit for leackage*

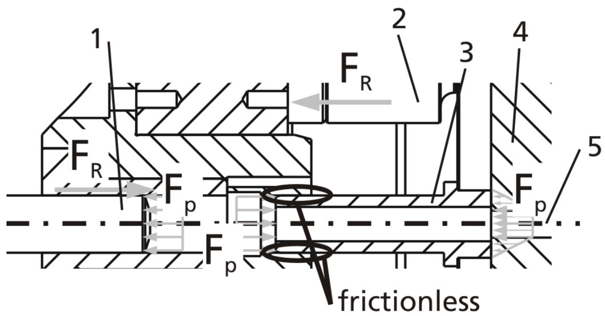

Fig. 2: *Balance of forces at the compensation piston: 1: piston; 2: piezo force sensor; 3: compensation piston; 4: housing; 5: pressure supply*

3 Kinematics

The forces attached to the piston can be sketched simply as shown in Fig. 3. Between the piston, that is tilted by the force F_Q, and the bushing mainly two contact forces F_1 and F_2 appear that induce the friction forces $F_{R,1}$ and $F_{R,2}$.

The resulting friction forces is given by Eq. 1. It depends on the direction of the piston movement, because the sign of the fraction of the denominator changes with the moving direction.

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

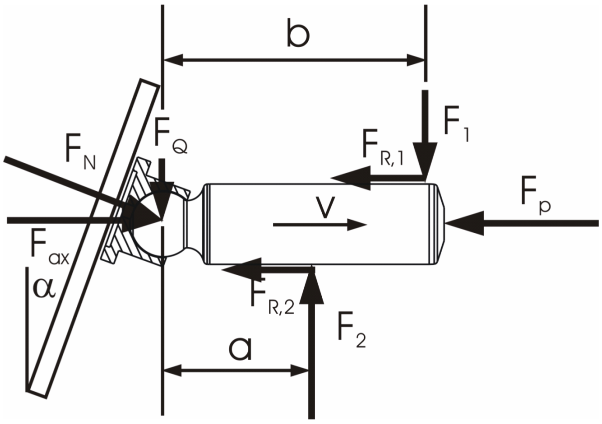

Fig. 3: *Balance of forces at the piston*

Thus a major effect is given that explains the typical friction measurement of pistons at low turning speed (Fig. 4).

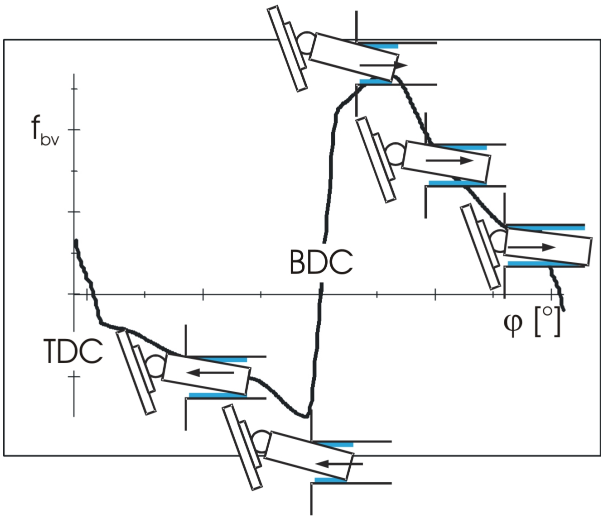

Fig. 4: *Typical friction progress for a piston/bushing contact at low turning speed*

The angle φ gives the position of the swashplate and f_{bv} is the friction force based on the pressure force F_p ($f_{bv} = \frac{F_R}{F_p}$) (Scharf and Murrenhoff 2005). During the motor modus, when the piston goes out of the bushing between the **T**op **D**ead **C**entre (TDC) and the **B**ottom **D**ead **C**entre (BDC), the absolute value of f_{bv} is much lower than during the pump modus between BDC and TDC. This special combination of pump and motor modus is no realistic performance modus and of pure academic interest.

4 Coating

The coating with a total height of 2 - 2.5µm is ZrC_g with either a spline shaped carbon gradient (Fig. 5) or a linear shaped carbon gradient. The point of maximum hardness is in the middle area of the ZrC_g layer or near the top layer. This means that the surface is exhibited to abrasive wear until the resistance against wear is, due to increasing hardness, high enough to stop the continuous loss of material.

A second effect that leads to a reduction of wear is the increasing size of the contact areas with increasing loss of material as a consequence of wear. Contact forces, that are integrations of the pressure distributed within the contact area, stay constant while the pressure decreases with growing sizes of these contact areas. So the wear causing load falls.

Fig. 5: *Carbon content and hardness of a ZrC_g-coating*

5 Fluid

The fluid used for the measurements of the coated piston is a biologically fast degradable synthetic ester without any additives. The temperature is held at a constant 40°C at the pressure connection. At that temperature the fluid has a kinetic viscosity of $\nu = 31$ cSt.

6 Cylindrical Piston

6.1 Not Coated Piston

As a reference piston a not coated piston of 42CrMo4 + QT (500 HV) was measured in a bushing of Eterna, which is a hard brass alloy. After a running in cycle of 8 h at 30 MPa and n = 1 rpm both components show traces of wear (Fig. 6). On the piston wear is symmetric to the piston cylinder axis and at the bushing it appears helical due to the movement of the inclined piston. The surface of the bushing is quite rough due to the production at the workshop which is not the same as in the series production. But it is obvious that at contact points the roughness is leveled to very low values.

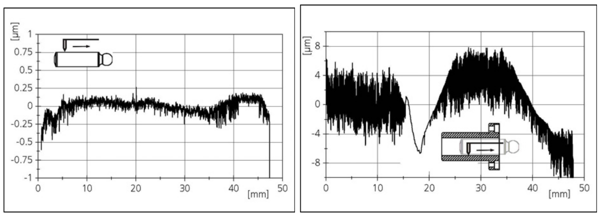

Fig. 6: *Wear at piston (left) and bushing (right) after running in cycle at 1 1/min*

The measurement of the section is done with a profilometer to indicate the locations of lost material and the adjustment of the profiles to the load situation between piston and bushing.

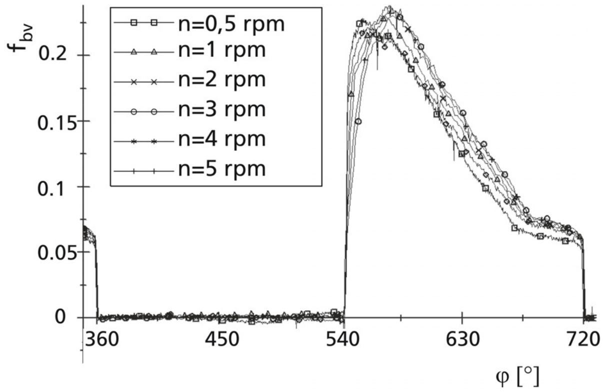

Fig. 7: *Friction between steel piston and brass bushing at a pressure of 30 MPa*

Friction in pump modus has a maximum at $f_{bv} = 0.225$ at the BDC (Fig. 7) that stays almost constant in spite of changing engine speed. With increasing squeeze effect the maximum is delayed, but in all cases it keeps mixed friction. The fluid used for the measurement is a HLP VG 46 at 36°C tank temperature and the pressure is at 30 MPa.

6.2 Coated Pistons

The coated pistons used for measurements are provided with two different variations of ZrC_g-coating. The maximum of hardness within the coating 1009.ZrC_g.09 is located close the surface whereas within the coating 1006.ZrC_g.06 it is located at medium coating height. The difference in wear and friction will be shown below. Both pistons run in bushings of 42CrMo4 + QT (500 HV).

The experiments start with a running in period that is controlled by repeated measurements of the section of the profile of piston and bushing. The engine speed is fixed at n = 1 rpm to have a compromise between sufficient friction and numbers of driven cycles within an acceptable time. The starting pressure is p = 10 MPa that is increased to 20 MPa and 30 MPa when with the actual constraints wear stopped.

<u>1009.ZrC_g.09:</u>

Due to the stoichiometric point located near to the surface the mass volume for geometry adjusting wear is small. The measurement shows that wear at piston and bushing is negligible (Fig. 8). Due to the missing adjustment of the surfaces to the load situation the friction, shown as f_{bv}, is even above the values of the reference piston. Especially in this case a partial adjustment would have been necessary because the profile of the bushing is far beyond of being perfect. The production of the surface of the bushing demands a high level of experience that at this state of the works was not yet existing.

Figure 9 demonstrates that the friction between piston and bushing is dominated by mixed friction, because the measurement gives no hint for changes, that depend of the velocity, which would show the partial existence of hydrodynamic lubrication. The main contact between both bodies is located at the right end of the bushing as shown in Fig. 8. Within the remaining parts of the gap pressure cannot be built up, because the fluid between piston and bushing is sqeezed out quickly through rather large and not reducing cavities.

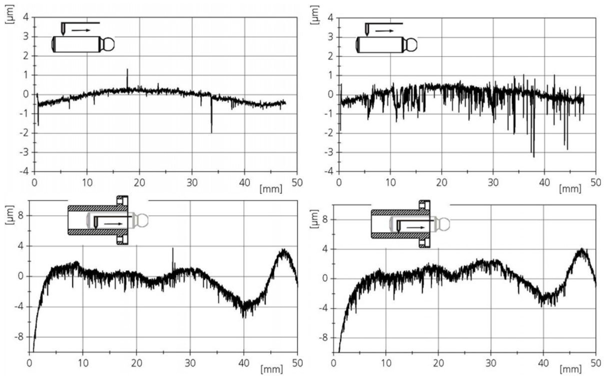

Fig. 8: *Profile of a 1009.ZrC_g.09-coated piston in new condition (left, top) and after running in (right, top) and the bushing of 42CrMo4+QT (500 HV) in new condition (left, bottom) and after running in (right, bottom)*

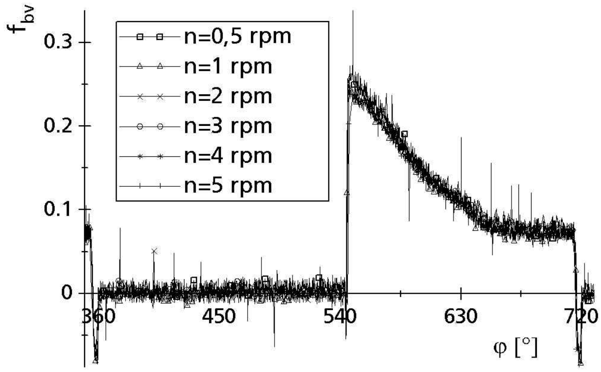

Fig. 9: *Friction between the 1009.ZrC_g.09-coated piston and bushing of 42CrMo4+QT (500 HV) at a pressure of 30 MPa*

<u>1006.ZrC_g.06:</u>

This coating is softer at the top layer area and allows limited wear. With increasing wear the hardness of the contact surface rises and builds up a stronger resistance against wear that after a certain time of performance wear reaches a final limit.

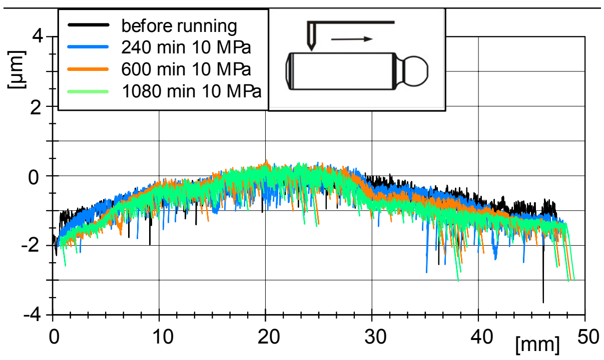

Fig. 10: *Profile of a ZrC_g-coated piston running at 1 rpm and 10 MPa*

Figure 10 illustrates the wear of a 1006.ZrC_g.06-coated piston performing at a pressure of 10Mpa and unit speed of 1 rpm. Abrasion mainly occurs in the dead centers of the piston movement, especially the bottom dead center, where sharp edges initiate stress on a very small distribution area.

With increasing pressure abrasion is placed towards the middle area of the piston. At both ends the profile already found its final shape (Fig. 11).

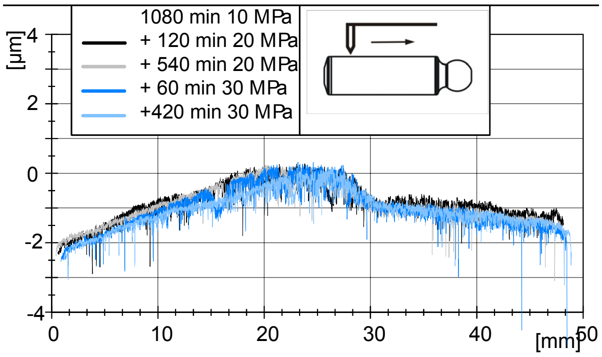

Fig. 11: *Profile of a ZrC_g-coated piston running at 1 rpm and 20 Mpa – 30 MPa*

Figure 12 shows the pistons position in the dead centers of the piston movement with the according marks within the piston's profile.

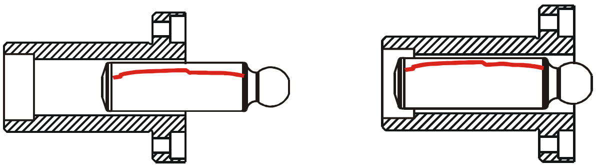

Fig. 12: *Characteristic profile marks due to the dead centers (left: BDC, right: TDC)*

The bushing (42CrMo4V+QT, 500 HV), shows much smaller wear. The position illustrated in Fig. 13 is the highest charged position, according to Fig. 12, left, and reveals a profile much less changed than the piston profile.

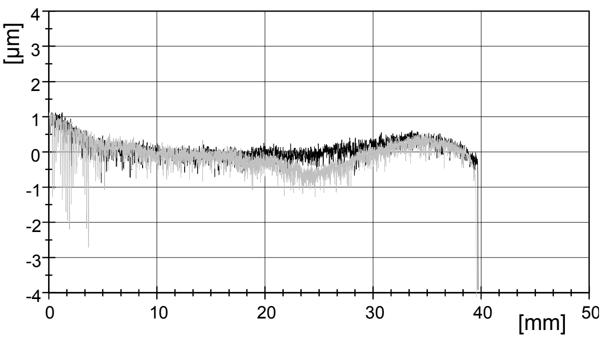

Fig. 13: *Bushing wear at a characteristic point*

The 1006.ZrC_g.06-coated piston shows a maximum value of f_{bv} = 0.15 at the bottom dead center at 30 Mpa (Fig. 14). With increasing engine speed the inclination of the friction force graph right after 540° gets smaller due to the squeeze effect. This effect describes the squeezing out of the fluid from the reduced volume between piston and bushing that is retarding the inclination speed of the piston and therefor also the direct contact between both solids. The reason for the maximum of friction force at the bottom dead center is that at this position the distance (b-a) (Fig. 3) has its minimum and F_1 and F_2, as idealised forces, a maximum.

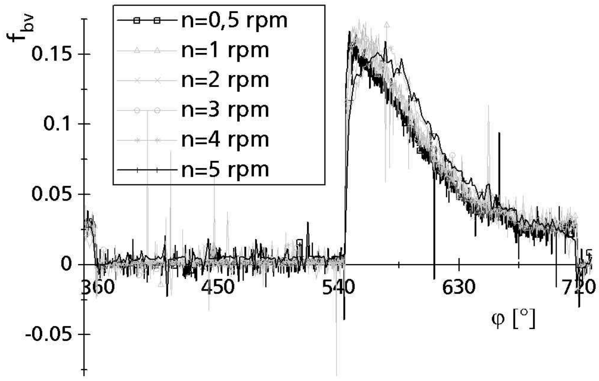

Fig. 14: *Friction between1006.ZrC_g.06-coated piston and bushing (42CrMo4V+QT, 500 HV) at 1 rpm*

The low friction compared with the reference piston demonstrates the friction reducing characteristic of ZrC_g. In both tribologic systems (steel piston/brass bushing and coated piston/steel bushing) the material combination allows an adjustment via wear, that supports convenient geometric conditions and a state of sufficient lubrication. That the system with the coated piston shows the lower friction and wear strongly verifies that the combination of ZrC_g/steel with not additivated biological fast degradable fluids exceeds the system steel piston/brass bushing with additivated mineral oil in means of tribology. A difference in wear is that at the compared steel/brass combination it mainly appears at the bushing (Fig. 6) and is not symmetric to the moving axis of the piston where at the coated combination it mainly emerges at the piston and is symmetric to its moving axis.

6.3 Friction Characteristics with Changing Temperature

Renius introduced the Guembel-Hersey-number ($\frac{\eta \cdot \omega}{p}$) (Renius, 1974) to the analysis of measurements of axial piston machines to give the possibility to compare friction over pressure, velocity and temperature, that has the major influence on the viscosity. The temperature is measured with a thermoelement that is fixed in a borehole close to the bushing surface.

The maximum friction value f_{bv} within one engine cycle of a steel piston in a brass bushing and the 1006.ZrCg.06-coated piston in a steel bushing (42CrMo4V + QT, 500 HV) listed over the Guembel-Hersey-number (Fig. 15) give a impressive image of the lower friction and therefor better tribological characteristics of coated pistons against others over a rather wide range of operating conditions.

Due to the slow engine speed of the test bench triction is still in the range of mixed friction and only experiments with increased velocity will give the point of change between mixed friction and hydrodynamic friction as shown in the corner at the top right of Fig. 15 with measurements of a rotational tribometre as it is explained in chapter 4.5.

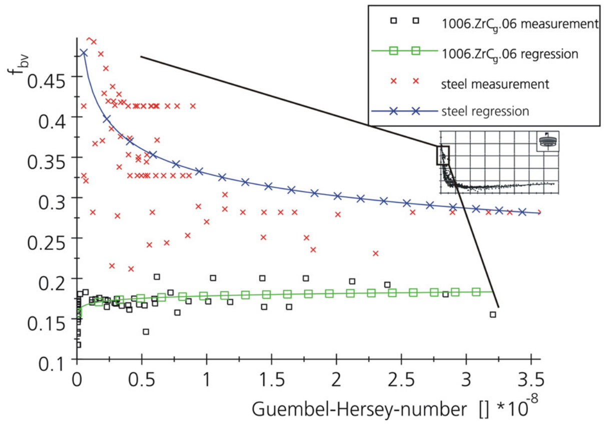

Fig. 15: *Bushing wear at a characteristic point*

7 Coated Piston with Shortened Contact Surface

ZrC_g coated piston with shortened contact surface are higher charged because the distance (b-a) between F_1 and F_2 from Fig. 3 gets smaller. Anyhow the coating withstands these extra load situation and even finds its final shape during duration runs at 10 MPa (Fig. 16). During further runs at 20 MPa and 30 MPa additional wear does not appear in a significant way. Analyses of hardness and material composition of the weared surface are still going on and will be subject of future publications. The initial intention was to apply a diametre reduction of just a few µm to smoothen the contact between piston and bushing, but that could not yet be realized due to the available production machines.

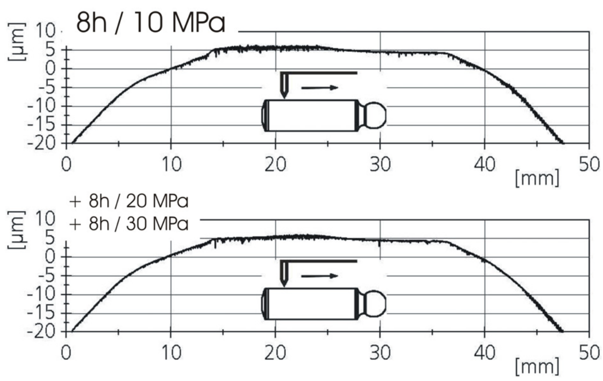

Fig. 16: *Wear on a ZrCg-coated piston with shortened contact surface after 8h at 10 MPa (top) and after additional 8h at 20 MPa and 8h at 30 MPa (bottom)*

8 Wear of Coated Specimens in a New Bio Fluid

The previous chapters gave a glance of the functionality of coated pistons with now available bio fluids. But within the Collaborative Research Center new fluids are developed as well as coatings to fit the requirements of a tribological system. Therefore at the Institute for Technical Chemistry and Heterogene Catalyse at RWTH Aachen University, which is a partner institute within the Collaborative Research Center 442, HISM has been designed (Liauw et. al. 2005). HISM is a biologically fast degradable fluid (Hydroxyisopropanlstearinacidester) that is treated by heterogene catalyse to diversify its characteristics instead of adding additives and is based on native esters as educts.

Former experiments with specimens proposed by Siebel and Kehl (Fig. 17) demonstrate that wear within the coated system stops after a certain time of performance. This results have to be redone by the new developed combination of material (ZrC_g/42CrMo4V + QT/HISM) to hit the goal of the research work of the Collabarative Research Center 442. The principle of the experiments of Siebel and Kehl is to press two ring formed surfaces against each other and one of them is turned by an hydrostatic displacement unit.

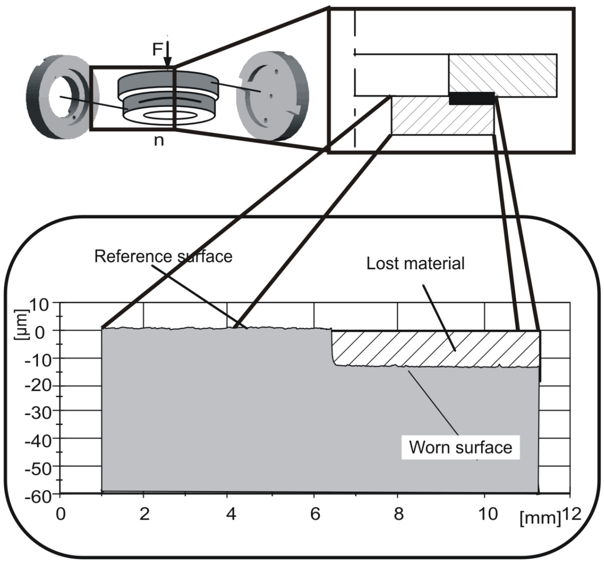

Fig. 17: *Specimen and principle of Siebel and Kehl (Bebber, 2003)*

The smaller ZrC_g-coated specimen, in a first step with a 1009.ZrCg.09 coating with the hard top layer, shows allmost no wear. The surface seems to be untouched (Fig. 18).

In spite of increasing load wear at the uncoated specimen (Fig. 19) is limited (Fig. 20). As a result it can be resumed: The combination of HISM and a ZrC_g-coating substitutes the function of additivated mineral oil based fluids paired with the materials brass/steel as far as wear is concerned. Further experiments have to confirm these results with a 1006.ZrC_g.06-coated piston performing with HISM as working fluid.

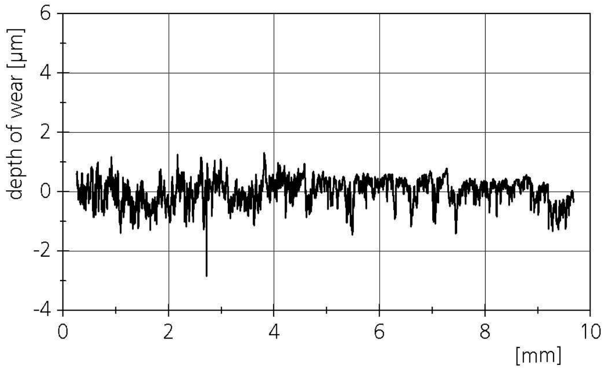

Fig. 18: *Wear at the coated specimen; the charged surface reaches from 5 mm to 10 mm*

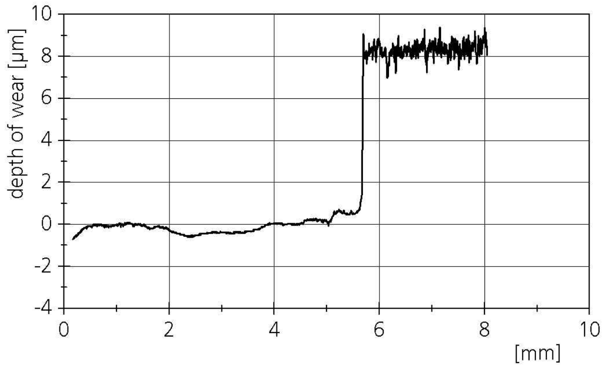

Fig. 19: *Wear at the uncoated specimen (42CrMo4V+QT, 500 HV)*

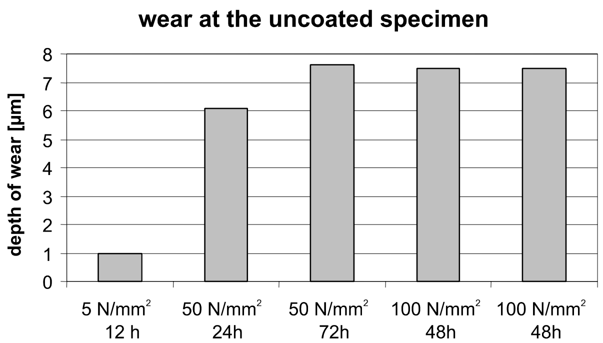

Fig. 20: *Wear at the uncoated specimen (42CrMo4V+QT, 500 HV) for increasing load at n = 70 rpm ≅ 0,2 m/s*

9 Conclusion and Outlook

Coatings with increasing content of carbon have the potential to lower friction within tribological systems of hydrostatic displacement units in an essential way for different loads, engine speed and temperature. In combination with biologically fast degradable fluids the possibility is given to use hydrostatic machines in environmentally sensitive areas and to rise the efficiency factor.

Wear during a running in period is necessary as well as a geometry that guarantees sufficient large contact areas between piston and bushing. To complete this geometry, that might be prepared during production as outlined within the next paragraphs, a coating with limited wear gives the best possibilities as described in this paper.

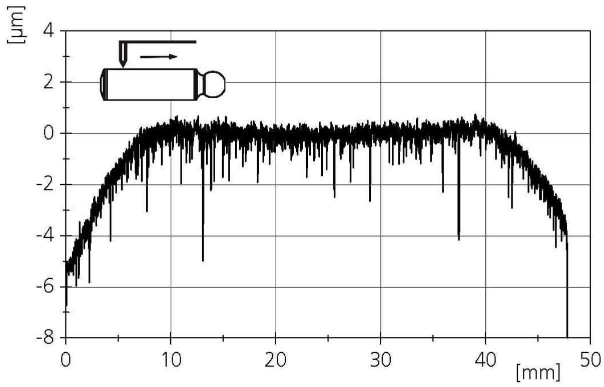

Fig. 21: *Profile of preshaped piston using a grinding machine*

Preshaped piston and bushings may even optimize these results. With modern grinding machines grinding discs are easily formed to give a user defined shape to produce pistons with arbitrary profiles (Fig. 21). This will be discussed in further publications.

References

Bebber, D. van 2003 (1). *PVD-Schichten in Verdrängereinheiten zur Verschleiß- und Reibungsminimierung bei Betrieb mit synthetischen Estern*, Dissertation, RWTH Aachen.

Lasaar, R. 2003. *Eine Untersuchung zur mikro- und makrogeometrischen Gestaltung der Kolben-/Zylinderbaugruppe von Schrägscheibenmaschinen*, Fortschritt- Berichte VDI Nr. 364.

Lasaar, R. and **Ivantysynova, M.** 2004. An investigation into micro- and macrogeometric design of piston-/cylinder assembly of swash plate machines, *International Journal of Fluid Power 5*, No. 1.

Liauw, M. A., **Eichholz, S.** and **Sabater, S.** 2005. Process Optimization in Synthesis of Lubricants: From OME to HISM, *Prieto Proceedings of AIChE Annual Meeting*, Cincinnati, OH, October 30-November 4, 2005.

Mang, T. and **Dresel, W.** 2001. *Lubricants and Lubrication*, Weinheim: Wiley-VCH.

Manring, N. D. 1999. Friction Forces Within the Cylinder Bores of Swash-Plate Type Axial-Piston Pumps and Motors, *ASME Journal of Dynamic Systems, Measurement and Control*, 121: 531-37.

Renius, K. T. 1973. Experimentelle Untersuchungen an Gleitschuhen von Axialkolbenmaschinen, *O+P Ölhydraulik + Pneumatik* 17, Nr. 3.

Renius, K. T. 1974. *Untersuchung zur Reibung zwischen Kolben und Zylinder bei Schrägscheiben-*

Axialkolbenmaschinen, VDI - Forschungsheft 561, VDI – Verlag, Düsseldorf.

Scharf, S. and **Murrenhoff, H.** 2005. Measurement of Friction Forces between Piston and Bushing of an Axial Piston Displacement Unit, *International Journal of Fluid Power* 6 No. 1, pp. 7-17.

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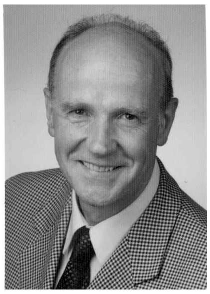

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