# VALIDATION OF A NEW TRIBOLOGICAL TEST BENCH FOR LIGHTWEIGHT HYDRAULIC COMPONENTS

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ABSTRACT

The presented paper deals with tribological contacts in "lightweight" hydraulic pumps. One approach for the mass reduction of hydraulic systems is the material substitution. According to this strategy, steel-components are replaced with polymeric material or hybrid design components. The consequences of this material substitution are tribological changes of the contact pairings. Hence, there is research necessary on the tribological behavior of the specific contacts between polymers or hybrid components. For the characterization of these tribological contacts, a test bench was developed. With this test bench the analysis of the tribological behavior of the contact between cylinder and control plate in the axial piston variable pump is possible. A description of the modeling and a validation of the test bench are presented in the this paper. The result is, the test bench indicates good measurement result quality. In addition a cooling system is necessary to guarantee a constant test fluid temperature.

Keywords: hydraulics, lightweight design, tribological testing

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## INTRODUCTION / MOTIVATION

In mobile systems, masses in motion are one of the biggest contributors for C02 emission. Due to this fact, especially for automotive applications, reducing masses in motion becomes even more relevant.

Present paper deals with research on the tribological contacts occurring in so called lightweight hydraulic systems. Hydraulic systems often are subject to oversizing, leading to unnecessary high masses of the systems and contributing to high costs. In the first steps of this work the lightweight potential of hydraulic systems was studied. For the validation of the changes of the properties due to material substitution the redesigned components have to be investigated experimentally. As a result, the main component of a hydraulic power train - an axial piston variable pump (Fig. 1) - is taken under investigation.

Replacing steel-components with polymeric material or hybrid design components could be a possibility for the mass reduction of an axial piston variable pump. This approach consequently leads to tribological changes of the system as polymeric or hybrid design components behavior in this application is insufficiently known.

In order to develop scientific understanding of the tribological contacts of an axial piston variable pump made of lightweight materials the test bench RPR (from the German acronym: ReibungsPrüfstand Rotatorisch / rotatory friction test bench) was developed at our IPEK. With the test bench RPR the analysis of the tribological behavior of the contact between cylinder (Fig. 1, number 1) and control plate (Fig. 1, number 2) is possible. The test bench aims at modelling tribological conditions observed in the real system.

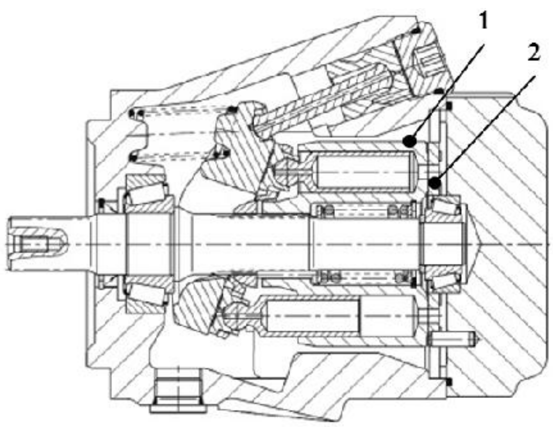


*Figure 1. Axial piston variable pump, design Bosch-Rexroth, type A10VNO [1].*

In the following section 2, a summary of the state of the art of the intended investigations is shown. In section 3 a comparison between the test bench model and the real system is presented before design and function of the developed test bench is presented in section 4. Results from experimental validation activities follow in section 5.

## STATE OF THE ART

There are various materials thinkable and available to decrease the overall mass of the system. The principle suitability of ceramics and polymers for their application in hydraulic systems is shown by Donders et al. [3].

The use of ceramics for components of hydraulic systems is discussed by Bartelt et al. [4] and Feldmann [5] and Schöpke [6]. Bartelt et al. [4] describe the development of test procedures for components of hydraulic systems made of ceramics. Feldmann [5] shows which developments in the field of ceramics could be of interest for hydraulic systems. A consequence of this work is that there is a potential of common ceramics for the use in components of hydraulic systems. He also shows that development is necessary to improve the components of hydraulic systems. Schöpke [6] works on simulation and testing of ceramics in the contact between cylinder and control plate of an axial piston pump.

Investigations on PVD-coated steel-components for the optimization of friction and wear in the tribological contacts of a hydraulic axial piston pump are shown by Murrenhoff / Scharf [7], van Bebber [8] and Kleist et al. [9]. PVD-coated components of a hydraulic axial piston pump are tested with test benches by Murrenhoff / Scharf [7]. Their main finding is that there are PVD-coatings with high potential for the use in components of hydraulic axial piston pumps. The scientific paper of van Bebber [8] also contains investigations on PVD-coated pistons and control plate of a hydraulic axial piston pump and shows that with PVD-coatings a minimization of friction and wear is possible. The result of the investigations of Kleist et al. [9] is that the properties of PVD-coatings have to be adapted to the case of application in hydraulic systems.

A multitude of polymers have been tested by Künkel [10] on a pin-on-disk test bench as well as on a journal bearing test bench. This way he gains knowledge for the estimation of the tribological behavior of the tested materials.

The general tribology of polymers, polymer composites and ceramics is studied in [11]. In [12] especially tribological tests and results are displayed.

Pin-on-disk sliding friction and wear experiments were conducted on two different titanium alloys by Qu et al. [13]. Disks of titanium alloys were slid against fixed bearing balls composed of stainless steel, silicon nitride, alumina, and PTFE. One of the results was that a higher friction coefficient with larger fluctuation and higher wear rate was observed at lower sliding speed. Another result was that ceramic sliders experienced much higher wear and created more wear on the counterfaces than the stainless steel sliders did.

The design of the test bench RPR with a first measurement result is shown in [15]. Present paper supplements measurement results for validation of the test bench RPR.

## TEST BENCH MODEL

During the pumping, the cylinder (Fig. 1, number 1) of the axial piston variable pump is rotating. The control plate (Fig. 1, number 2) is fixed. As a result, the model of the tribological test bench is realized with a nonrotating disk (disk 1 in Fig 2) applying a pressure load $F_D$ on a rotating disk 2 according to Fig. 2.

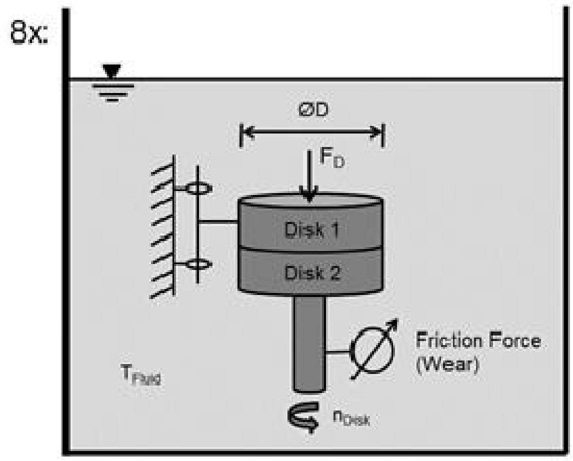


*Figure 2. Modeling of the test bench RPR.*

The surface design of the disks can be designed similar to the surface design of the disks used in the real pump. That means the disks can be drilled like the cylinder and control plate in the axial piston variable pump. Also the surface roughness of disks can be adapted for different investigations. Both disks can be made of various lightweight materials.

The RPR characteristics are displayed in Table 1. The range of the disk diameter $D$ gives the possibility to test the tribological behavior between different diameters for cylinder and control plate made of lightweight materials. With the ranges of the characteristics $F_D$ and $n_{Disk}$ (cp. Table 1), tests for different pump sizes in different operating modes are possible.

Disk 1 and 2 are working under oil-bath lubrication with various possible lubricants for tests. The lubricant’s temperature is regulated by means of a heating system within a certain range also given in Table 1.

There is a difference between the form of lubrication of the test bench RPR and the actual one of the tribological contact between cylinder and control plate in the axial piston variable pump. Due to the fact that there is oil pressure inside the pump there is in the real system a pressure-feed oil lubrication between the cylinder and the control plate with variable pressure. Compared to the real lubrication case, an oil bath lubrication induces more friction and wear. The reason for this is that the hydrodynamic load of the lubricant is higher at pressure-feed oil lubrication. On this account the tests achieved with the RPR take into account the worse case occurring in the pump.

Another difference between the test bench model and the original tribological contact in the real pump is the possible temperature of the lubricant: when the axial piston variable pump operates in a cold environment, the temperature of the lubricant can be below 20°C at the beginning, a special case which is not yet taken into account with the actual test bench.

In addition, there is a constant pressure load $F_D$. In the pump the pressure load is not kept constant over the contact area between cylinder and control plate. This is because of the issue that one fraction of the pistons is pushing the oil and the others are sucking the oil while the cylinder is rotating. The constant pressure load of the RPR test bench is to guarantee a good result quality.

*Table 1. Characteristics of the test bench RPR.*

| | |
|---|---|
| Diameter of the test disks $D$ | 50 – 100mm |
| Pressure Load $F_D$ | < 6kN |
| Rotating speed $n_{Disk}$ | < 4000 1/min |
| Oil lubrication process | Oil-bath lubrication |
| Lubricant Heating Temperature $T_{Fluid}$ | 20 – 120°C |
| Number of tribological contacts during simultaneous testing | 8 |

There is the possibility to measure the friction force between disk 1 and disk 2. Wear measurement is realized in an indirect way: disk 1 and 2 are weighed before and after defined testing cycles.

To get a sufficient statistical probability distribution of the test results the test bench RPR features a testing of eight specimen with the same testing-parameters at the same time.

## DESIGN OF THE TEST BENCH

Fig. 3 shows the design of the RPR test bench. The main component of the test bench RPR is the test chamber (Fig. 3, number 1) appended on a table (Fig. 3, number 6). The test chamber features eight mounts for the test components for simultaneous testing (Fig. 3, number 2) in order to guarantee reproducible results. Currently two of the eight mounts for test components are equipped with sensors for friction force measurement (Fig. 3, number 3). It is possible to equip additional mounts with friction force measurement system. The test chamber is driven by an electrical drive (Fig. 3, number 5) with belt transmission (Fig. 3, number 4).

Fig. 4 shows the test chamber in sectional view. Only two of the eight disk-on-disk-contacts of the test chamber are visible in this figure. The test components are at positions 4 and 5 of Fig. 4. Position 4 is the nonrotating and position 5 the rotating test component according to Fig. 2.

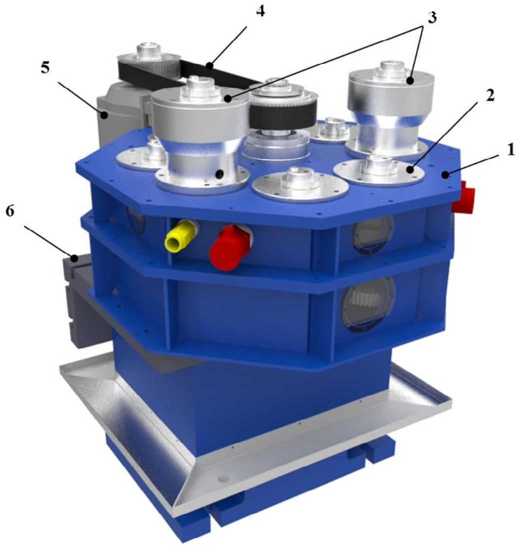


*Figure 3. Design of the test bench RPR.*

The input shaft (Fig. 5, number 1) is driven by electrical drive via belt transmission. The gear drive (Fig. 5, number 2) transmits the power to the shafts connected to the rotating test components (Fig. 5, number 3).

The nonrotating test components are guided in radial direction. A particular designed torque damper inhibits the rotation and allows the axial movement of the non-rotating test components according to the principle sketch in Fig. 2 This is necessary in order to keep normal load constant in case of wear during the test.

Fig. 4 shows the torque damper mounted to the friction force measurement assembly. The torque damper consists of a disk (Fig. 4, number 1) which is guided in radial direction by the housing (Fig. 4, number 2). At the outer-diameter of the disk there is a notch for a pin (Fig. 4, number 3) which is fixed with the housing. The pin inhibits the rotation and allows the axial movement of the disk. The disk is conjunct with the friction force measurement system (Fig. 4, number 4) on the one side. At the other side of the friction force measurement system there is a coupling disk (Fig. 4, number 5) guided in radial direction with a roller bearing (Fig. 4, number 6). The coupling disk guides the non-rotating test components (Fig. 4, number 7) in radial direction and inhibits the rotation of the nonrotating test components. Additionally notches on the outer-diameter of the non-rotating disk (Fig. 6) are necessary for the coupling disk. As parts number 8 and 9 are not in contact (Fig. 4) with the housing, they do not influence the friction force measurement. The torque damper connected to the test components without friction force measurement works in the same way but without friction force measurement system.

The pressure load $F_D$ according to Fig. 2 is generated by hydraulic cylinders (Fig. 4, number 6). A constant pressure load is guaranteed by a particular closed loop control with a pressure proportioning valve.

For the friction force measurement a torque sensor (Fig. 5, number 7) is integrated in the torque damper of the nonrotating test component. The torque sensor is manufactured by NCTE (Series 2000) [2]. The measurement principle of the torque sensor is inverse magnetostriction. This measurement principle guarantees a stable measurement of the friction force.

Fig. 5, number 8 shows the available volume for the lubrication fluid for the tests. There is the possibility for heating the lubrication fluid with a flow heater. A constant test fluid temperature inside the test chamber is guaranteed by a particular temperature regulation loop with type J thermocouples in the test chamber. The test fluid is sealed from the oil bath of the drive gear (Fig. 5, number 9). A contamination of the test fluid by the oil bath of the drive gear is prevented.

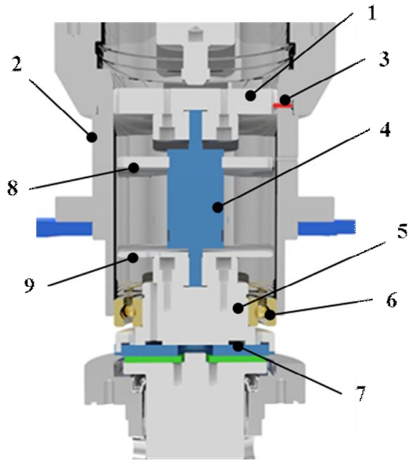


*Figure 4. Torque damper for the non-rotating test components with friction force measurement system.*

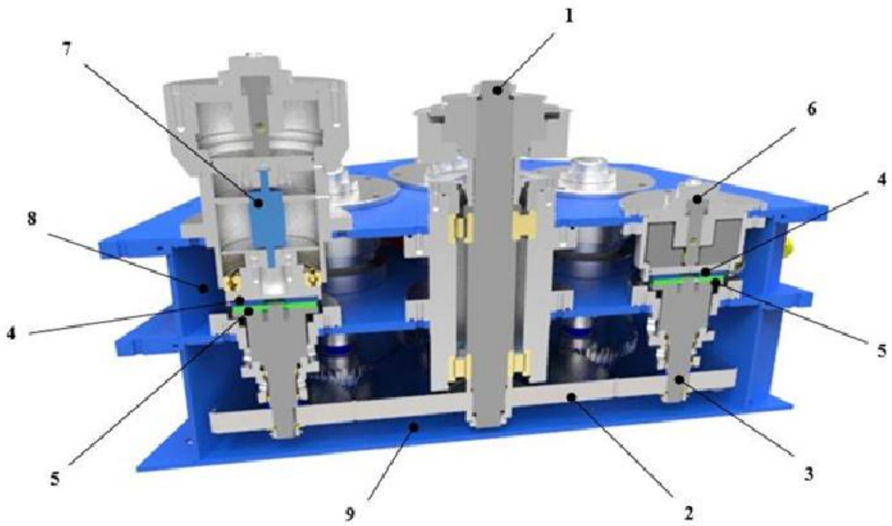


*Figure 5. Sectional view test chamber test bench RPR.*

VALIDATION

For the validation of the test bench RPR, test components made of common materials for the cylinder and control plate in axial piston variable pumps are used. Table 2 illustrates the validation parameters.

The geometries of the nonrotating and rotating disks are displayed in Fig. 6. The total diameter of the nonrotating disk is greater than the outer diameter $D_a$ of the rotating disk. The reason for this is that there are notches necessary on the outside diameter of the nonrotating disk. The notches are for the torque damper and outside of the tribological contact.

*Table 2. Validation parameters test bench RPR.*

| | |
|---|---|
| Material rotating disk | brass |
| Material non-rotating disk | nodular graphite cast iron |
| Lubricant | H-LP 32 (DIN 51524) |
| Diameter of the disk $D_a$ | 83,5mm |
| Diameter of the disk $D_i$ | 40mm |
| Pressure load $F_D$ | 350N |

Fig. 7 shows the defined test procedure for the validation of the test bench RPR. There is a first rotational speed ramp from 0 up to 4000rpm with a duration of 12 minutes. After a short time at 4000rpm there is another ramp from 4000rpm down to 0 also with duration of 12 minutes. As this procedure is repeated 3 times, there are 3 ramps from 0 to 4000rpm and 3 ramps from 4000rpm to 0rpm. Over each of the 6 ramps the friction torque is measured with both mounts of the test bench equipped with friction force measurement system. The friction coefficient is calculated by integration of the measured friction torque over the area between Diameter $D_a$ and $D_i$ of the rotating disk. As the tests are displaying a qualitative comparison, there is no necessity to take into account the exact contact area. As a consequence, the holes of the rotating disks are not taken into account for the friction coefficient calculation.

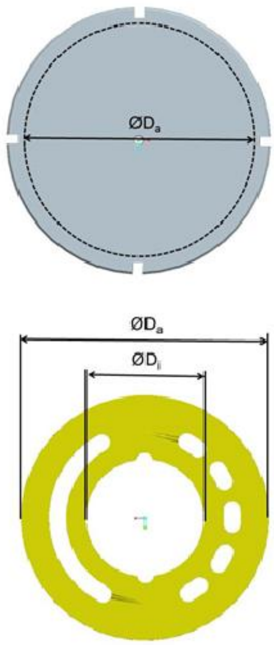


*Figure 6. Geometry of the nonrotating disk (left) and rotating disk (right).*

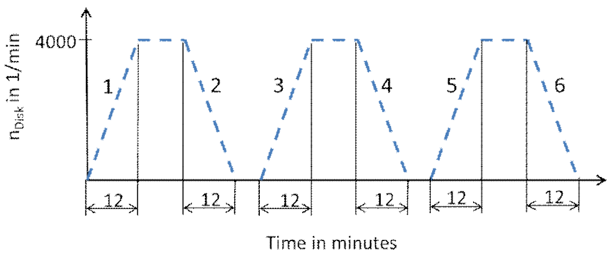


*Figure 7. Validation test procedure for the test bench RPR.*

In Fig. 8, Fig. 9 and Fig. 10 the curve of the friction coefficient is shown for different rotational speed ramps. The friction coefficients 1 and 2 are the friction coefficients measured on mount 1 and 2 equipped with friction force measurement system. The figures also show the arithmetic averages of the measured friction coefficients and the temperature of the oil bath lubrication.

If we compare Fig. 8 with Fig. 9 there is a higher difference between both measured friction coefficients in the rotating speed area between 0 and 2500rpm for rotating speed ramp 2. The curve of the arithmetic averages of the friction coefficients in Fig. 8 and Fig. 9

are close together over all rotating speeds. The same also for all other rotating speed ramps. When the ramp goes from 0 up to 4000rpm the friction coefficients are closer together compared to the case when the ramp goes from 4000rpm down to 0.

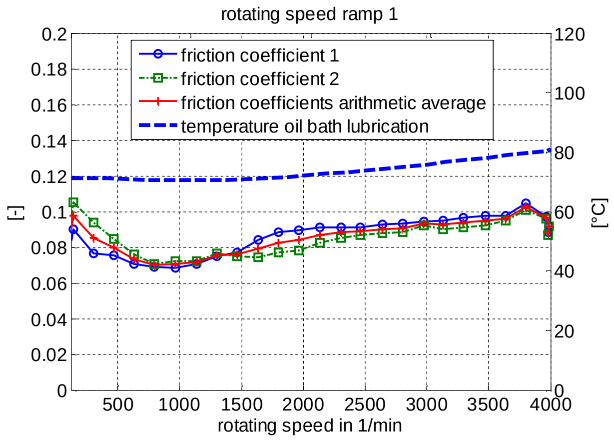


*Figure 8. Friction coefficients and temperature of the oil bath lubrication at rotating speed ramp 1.*

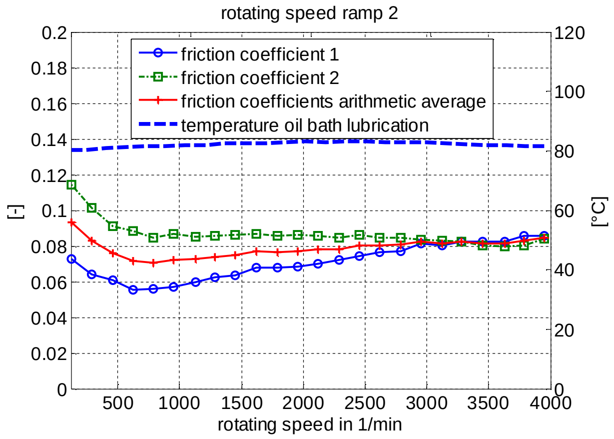


*Figure 9. Friction coefficients and temperature of the oil bath lubrication at rotating speed ramp 2.*

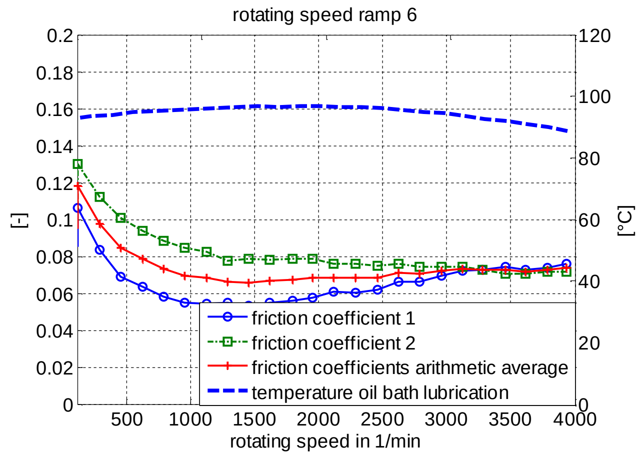


*Figure 10. Friction coefficients and temperature of the oil bath lubrication at rotating speed ramp 6.*

In Fig. 8, Fig. 9 and Fig. 10 the curve of the oil bath temperature is not constant at all rotating speeds. This is because the friction heats up the oil bath and test bench. If we compare the arithmetic averages of the friction coefficients in the figures it is visible, that the influence of the temperature difference of the oil bath on the tribological contact is not as high.

In Fig. 11 the arithmetic averages of the measured friction coefficients of all of the 6 rotational speed ramps are visible. The friction coefficients are at low rotational speeds in the area of 0.11 and at higher rotational speeds in the area of 0.08. Hence, the measured friction coefficients are in the area of the friction coefficients between grey cast iron and bronze for lubricated contacts and for static and sliding friction given for example in [14].

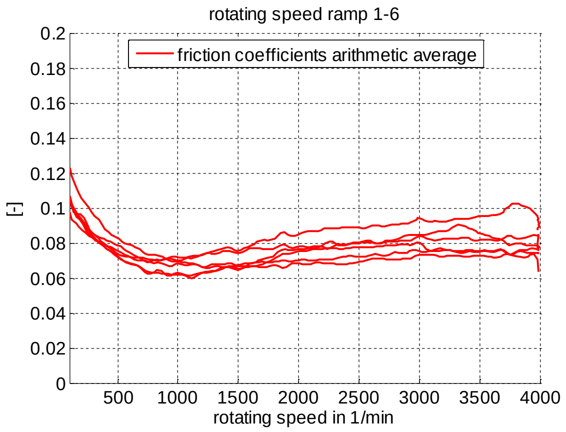


*Figure 11. Arithmetic averages of the measured friction coefficients at rotating speed ramp 1 to 6.*

In Fig. 11 there are no characteristics for hydrodynamic friction behavior visible. The measured friction coefficients are almost constant for higher rotating speeds. The reason for this may be the geometry of the test components. Compared to conventional axial bearings there are no wedges or step shaped geometries inducing hydrodynamic forces.

After the measurements described above, the positions of the test components were changed in such way that the test components of mount 1 moved to mount 2 and the test components from mount 2 moved to mount 1 equipped with friction force measurement system. With the new arrangement of the test pieces and with the same test surfaces than the earlier test the test procedure in Fig. 7 was conducted again. The reason for changing the test components was to check if there is a difference between the measurement result of the friction force measurement system of mount 1 and 2.

In Fig. 12 the arithmetic averages of the measured friction coefficients of all of the 6 rotational speed ramps with the new test component arrangement are visible. By comparing Fig. 11 and Fig. 12 for higher rotating speed rates than 750rpm the arithmetic averages of the measured friction coefficients are in the area of 0.08. For smaller rotating speed rates than 750rpm they are divergent.

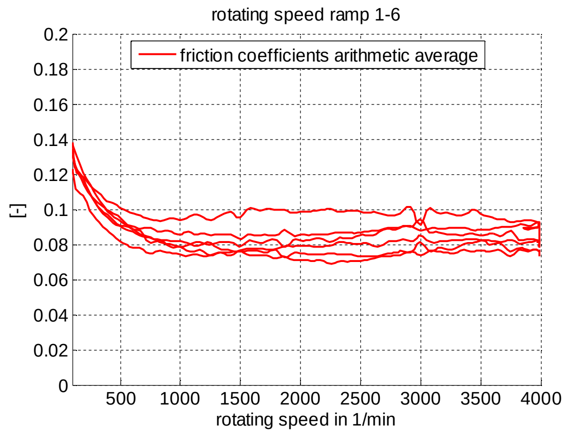


*Figure 12. Arithmetic averages of the measured friction coefficients at rotating speed ramp 1 to 6 with changed test components at the friction force measurement mounts.*

## CONCLUSIONS

A test bench for the analysis of the tribological behavior of the contact between cylinder and control plate in the axial piston variable pump was developed. The main objective of the developed test bench is to test the tribological behavior of lightweight materials in this specific contact. The test bench aims at modelling tribological conditions observed in the real system.

Measurements done for validation of test bench indicated reproducible test results especially for rotating speeds higher than 750rpm. It was also evident that the oil bath temperature could not be hold constant. The reason for this is that the friction heats up the oil bath and test bench structure and the test bench is not equipped with an oil cooling system at this time. For further investigations a cooling system is necessary to guarantee a constant oil bath temperature. Thus, a cooling system for the test bench RPR is planed.

Measurement are done with a pressure load of 350N. Some more measurement results with different pressure loads are necessary for the validation of the test bench.

The function of the developed test bench is verified. There will be test with lightweight materials in the future.

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