The Piston Cylinder Assembly in Piston Machines – a long Journey of Discovery

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Abstract: This paper summarizes the main contributions of researchers and engineers to the discovery of physical phenomena defining the fluid film properties and the operational conditions of the piston cylinder interface in hydrostatic piston machines. The main focus of this paper is the piston cylinder assembly of designs, where the design principal is based on torque generation requiring a large side load of the piston. Since 1965 more than 20 dissertations have been completed on theoretical and/or experimental studies of the piston cylinder interface. Listing all of the papers published worldwide on analysis of piston kinematics and dynamics, modelling and simulation of piston/cylinder interface and experimental studies would exceed the allowable length of this paper. Therefore only major milestones in discovery will be discussed.

Keywords: hydrostatic piston machines, fluid film properties, theoretical and experimental studies, lubrication gap, swash plate, piston kinematics, forces, test rig, numerical model, measurement, piston design

1 Introduction

Among all types of positive displacement machines piston machines can be designed for very high operating pressures (above 40 MPa). Piston machines have the potential to achieve very high efficiencies and allow for variable displacement units. These three attributes form a strong basis for component selection for highly efficient and compact fluid power drive systems. Since Ramelli's first axial piston pump designed at the beginning of the 17th century many generations of piston machines have been invented, designed and produced. The designs have been continuously simplified over the last decades. Most of the current manufactured piston pumps and motors have much lower number of parts than machines designed 50 years ago. A successful designed piston machine requires elegant solutions for many parts, but especially for the highly loaded sealing and bearing interfaces. Most piston machines have at least three different types of sealing and bearing interfaces as shown in *Figure 1* exemplarily for swash plate type axial piston and radial piston machine with external piston support. The piston cylinder interface is the most complicated interface in terms of balancing high pressure forces. The piston cannot be hydrostatically balanced like the slipper/swash plate or slipper/outer ring and the cylinder block/valve plate interface, which can be designed as combined hydrostatic/hydrodynamic bearing. The successful design of the piston cylinder interface determines the achievable maximum operating pressure, maximum speed and maximum swash plate angle. A very challenging design goal is to avoid wear by creating a sufficient load carrying ability of the fluid film and to minimize energy dissipation in a wide range of operating conditions in order to achieve a very high efficiency. Many researchers have studied the behaviour of slippers in axial and radial piston machines as well as the cylinder block valve plate interface of axial piston machines in order to find optimal design solutions. However the amount of research conducted on the piston cylinder interface is much larger. The piston cylinder interface represents a very complex hydrodynamic bearing, which has to perform simultaneously as a sealing element to seal the displacement chamber from case pressure. This double function is especially difficult to solve in those piston machines, where the moment generation leads to a high side force acting on the piston. Consequently the majority of past research related to piston cylinder studies focus on designs involving high side load of the piston. The situation is different in bent axis ma-

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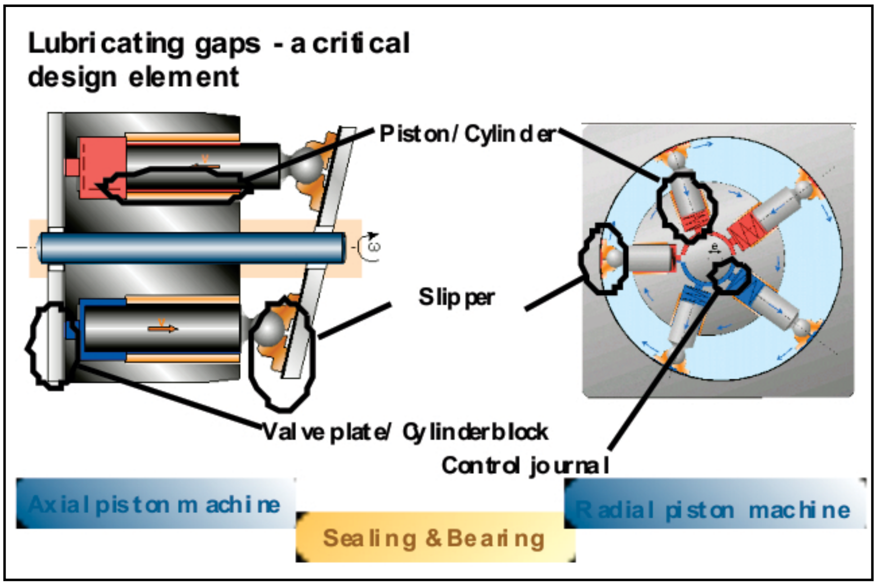

Figure 1. *Lubricating gaps in piston machines*

chines and radial piston machines utilizing special mechanism to minimize the piston side load. Piston machines with low side force can utilize piston rings like combustion engines. Piston rings allow for a much better sealing of the displacement chamber and also help to reduce piston friction. Much research has been conducted on the piston cylinder interface for combustion engines. Many of those research results have been translated and utilized in hydrostatic piston machines like bent axis machines, which can use piston rings. This might explain that there has been less research conducted on piston/cylinder assembly for bent axis machines. In this paper only work related to piston machines with high side forces will be reported.

1 The piston motion

As it will be explained in more detail in this chapter the piston conducts a very complex periodical motion, which can be divided in two parts: the piston macro and the piston micro motion. The piston macro motion can be derived analyzing the piston kinematics.

1.1 Kinematics of the piston

The kinematics of the piston can be determined from the basic mechanism describing the rotating group of each design. A comprehensive overview can be found in Ivantysyn and Ivantysynova (1993). *Figure 2* shows the basic kinematic relationship for swash plate type machines.

The piston stroke s_K is dependent on pitch radius R, swash plate angle β and angular position of the block φ.

$$s_K = -R \cdot \tan\beta\left(1-\cos\varphi\right) \qquad (1)$$

The piston stroke H_K, which represents the displacement of the piston from outer dead center (ODC) to inner dead center (IDC) yields:

$$H_K = 2 \cdot R \cdot \tan\beta \qquad (2)$$

Based on the maximum piston stroke, the piston velocity v_K and the piston acceleration a_K can be derived as follows:

$$v_K = \frac{ds_K}{d\varphi}\omega = -\frac{1}{2}\omega \cdot H_K \cdot \sin\varphi \qquad (3)$$

$$a_K = \frac{dv_K}{d\varphi}\omega = -\frac{1}{2}\omega^2 \cdot H_K \cdot \cos\varphi \qquad (4)$$

Therefore, for a given pump or motor design, the piston sliding velocity and acceleration are a function of the angular position and angular velocity of the shaft.

Due to the rotation of the shaft and the cylinder block or in case of inverse kinematics due to the rotation of the swash plate (wobble plate design), the piston has to move along the piston axis to accomplish the piston stroke. This axial piston motion represents a forced motion. Besides that the piston has the freedom to turn about its own axis. The piston turning/spinning motion is not forced and therefore depends on the friction between cylinder and piston and between piston and slipper. Several researchers have studied the piston relative motion (Renius 1974, Regenbogen 1978, Hooke and Kakoullis 1981, Harris et al 1993, Fang and Shirakashi 1995, Wieczorek and Ivantysynova 2002 and Lasaar 2003). Renius (1974) accomplished a very comprehensive experimental study and confirmed the piston spin motion in swash plate type axial piston machines through measurements of piston friction forces in axial and circumferential direction. Hooke and Kakoullis (1981) studied experimentally the dependence of piston spin motion on pump speed

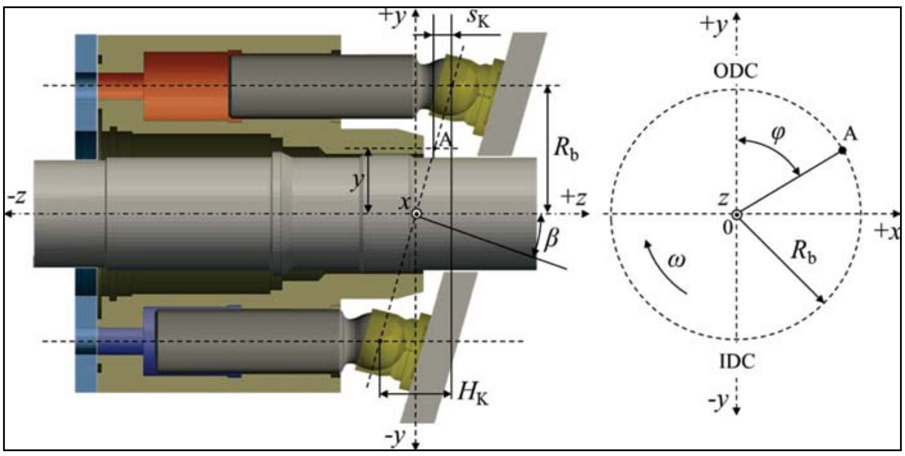

Figure 2. *Swash plate type axial piston pump kinematics*

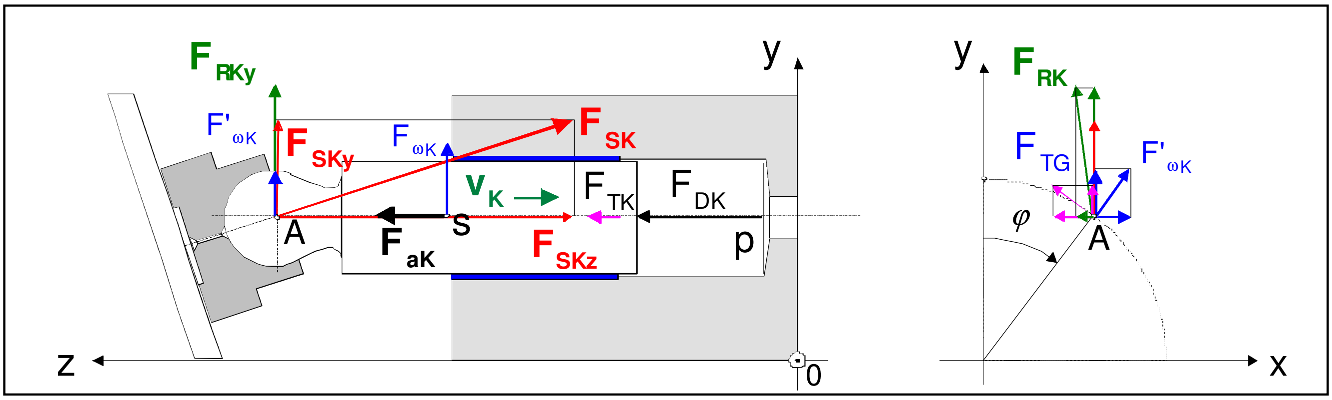

Figure 3. *Piston free body diagram*

and operating pressure and concluded that increasing operating pressure supports the piston spin motion due to high piston forces pressing the piston into the slipper ball joint and preventing any relative motion between piston and slipper. Ivantysynova (1983) considered the piston relative motion in her numerical model. She compared calculated pressure fields with measured pressure profiles, which explained that the piston spin motion contributes to hydrodynamic pressure built up. Fang and Shirakashi (1995) measured the piston spin motion on a modified wobble plate pump and explained that the spin motion contributes to the load carrying ability of the piston and therefore helps to prevent mixed friction. Also Lasaar (2003) confirmed with his piston friction force measurements using a specially designed swash plate type machine that the piston spin motion is present and its value depends on operating conditions.

In addition to the described axial motion and rotation about its axis the piston is conducting a complex micro motion within the cylinder bore. The micro motion can be derived from the force balance of the piston.

1.2 Forces acting on the piston

Forces acting on the piston have been studied by many authors (Brangs 1965, Yamaguchi 1976, van der Kolk 1972, Dowd and Barwell 1974, Renius (1974), Hooke and Kakoullis 1981, Ivantysynova 1983, Harris, Edge and Tilley 1993, Ivantysyn and Ivantysynova 1993, Fang and Shirakashi 1995, Kleist 1995, Jang 1997, Manring 1999, Sanchen 1999, Olems 2001, Wieczorek 2002, Scharf and Murrenhoff 2005). *Figure 3* shows a free body diagram of the piston. The figure shows all external forces acting on the piston. When the pump or motor runs under pressure the largest force is usually the pressure force F_{DK}.

The pressure force is determined by the instantaneous cylinder pressure and therefore changes periodically over one shaft revolution. Similar the piston inertia force F_{aK} changes periodically with the piston acceleration. Dependent on the fluid film conditions and the resulting flow between piston and cylinder the piston friction force F_{TK} will also act on the piston. Finally the reaction force of the swash plate F_{SK} and the piston centrifugal force $F_{\omega K}$ will act on the piston. As shown in *Figure 3* the component F_{SKy} of swash plate reaction force represents a radial force acting on the piston. In addition the y-components of the piston centrifugal force and the slipper friction force F_{TG} will contribute to the resulting side load of the piston. The resultant force F_{RK}, which acts on the piston perpendicular to the piston axis at point A of the ball joint, can be determined by vectorial addition of the forces F_{SKy}, $F'_{\omega K}$ and F_{TG}:

$$F_{RK} = \sqrt{(F_{SKy} + F'_{\omega Ky} + F_{TGy})^2 + (F'_{\omega Kx} + F_{TGy})^2} \quad (5)$$

Under the assumption of full fluid film lubrication and absence of metal to metal contact, the force F_{RK} must be compensated by the fluid film between piston and cylinder, i.e. resulting pressure force and moment generated from the fluid pressure field. In other words, the hydrodynamic pressures built up in the fluid film must develop fluid forces and moments sufficient to balance the external forces. To illustrate the amount of force to be carried by the fluid film lets take a swash plate pump with a 20 mm piston diameter and a swash plate angle of 21 degree. The side force F_{RK} acting on a single piston is approximately 5000 N when the pump runs at 45 MPa pressure.

Because of the nature of the basic operation of piston machines, most of the external forces change periodically with the rotating angle of the shaft. The periodically oscillating external forces lead to a complex micro motion of the piston, where the piston changes its inclination angle in the cylinder bore till the resulting hydrodynamic pressure field generates the required resulting fluid force and moments to balance the external forces. Fang and Shirakashi (1995) first time proposed a

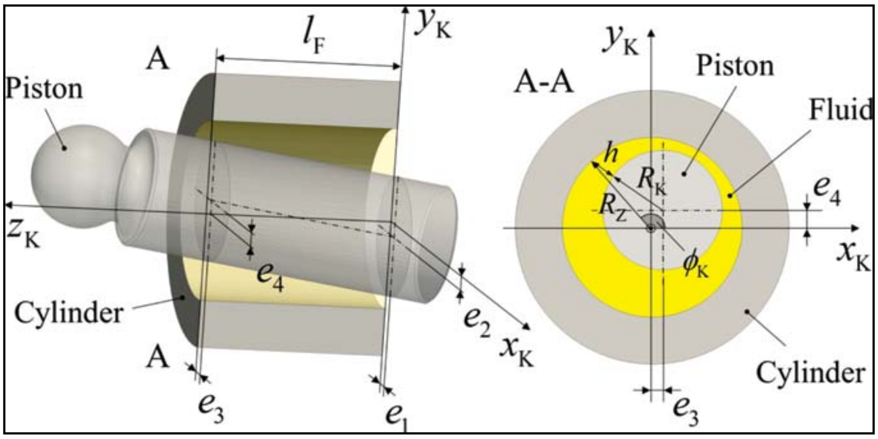

Figure 4. *Inclined position of the piston in the cylinder*

method to calculate the changing inclined positions of the piston during one shaft revolution that would balance the external piston forces. Kleist (1995) developed a method to calculate the piston micro motion in order to balance external forces. Similar models were used by Sanchen (1999), Olems (2001) and Wieczorek (2002). *Figure 4* shows the resulting inclined piston position of the piston in the cylinder bore at a given angular position of the shaft.

2 Experimental work

Many researchers world wide built special test rigs using different simplified versions of the piston cylinder assembly or modified pumps (van der Kolk 1972, Yamaguchi 1976, Dowd and Barwell 1975, Renius 1974, Regenbogen 1978, Hooke and Kakoullis 1981, Ivantysynova 1983, Koehler 1984, Ezato and Ikeya 1986, Fang and Shirakashi 1995, Donders 1998, Manring 1999, Sanchen 1999, Tanaka 1999, Olems 2001, Ivantysynova and Lasaar 2000, Oberem 2002, Ivantysynova, Huang, and Behr 2005) to study the piston cylinder interface. Most of the test rigs were based on a single piston design and inverse kinematics. One of the first piston cylinder test rigs was built by van der Kolk (1972). The design did not allow for any axial piston motion. A side force was applied from outside. The arrangement simplified the conditions to a tilted journal bearing with increased outside pressure. The test rig built by Renius (1974) was also based on a single piston assembly using inverse kinematics. The design allowed for a more realistic study of the condition of the piston cylinder interface by accounting for complex piston motion (axial and spin) under varying pressures in the displacement chamber. Renius measured piston friction forces in axial and circumferential direction. His extensive experimental work formed a major milestone in the discovery of physical behavior of the piston cylinder interface. The test rig used by Dowd and Barwell (1975) was proposed to study the impact of material properties and surface roughness on friction and wear of the piston cylinder interface. As shown in *Figure 5* the test rig was also based on inverse kinematics. In addition to the missing centrifugal force also the impact of transient external side load and piston spin motion on the hydrodynamic pressure built up are neglected in the experimental study conducted by Dowd and Barwell.

Dowd and Barwell (1975) conclude that "the clearance between piston and cylinder is extremely small and the slope of the asperities is too slight to provide sufficient hydrodynamic lift." They also wrote in their conclusion that elastohydrodynamic effects might balance the side force. Yamaguchi (1976) built a single piston test rig to study the piston mo-

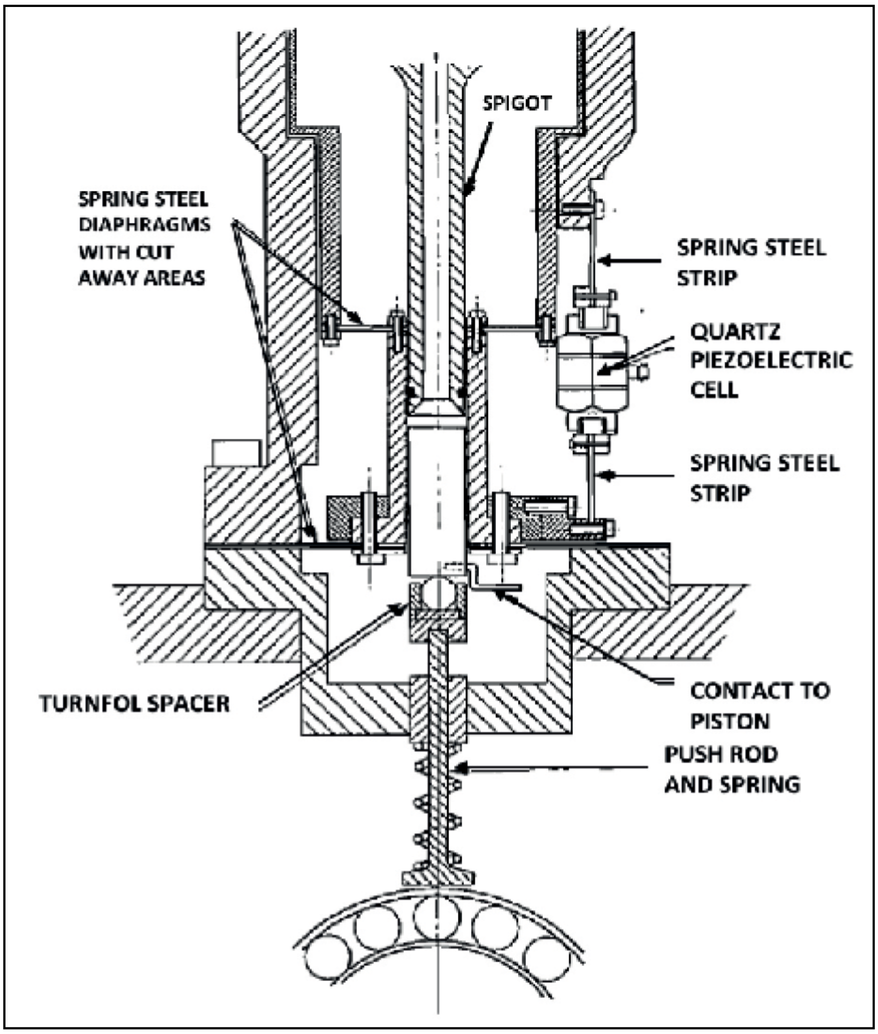

Figure 5. *Test rig used by Dowd and Barwell (1975)*

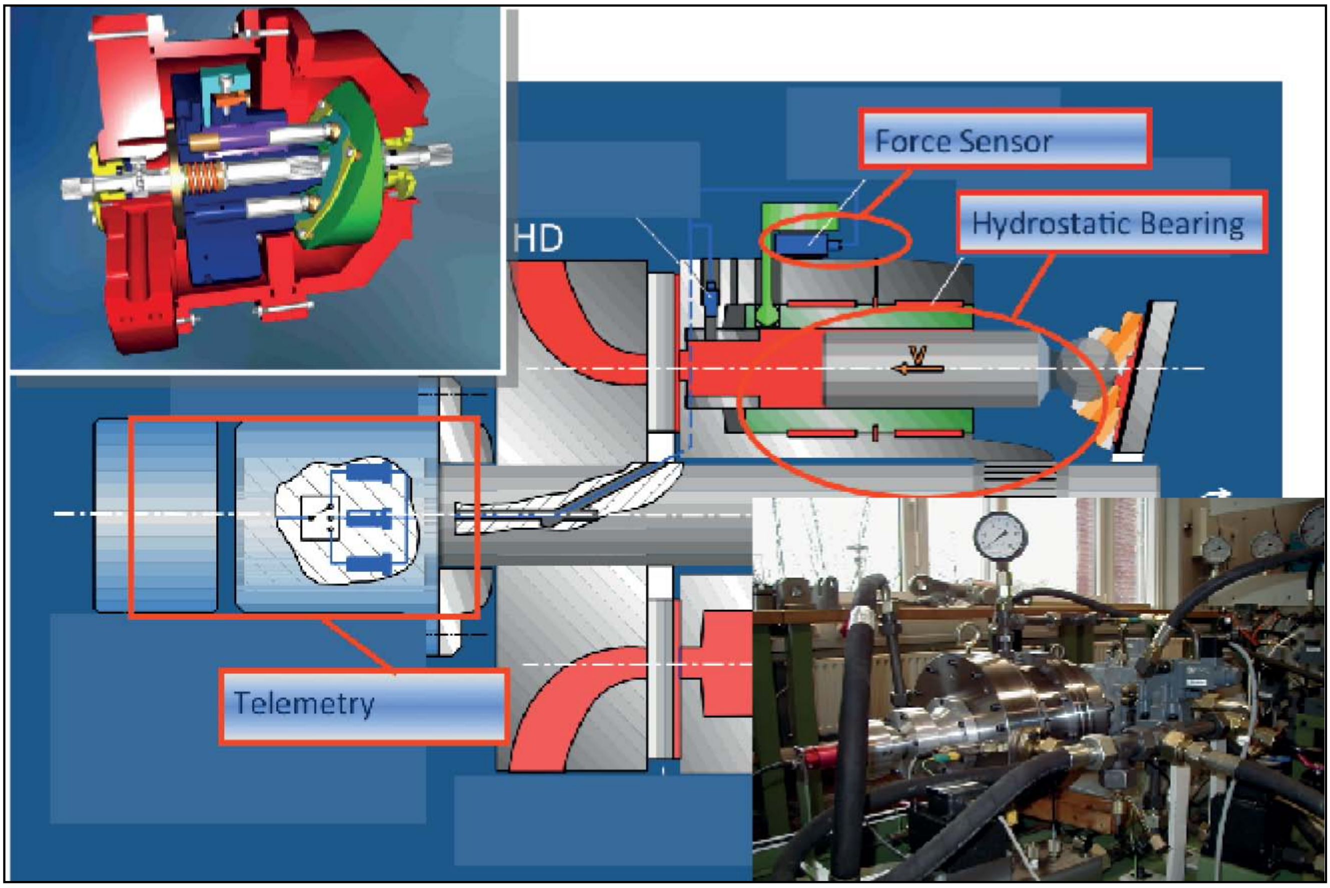

Figure 6. *Cross section of the test pump*

tion and fluid film conditions of piston machines. His test rig simulates the conditions in bent axis machines using a design with connecting rod. He measures gap heights between the piston and cylinder under very low pressures (maximum operating pressure is 5 MPa). Several more authors built test rigs to measure the piston friction force (Ezato and Ikeya 1986, Donders 1998, Manring 1999, Ivantysynova and Lasaar 2000). The test rig proposed and built by Ivantysynova and Lasaar (2000) first time allowed to measure piston friction forces on a swash plate machine under machine conditions comparable to standard designs. The specially designed pump allows measurements of axial and circumferential friction force in pumping and motoring mode in a wide range of operating conditions. *Figure 6* shows a cross section of the test pump using a 3 component piezoelectric force sensor mounted to a hydrostatically beard bushing.

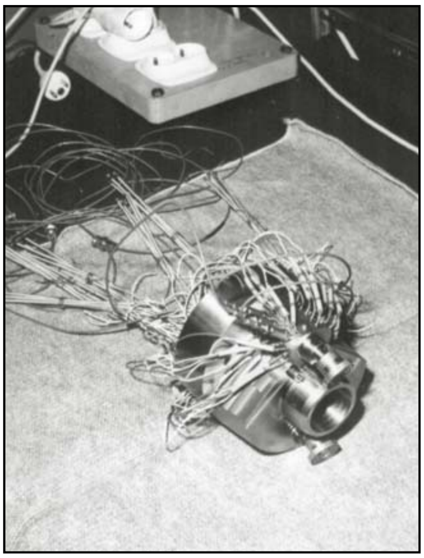

Figure 7. *Test pump with implemented thermocouples and pressure sensors*

The instantaneous cylinder pressure is measured with another piezoelectric pressure sensor. A telemetry is used to transfer the measured data wireless from the rotating cylinder block to the data acquisition system. Results of a study of impact of surface roughness as well as measurements using a half-barrel like piston shape on piston friction force are summarized in Lasaar (2002). Donders (1998) used a special single piston test rig built at the Institute for Fluid Power Drives and Control in Aachen to investigate the piston friction force development under different operating conditions using HFA fluids and mineral oil. The test rig allows direct measurement of the piston friction force using a force sensor mounted to the piston. Also this test rig uses a hydraulic cylinder to apply a piston side force. Oberem (2002) used the same test rig later for a comprehensive study of different piston designs and the influence of clearance on the axial piston friction force. Ivantysynova (1983, 1985) used a modified swash plate type pump with inverse kinematics to measure the temperature field between piston and cylinder. 30 thermocouples were implemented around a single piston. In addition the pressure in the gap between piston a cylinder was measured using 3 piezoelectric pressure sensors. *Figure 7* shows the implemented thermocouples and the pressure sensors.

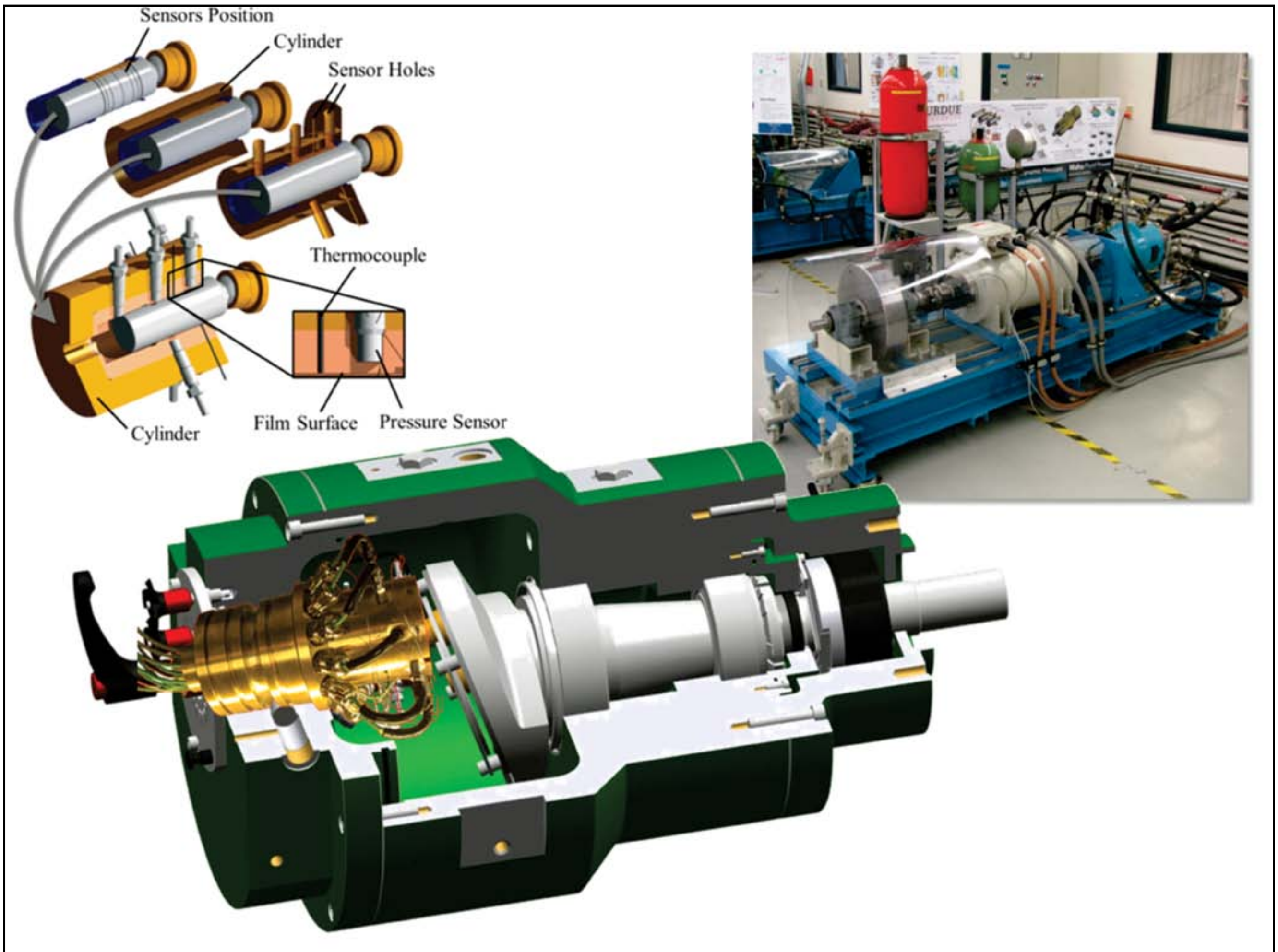

Figure 8. *Test rig for direct measurement of the pressure field in the fluid film between piston and cylinder*

The goal of the measurements conducted in pumping and motoring node was to verify the developed non-isothermal gap flow model. Olems (2001) implemented 65 thermocouples around the cylinder surface and in the cylinder block to measure the surface temperature between piston and cylinder and the heat transfer through the cylinder block into the housing. The test rig shown in *Figure 8* allows first time direct measurement of the pressure field in the fluid film between piston and cylinder.

The idea was experimentally to confirm that elasto-hydrodynamic pressure built up is present in this interface. The test rig uses a single piston and inverse kinematics. A special locking devise allows turning and fixing the block in 180 predefined positions. Therefore dynamic pressure can be measured at a grid of totally 1620 measurement locations around the piston using 9 pressure sensors only (Ivantysynova, Huang and Behr 2005). The test rig also allows measuring the temperature field between piston and cylinder on a fine grid of 1620 grid points using 9 thermocouples and the described locking device.

In summary, the above briefly described experimental work allowed for major steps in the discovery of what effects are involved and how those effects contribute to the resulting piston motion and fluid film behavior. Many of the test rigs were built to support and confirm modeling of the piston cylinder interface. The next chapter will summarize major milestones in the area of modeling and simulation.

3 Modeling of piston cylinder interface

Gerber (1968) presented the first model for the calculation of gap flow and friction between piston and cylinder. Gerber does not consider any hydrodynamic pressure built up and proposes to use different coefficients to calculate the friction between piston and cylinder. Van der Kolk (1972) first time calculated the pressure field between piston and cylinder by solving the Reynolds equation. He neglected the axial piston motion and calculated the pressure field based on the spin motion of the piston and an inclined piston position, i.e. conditions similar to a tilted journal bearing. Due to limitations in the available computing power a very rough computing grid of just 48 elements was used to describe the fluid film flow. Yamaguchi (1976) studied the motion of the piston in piston machines based on the hydrodynamic lubrication theory solving the Reynolds equation. He considered already the impact of change of gap height with respect to time due to changing pressure in the displacement chamber. Based on his calculations

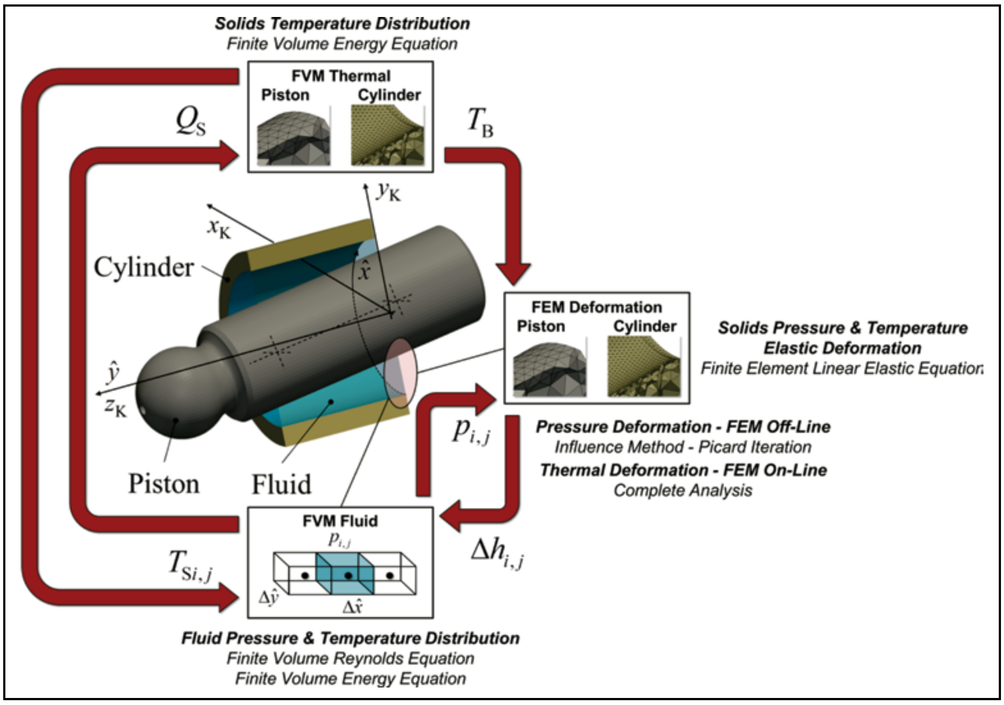

Figure 9. *Piston cylinder coupled physical domains*

he proposes a tapered piston shape to stabilize the piston motion, i.e. to achieve force equilibrium between fluid forces and external forces. Ivantysynova (1983, 1985) presented a non-isothermal model, where in addition to the Reynolds equation the energy equation is solved in order to consider the impact of change of viscosity due to temperature and pressure on gap flow conditions. The presented model solves the pressure and temperature field between piston and cylinder for an assumed inclined position. The resulting fluid forces and moments are compared with external forces, but no additional iteration loop is added to the simulation routine to determine the micro motion of the piston. Fang and Shirakashi (1995) first time proposed a calculation method to determine the piston position considering force balance between external and fluid forces by varying the piston position. In the case that no force balance can be found they assume contact forces and mixed friction. Fang and Shirakashi neglect the influence of squeeze film effect on hydrodynamic pressure built up. Kleist (1995) presented a new approach to calculate the piston position within the cylinder bore based on the balance between external and fluid forces and moments. He considers also the micro motion of the piston and its effect on hydrodynamic pressure built up. Kleist uses a modified Reynolds equation, where the influence of surface roughness is considered using a statistical model, which was originally proposed by Patir and Cheng (1979). The model was later integrated by Sanchen (1999) in the simulation program PUMA. Olems (2002) used a non-isothermal model together with Newton Raphson's approach to solve for the force balance between external and fluid forces and to determine the piston micro motion and resulting additional load carrying ability of the fluid film. Wieczorek (2002) integrated Olem's piston cylinder model into the simulation program CASPAR. CASPAR calculates all three connected lubricating gaps of a swash plate type axial piston machine based on non-isothermal gap flow models and micro-motion of moveable parts.

Figure 10 shows the flow chart describing the implemented numerical algorithms.

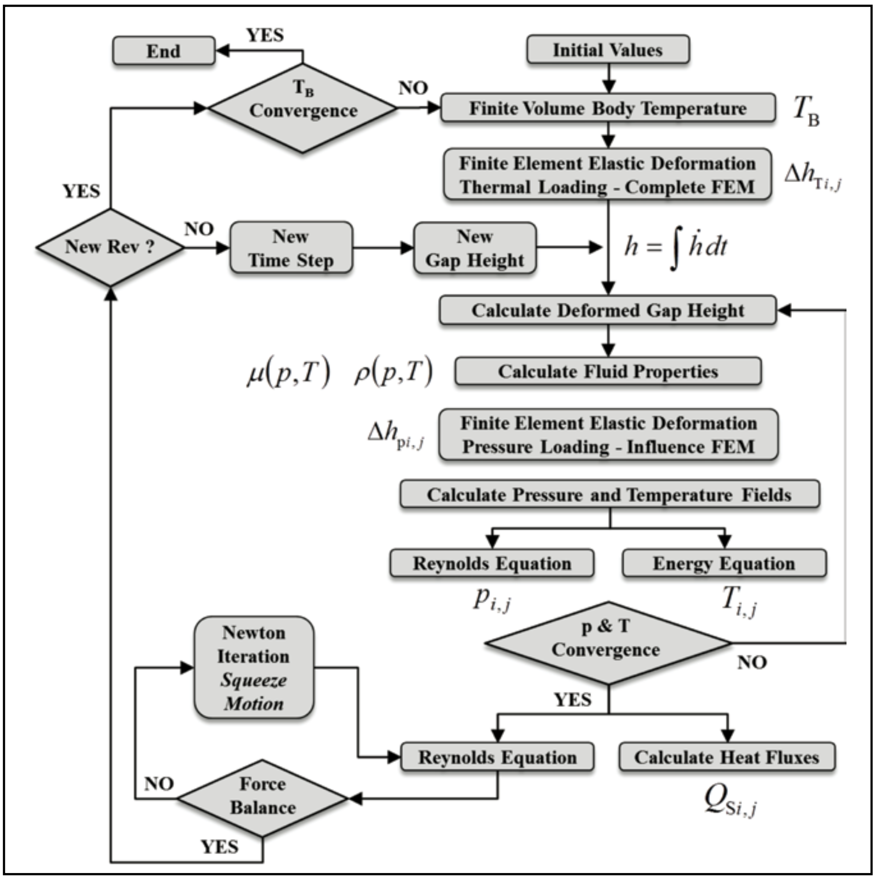

Figure 10. *Flow chart of implemented numerical algorithms*

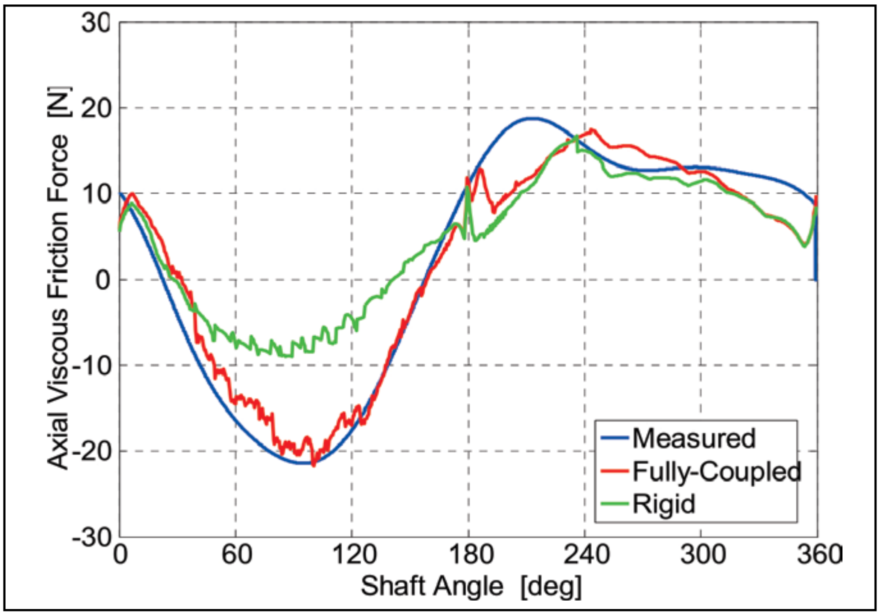

Figure 11. *Comparison of measured and calculated piston friction forces*

The investigation of elastohydrodynamic effects within the piston cylinder assembly was inspired from various research in the field of tribology, where various procedures for solving the complex elastohydrodynamic lubrication problem were reported already in the sixties. For example Dowson and Higginson (1959) described an iterative procedure that solved the elastohydrodynamic problem for line contacts. Knoll (1974) studied the load carrying ability of journal bearings under elastohydrodynamic lubrication. Dowson et al (1982) studied the elastohydrodynamic lubrication of piston rings in the combustion chamber of an internal combustion engine. Also Reiners (1987) studied elastohydrodynamic effects on piston cylinder assembly of combustion engines under the consideration of primary and secondary piston motion.

Ivantysynova and Huang (2002) proposed first time to include an elasto-hydrodynamic model for the simulation of the gap flow between piston and cylinder of a swash plate type axial piston machine. It is assumed that high pressure peaks developed in the fluid film lead to local elastic surface deformation of the piston and cylinder. This will lead to a local change of film thickness and therefore impact the resulting pressure field and all other gap flow parameter including friction. The change of fluid film thickness due to elastic deformation of piston and cylinder was calculated using an influence matrix approach. The influence matrices were calculated using ANSYS. A similar piston cylinder simulation model considering elastohydrodynamic effects was presented by Fatemi et al (2008). This model coupled commercial FEM and multi-body simulation software with an in-house fluid film model. A fully coupled fluid-structure thermal and multi-body dynamics simulation model for the piston cylinder interface was first time published by Pelosi and Ivantysynova (2009). The model does not only consider elastic deformation of the piston and cylinder due to pressure load, but also considers thermal expansion due to energy dissipation in the fluid film. The proposed algorithm includes a complete heat transfer model of the cylinder block and piston assembly. *Figure 9* illustrates the numerical coupling of different physical domains considered in this new piston cylinder model.

The accuracy of gap flow prediction of this new fully coupled fluid structure interaction model has been verified through comparison of measured and calculated piston friction forces, as shown in *Figure 11*. The measurements were made using the tribo test rig developed by Lasaar (2003).

Figure 12 shows a comparison of measured and simulated temperature field between piston and cylinder.

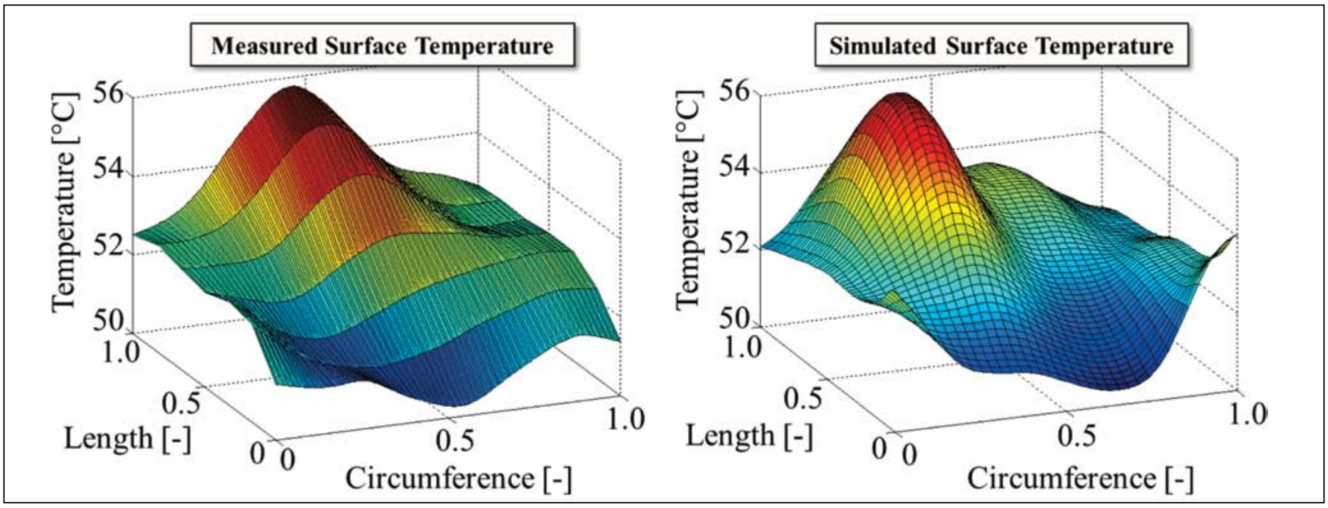

Figure 12. *Comparison of measured and simulated temperature field between piston and cylinder at* $\Delta p = 175$ bar, $n = 1500$, $T_{case} = 51.8$ °C, $T_{HP} = 45.8$ °C, *and* $T_{LP} = 43.1$ °C

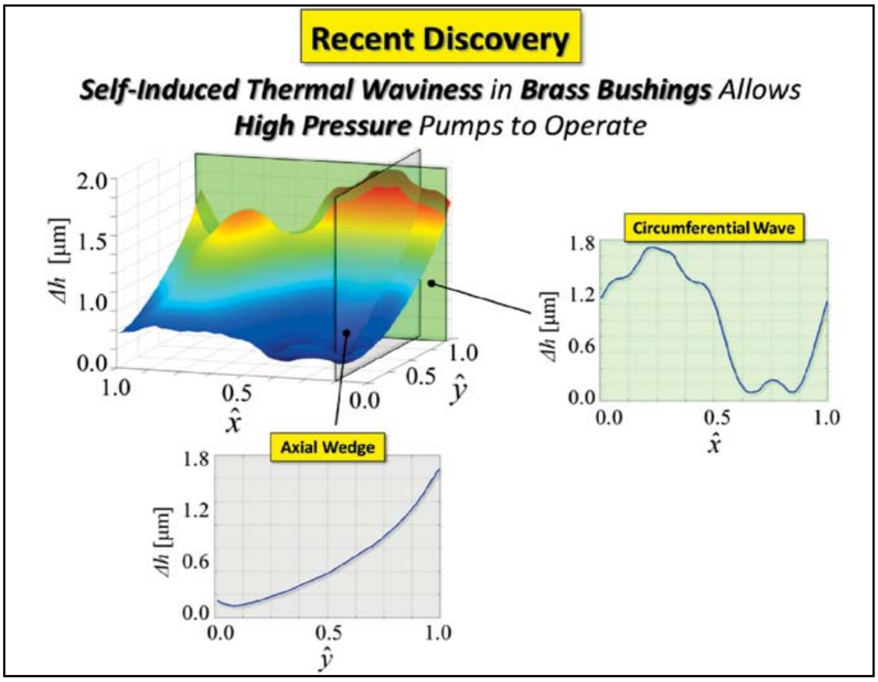

Figure 13. *Change of resulting gap height*

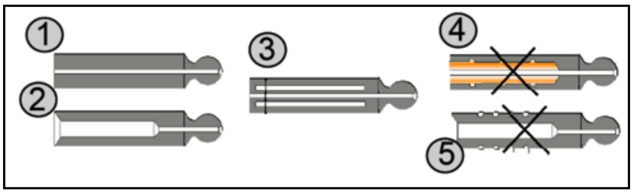

Figure 14. *Current standard piston designs*

The measurements were made using the test pump shown in Figure 8. More results have been published in Pelosi and Ivantysynova (2011). The new model was further used to investigate the deformation of piston and cylinder due to pressure and temperature for a standard swash plate unit with brass bushing pressed into the cylinder block made from steel. It was discovered that the resulting elastic deformation of the brass bushing due to thermal ad pressure load leads to a waviness of the surface. Simulation results show surface deformations due to thermal loading and pressure of the piston and bushing for a 75 cc swash plate type pump running at 35 MPa differential pressure, 1500 rpm and maximum swash plate angle. Port and case temperatures were taken from steady state measurements of the same pump, i.e. inlet temperature 49.7°C, temperature at high pressure port 54.6°C and case temperature 65°C. As shown in *Figure 13* the elastic deformation of the piston and bushing lead to change of resulting gap height in order of several microns.

More important is the generation of waviness in circumferential direction of the bushing surface and the generation of a wedge in axial direction of the gap between piston and cylinder. Both the circumferential waviness and the wedge effect will contribute to additional pressure built up and therefore increased load-carrying ability of the fluid film. This discovery is important for the understanding of physical phenomena contributing to fluid film conditions. It will also allow further optimization of the piston cylinder assembly to further reduce energy dissipation and increase power density of the machines.

■ 4 Design studies of the piston- cylinder assembly

Dowd and Barwell wrote in their paper published in ASLE Transactions in 1974:" *Conventional high--pressure hydraulic pumps require a delicate balance between clearance, surface finish, and pump materials if a long, efficient life is to be maintained.*" Thanks to the research efforts of so many researchers over the last 40 years we now know a little more about the important contribution of different physical phenomena to load carrying ability of the fluid film, but still need to continue to study the sensitivity of the different design options to the resulting fluid film behavior. In this chapter the research activities related to study the impact of piston and cylinder shape as well as material options will be discussed. *Figure 14* shows several standard piston designs, which have been and are still used by many manufactures worldwide.

Whereas piston # 1 is too heavy and will minimize pump speed, piston 2 introduces too high compression loss due to high dead volume. Dependent on the material used insi-

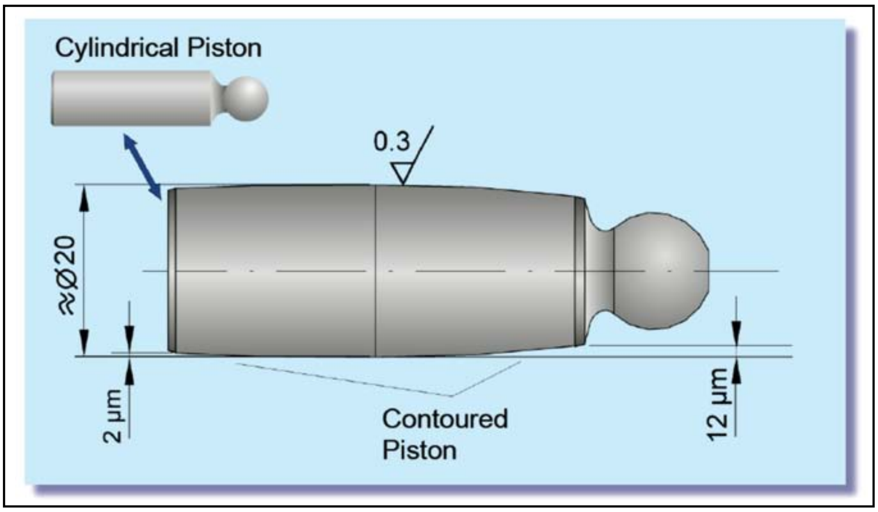

Figure 15. *Piston shape proposed by Lasaar and Ivantysynova*

de the hollow piston piston #4, the piston will have similar problems like piston # 1 or #2. The ring grooves on the outer surface of piston # 5 are clearly destroying the hydrodynamic pressure built up in the groove area and are therefore not helping to increase load carrying ability of the piston, i.e. such design should not be used at all. The best design considering a cylindrical piston shape represents piston # 3. However when studying the governing equation describing the fluid film behavior, it is obvious that shaping of the piston and or cylinder is a very helpful design element to increase load carrying ability of the piston cylinder interface, reduce energy dissipation and to avoid wear and mixed lubrication. Yamaguchi (1976) first time proposed a tapered piston shape to utilize an improved hydrodynamic pressure built up with axial piston motion. Ivantysynova (1983) proposed a barrel like piston with a large curvature radius leading to a diameter reduction of 4 micrometer at both piston ends. The idea of optimizing the piston shape to minimize energy dissipation within the piston cylinder interface was continued by Lasaar (2003), who proposed a half barrel like piston as shown in *Figure 15*.

Kleist (1997) proposed and tested different shaped pistons for radial piston machines. The shaped piston showed lower friction and comparable leakage than the cylindrical piston. Murrenhoff et al (2010) presented several research works on shaping of the piston and bushing together with piston and bushing coatings as well as first investigations into surface texturing. The paper reports reduction of piston friction force for coated and shaped pistons, however the presented measurement results show improvements only for the low pressure/suction stroke. These results are very different from the results Lasaar obtained for his shaped piston, where reduction of axial friction force was obtained especially during the high-pressure stroke. The newest version of the fully coupled

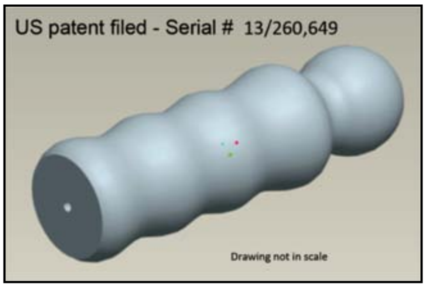

Figure 16. *Waved Piston*

fluid structure interaction model of the piston cylinder interface developed by Pelosi and Ivantysynova (2011) was used to conduct a simulation study for waved pistons. A US patent application has been filed for the waved piston shown in *Figure 16*. Recent simulation results have demonstrated a huge potential to achieve major reduction of energy dissipation for the piston cylinder interface. *Table 1* shows a comparison of average reduction of power loss achieved for all eight simulated operating conditions for three different waved piston designs. The simulations were made for pump operation at high and low speed (1000 and 3000 rpm), high and low pressure (400bar and 100 bar) and 100% and 20% swash plate angle. The waved design with 2.5 sin waves and an amplitude of +/- 6 micrometer showed reduction of power loss for all operating conditions with the highest of 80% at simulated operating point 4 (3000 rpm, 400 bar and 20% swash plate angle). These very recently obtained research results have not been published elsewhere. Besides shaping and surface structuring several researchers have investigated the use of new materials and coatings for the piston and cylinder. Feldmann and Bartelt (2003) reported results of testing of ceramic pistons and bushings in a swash plate type axial piston machine. After 600h testing at maximum pressure of 320 bar and speed of 2000 rpm piston and bushing showed no wear , but the overall efficiency of the unit was lower than the standard one. Another unit equipped with ceramic pistons failed after short operation. Scharf and Murrenhoff (2004) concluded after they conducted a comprehensive study on different physical vapor deposition (pvd) coatings of different pump parts that the piston reveals the weak point of Zirconium Carbid coating because the piston cylinder interface represents the most complex interface within the displacement unit.

The main idea of most of the previous research on new material combinations and coatings for the piston cylinder assembly was to prevent wear and reduce friction. The newest finding of the behavior of the brass bushing pressed in to the steel cylinder block reported in chapter 3, reveal that the piston and cylinder material needs to support the load carrying ability of the fluid film. This is the only successful way of increasing load carrying ability of the fluid film, which will result in reduction of energy dissipation and friction and with that also avoid wear. Dowd and Barwell (1974) concluded in their paper that the side load of the piston must be balanced by hydrodynamic lift generated by microaspirities and elasto-hydrodynamic contact between the two surfaces. Our newest findings confirm that elasto-hydrodynamic effects play a major role and help current designs to run successful. This discovery however opens the door to interesting new research in investigating new materials, coatings and shapes for this heavy loaded

Table 1. *Average reduction in power loss due to piston shaping*

Design	Clearance [μm]	% Reduction in Power Loss Leakage	% Reduction in Power Loss Friction	% Reduction in Total Power Loss
6μm-2 peaks	34μm	8.65%	8.3%	8.35%
6μm-2 peaks	18μm	80%	61%	68%
6μm-2.5 peaks	18μm	78.8%	56.8%	64.7%

interface.

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Bat v izvrtini znotraj hidrostatičnih batnih pogonov – dolga pot odkrivanja

Razširjeni povzetek

Prispevek prikazuje pomembnejše rezultate raziskovalcev in inženirjev pri odkritjih fizikalnih zakonitosti, povezanih z lastnostmi filma kapljevine in delovnih pogojev bata v izvrtini znotraj hidrostatičnih batnih pogonov. Osredotočen je na konstrukcijsko izvedbo bata v izvrtini za primer ustvarjanja potrebnega momenta za premagovanje večjih bočnih sil na bat. Od leta 1965 je bilo izvedenih preko 20 različnih doktorskih disertacij, ki so obravnavale teoretične in/ali eksperimentalne študije mejnih kontaktnih ploskev med batom in izvrtino znotraj hidrostatičnih batnih pogonov. Prispevek prikazuje le glavne mejnike v razvoju hidravlično-tribološkega para, bata v izvrtini.

Slika 1 prikazuje mazalne reže znotraj aksialnih in radialnih batnih črpalk. *Slika 2* prikazuje osnovne kinematične razmere za primer aksialne batne črpalke z nagibno ploščo. *Slika 3* prikazuje diagram sil na bat v izvrtini znotraj aksialne batne črpalke. *Slika 4* prikazuje nagnjen položaj bata v izvrtini pri določenem položaju gredi. Na *sliki 5* je prikazano eno prvih tovrstnih preizkuševališč, namenjeno raziskavam vplivov različnih materialov in površinskih hrapavosti na trenje in obrabo bata v izvrtini. Preizkuševališče sta leta 1975 razvila Dowd in Barvell. Prerez testirane črpalke s tremi piezomerilniki sile je prikazan na *sliki 6*. Testirana črpalka z integriranimi termočleni in tlačnimi zaznavali je prikazana na *sliki 7*. Na *sliki 8* je preizkuševališče za neposredno merjenje tlačnih polj in filma kapljevine med batom in izvrtino. Numerično združevanje različnih fizikalnih domen za nov numerični model sestava bata v izvrtini je prikazano na *sliki 9*. Blokovni diagram izboljšanega numeričnega algoritma za preračun razmer med batom in izvrtino kaže *slika 10*. *Slika 11* prikazuje primerjavo sile trenja bata v izvrtini med rezultati meritev in numeričnega izračuna. Na *sliki 12* je prikazana primerjava med izmerjenimi in simuliranimi temperaturnimi polji med batom in izvrtino aksialne batne črpalke. Elastična deformacija bata in puše vpliva na spremembo višine reže med njima (*slika 13*). Na *sliki 14* so prikazane obstoječe oblike standardnih batov. *Slika 15* prikazuje novo, za 4 mikrometre po premeru, sodčkasto obliko bata. *Slika 16* prikazuje patentiran bat z valovito površino. *Preglednica 1* prikazuje nekaj vplivov na povprečno zmanjšanje izgub za osem različnih simuliranih delovnih pogojev in za tri različno valovite površina bata. Simulacije so bile izvedene za črpalko pri visoki in nizki vrtilni hitrosti (3000 vrt./min. in 1000 vrt./min.), pri visokem in nizkem delovnem tlaku (400 bar in 100 bar) in pri 100-odstotnem ter 20-odstotnem kotu nagibne plošče. Bat s površino, ki je imela 2,5 sinusnih valov, z amplitudo +/–6 mikrometrov, je pokazal največje zmanjšanje izgub (80 % zmanjšanje pri 3000 vrt./min, 400 bar in pri 20-odstotnem kotu nagibne plošče).

Poleg naštetih raziskav vpliva različnih oblikovnih in površinskih izvedb drsne površine bata so se nekateri raziskovalci osredotočili tudi na materiale in prevleke batov in izvrtin. Glavni namen omenjenih raziskav, povezanih z materiali in prevlekami, je zmanjšanje trenja in obrabe bata v puši aksialne batne črpalke. Rezultati omenjenih raziskav odpirajo vrata raziskavam novih materialov, prevlek in oblik drsnih površin težko obremenjenih mejnih ploskev batnih črpalk in motorjev.

Ključne besede: hidrostatični batni pogoni, lastnosti mazalnega filma, teoretične in eksperimentalne študije, mazalna reža, nagibna plošča, kinematika bata, sile, preizkuševališče, numerični model, meritve, oblika bata