The Impact of Micro-Surface Shaping of the Piston on the Piston/Cylinder Interface of an Axial Piston Machine

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Abstract
Axial piston machines of the swashplate type are commonly used in various hydraulic systems and with recent developments in displacement control, it is essential to maximize their efficiency further reducing operation costs as well as improving performance and reliability. This paper reports findings of a research study conducted for the piston-cylinder interface utilizing a novel fluid structure thermal interaction model considering solid body deformation due to thermal and pressure effects in order to accurately predict the transient fluid film within the gap. A large reduction in energy dissipation is possible due to reduced clearances allowable due to the surface shaping of the piston resulting in a reduction in leakage. From this study, it is shown that surface shaping of the piston in combination with a reduced clearance is not only beneficial by improving the efficiency of a machine, but also increases the reliability and the performance of the machine as the load support is enhanced.

KEYWORDS: Axial piston machine, surface shaping, piston-cylinder interface

1. Introduction
Axial piston machines of the swashplate type are widely used in industrial applications due to the ability to operate at high pressures and variable displacements while maintaining favorable efficiencies and reasonable operating costs. Effective and efficient operation of the unit strongly depends on the three main lubrication interfaces as shown in Figure 1. More specifically, the piston-cylinder interface is a key design element of such operation. By adding a micro-surface shape on the piston surface allowing for a decrease in the clearance between the piston and cylinder, the leakages can be greatly reduced while manipulating the fluid film to build up sufficient load support. In other words, the addition of a surface profile could impact the interface in such a way that the energy dissipation is reduced while maintaining or even improving the overall machine operation.
1.1. Piston-Cylinder Interface

In order to better understand the design challenges that arise when designing the piston-cylinder interface to provide an adequate bearing function, minimizing areas of minimum fluid film thickness resulting in either contact or increased viscous friction, while also a sufficient sealing function, minimizing leakages, the forces interacting with the interface, as shown in Figure 2, must be accurately defined under normal machine operation. One of the major forces being the pressure force, $F_{DK}$, acting on the bottom area of the piston due to the displacement chamber pressure, $p_{DC}$. Also acting in the axial direction is the inertial force, $F_{aK}$, due to the acceleration of the piston and the viscous friction forces, $F_{TK}$, due to the viscosity of the fluid. The sum of the axial forces are reciprocated by a reaction force from the swashplate, $F_{SK}$.

The reaction force from the swashplate results in a remaining side force, $F_{SKy}$. Further aiding in the side loading of the piston includes the centrifugal force, $F_{ωK}$, due to the rotation of the mass of the piston-slipper body and the viscous friction force from the slipper-swashplate interface, $F_{TG}$. This side load must be balanced by a hydrodynamic load generated through the fluid pressure of the lubricating gap in order to prevent contact resulting in wear or failure. Sufficient fluid support is generated through a squeeze effect from the motion of the piston, the deformations of the solid bodies due to thermal and pressure loading, and the self-adjustment of the location of the piston within the cylinder bore. This phenomena is strongly dependent on the dynamically changing fluid film geometry as defined in /1/.

![Figure 1: Axial piston machine of the swashplate type](image1)

![Figure 2: Force acting on the piston-cylinder interface](image2)
2. State of the Art

Previous research has been conducted in regards to surface shaping on the piston-cylinder interface. Through various experimental and analytical studies, it was realized that by altering the shape of the piston, the efficiency of the machine could be improved as well as the overall performance. In 1976, Yamaguchi /2/ proposed a tapered piston followed by Ivantysynova /3/ in 1983 in which a barrel like piston was proposed. Further analytical analysis and experimental research was performed by Lasaar and Ivantysynova /4/ on a barrel like piston resulting in a 20% energy dissipation reduction and a 50% reduction in leakages. To further support this study, measurements of such a piston were performed on a specialized test rig /5/ confirming the reduced friction forces. More recently in 2010, Gels and Murrenhoff /6/ studied a contoured piston in combination with a varying gap width and guide length using a simplified modeling approach also demonstrating a considerable reduction in losses.

3. Fluid Structure Thermal Interaction Model

Figure 3 details the fluid structure thermal interaction model utilized for the following research study as proposed by Pelosi /7/. This model solves for the motion of the piston within the cylinder bore based on a balance of external and fluid forces considering the thermal and elastohydrodynamic effects.

![Figure 3: Piston-cylinder fluid structure thermal interaction model](image)

The physical behavior of the piston-cylinder interface is captured through a non-isothermal fluid film flow governed by the Reynolds and energy equations that are simultaneously solved through a finite volume method in module 1. From the energy equation, the energy dissipated in the lubricating gap can be calculated. This energy dissipation is then used to predict the fluid temperature which can be used to determine the fluid viscosity as well as the heat flux on the bounding solid parts. The heat transfer
problem is solved utilizing a finite volume method as shown in module 2. Module 3 then solves for the elastic deformations due to pressure and thermal loading via a finite element method. Since these three modules are co-dependent, several iterations are required to reach a converged predicted fluid film thickness.

4. Novel Piston Designs

Various novel piston micro-surface shapes were investigated using the innovative fluid structure thermal interaction model. In order to quantify the improvements of such designs on the piston-cylinder interface a baseline simulation was conducted for comparison. The baseline unit is a nine piston, 75 cc stock swashplate type axial piston machine in which measured wear profiles were taken into account on both the piston, \( PW_N \), normalized to the piston radius, \( R_k \), as \( \frac{PW_N}{R_k} = 0.1 \) and the cylinder bore, \( BW_N \), normalized to the cylinder radius, \( R_z \), as \( \frac{BW_N}{R_z} = 1.21 \).

With a standard minimum relative clearance corresponding to the gap height between the piston and the cylinder, \( h \), of \( MRC = \frac{h}{R_k} = 1.64\% \).

As a first alternative design a micro sinusoidal wave on the axial length of the piston was studied; the design presented in Figure 4A. The normalized design parameters of the geometry of the sinusoidal wave, the amplitude \( A \) and the wave length \( \lambda \) along the length of the piston \( L_{\text{piston length}} \), in which the results are presented in this investigation are defined as:

\[
A_N = \frac{A}{R_k} = 0.29 \quad (1)
\]
\[
\lambda_N = \frac{\lambda}{L_{\text{piston length}}} = 0.4 \quad (2)
\]

A flat sinusoidal wave along the length of the piston was studied as a second alternative design as presented in Figure 4B. This design was proposed as a flat, cylindrical piston with a slight sinusoidal wave peak introduced on both ends of the piston; the concept being to represent a pre-manufactured wear profile. The amplitude studied remained the same as the sinusoidal wave profile, (1).

As a third design a barrel surface profile along the length of the piston was also studied, Figure 4C. The geometry of the barrel defined in this research study is based on the radii at the ends, \( R_1 \) on the DC end and \( R_3 \) on the case end, and the apex, \( R_2 \), as well as the location of the apex, \( L_{\text{Apex}} \), along the length of the piston:
As a fourth design a direct combination of the sinusoidal wave and the barrel, a sinusoidal waved barrel micro-surface profile along the axial length of the piston as shown in Figure 4D has been analyzed. The design parameters designated in this study include the amplitude, (1), and the wave length of the sinusoidal wave, (2), and a fixed apex location, (4). Due to the sinusoidal wave overlaid on the barrel surface profile with a fixed apex location, the radius of the apex slightly increases:

\[
\frac{R_1}{R_2} = 0.9993 \quad \text{and} \quad \frac{R_3}{R_2} = 0.9989
\]  

(5)

Lastly, a sinusoidal wave around the circumference of a cylindrical piston was investigated as demonstrated in Figure 4E. Although the amplitude of this design remained the same to that of the axial design, (1), the number of waves changed in reference to the circumference of the piston, \(C_k\):

\[
\lambda_N = \frac{\lambda}{C_k} = 0.167
\]

(6)

The geometry specifications considered in this investigation were chosen based on a comprehensive design parameter study.

**Figure 4:** Micro-surface shaped piston designs
5. Piston Micro-Surface Shaping and Clearance Study

An investigation of the impact of various piston micro-surface shapes in combination with different clearances between the piston and the cylinder bore was conducted over a range of critical operating points. The reduced clearance between the piston and cylinder bore was possible due to the surface shaping of the piston in which a balance between the sealing and bearing function of the interface must be maintained. This was observed over a range of clearances reduced from the baseline of 1.64‰ to 1.45‰, 1.21‰, 0.96‰, 0.72‰ and 0.48‰.

5.1. Operating Conditions

This study is performed over the corner operating conditions of a baseline, 75 cc stock unit. The 16 corner operating conditions consist of a low differential pressure of 50 bar and high differential pressure of 450 bar at a low speed of 500 rpm and high speed of 3600 rpm for partial displacement ($\beta$) 20% and full displacement in both pumping and motoring mode. For all operating conditions studied, the inlet temperature remained constant at 52°C reflecting an oil viscosity of 20 cSt. Steady state measurements were not available for these conditions of the unit studied and therefore an internal thermal model developed by Shang and Ivantysynova /8/ was used to predict the outlet and case temperatures that were applied as boundary condition inputs for the model.

The authors have published results of similar investigations of various surface profiles utilizing the fluid structure interaction model in which a range of various other, more common operating conditions have been studied and compared to steady state measurements /9/ and /10/.

6. Results

The resulting losses of all nine piston-cylinder interfaces are presented in the following plots as a consequent of piston micro-surface shaping in combination with decreased clearance in which the machine was simulated at the corner operating conditions. For the indicated plots, the operating condition is denoted by a symbol: ● - 500rpm 50bar, ■ - 500rpm 450bar, ◆ - 3600rpm 50bar, ▲ - 3600rpm 450bar. The various surface profiles are denoted by a line/color: sine wave - medium grey dotted line, flat – dark grey solid line, barrel – medium grey solid line, waved barrel – light grey dotted line, cylindrical circumferential sine – light grey solid line, baseline – black single symbol.
6.1. Pumping Mode Results

The losses due to the addition of a surface shape on the piston as well as reducing the clearance are shown in Figure 6 and Figure 9 at the corner operating conditions in pumping mode. Note that for partial displacements, a low speed of 1500 rpm at a high pressure of 350 bar was rather investigated due to convergence issues.

6.1.1. Full Displacement

![Figure 6: Losses for pumping, \(\beta=100\%\); Energy dissipation (left), Leakages (right)](image)

With the addition of a surface profile leading to the possibility of further reduced clearances, the energy dissipation and leakages can be greatly decreased from the baseline (black symbols); especially so for the higher pressure operating conditions. At higher pressures (curves marked with ■ and ▲), the smaller the clearance the better the improvement as the leakages are greatly decreased without largely increasing the torque losses. This trend holds until the fluid film can no longer support the load. However, at the lower pressure operating conditions (curves marked with ● and ♦), the energy dissipation tends to slightly decrease at the larger clearances studied in which the various surface profiles no longer have an effect. In comparison to the various surface profiles, the waved barrel surface profile (light grey dotted line) does best at the further reduced clearances (0.48‰) in which it is able to generate the required load support for the operating conditions shown while further reducing the leakages in comparison to other surface profiles, such as the barrel (medium grey solid line), at intermediate clearances (0.96‰). The flat surface profile (dark grey solid line) and the sinusoidal wave...
profile (medium grey dotted line) are shown to fail at the further reduced clearances at the higher pressure operating conditions in which these designs are not feasible.

![Image of Piston Axial Friction](image1.png)

**Figure 7:** Piston axial friction forces for 3600rpm, 450bar, 100%, pumping mode; Baseline, 1.64‰ (left), Barrel, 0.72‰ (middle), Waved Barrel, 0.72‰ (right)

A comparison between the piston axial friction forces over one revolution at a high pressure (450 bar), and a lower clearance (0.72‰) is made between the baseline, the barrel surface profile, and the waved barrel surface profile in **Figure 7**. It can be seen that the barrel profile fails under these conditions since the friction forces are greatly increased leading to failure in the low pressure stroke. As for the waved barrel, the piston axial friction forces are similar to that of the baseline although the clearance is greatly reduced; the addition of such a wear profile results in manipulation of the fluid film in which the required fluid support is generated.

Multi-plots are shown to better understand the phenomena occurring within the fluid film resulting in the trends shown. Each plot show the film thickness (white contour lines) with the overlaid resulting fluid film pressure (filled contour) for an unwrapped gap between the piston and the cylinder; the x-axis ($\bar{x}$) representing the gap length and the y-axis ($\bar{y}$) the gap circumference as the cylinder block rotates. The plots are shown for two different rotational angles, $\varphi$, measured from outer dead center (ODC).

![Image of Multi-plots](image2.png)

**Figure 8:** Multi-plots for 3600rpm, 450bar, 100%, pumping mode; Baseline, 1.64‰ (left), Barrel, 0.72‰ (middle), Waved Barrel, 0.72‰ (right)
The multi-plots are respectively shown for the higher pressure operating condition (450 bar) for the baseline in comparison to the barrel and the waved barrel at a reduced clearance (0.72‰) in Figure 8. The top row shows the fluid film and pressure build-up during the high pressure stroke (at $\varphi=90^\circ$). The increased friction forces for the barrel profile is a consequence of the larger areas of minimum fluid film thickness on both ends of the gap. As for the waved barrel design, since the friction forces are simply shifted in comparison to the baseline, as shown in Figure 7, the friction forces are larger at this particular rotating angle ($\varphi=90^\circ$) in which larger areas of minimum fluid film are observed. During the low pressure stroke (at $\varphi=270^\circ$) on the bottom row, the barrel surface profile slides across the cylinder bore due to insufficient load support leading to the increased friction forces and eventually failure. The waved barrel tends to tilt under these conditions resulting in some areas of minimum fluid film thickness on each end of the gap producing the slightly larger friction forces shown in the low pressure stroke, but operation is still possible.

6.1.2. Partial Displacement

![Figure 9: Losses for pumping, $\beta=20\%$; Energy dissipation (left), Leakages (right)](image)

The investigation at partial displacement in pumping mode, revealed that since the side load acting on the piston is greatly reduced the resulting torque losses are minor in which the overall energy dissipation strongly relies on the decreased leakages. Also, this means that since the piston does not tilt as much within the bore, the load is able to be supported in compromise to less of a sealing function contributing to slightly increased leakages. Thus, by reducing the clearance at higher pressures (curves marked with ■ and ▲), the leakages can be greatly decreased in turn reducing the overall energy dissipation; the smaller side loads in combination with the smaller forces at lower pressures (curves marked with ● and ♦) has a negligible impact on the energy dissipation.
6.2. Motoring Mode Results

As a result of surface shaping in combination with decreased clearances at the corner operating conditions in motoring mode the losses are shown in Figure 10 and Figure 11. Note that in motoring mode a differential high pressure of 400 bar was considered in order to not exceed the constraints of the maximum continuous oil temperature specified.

6.2.1. Full Displacement

Figure 10: Losses for motoring, $\beta=100\%$; Energy dissipation (left), Leakages (right)

In comparison to pumping mode, the trends of the varying clearance among the various surface profiles studied are similar. It is likewise the tendency that the leakages are greatly decreased at the reduced clearances, especially so at higher pressures (curves marked with ■ and ▲) in which the torque loss are not largely increased resulting in decreased energy dissipation overall in the case that the fluid film is able to support the load. The slight variations in motoring mode is based on the slightly different operating condition as well as the opposite motion of the piston relative to the high and low pressure in the displacement chamber leading to different forces in comparison to pumping mode.

6.2.2. Partial Displacement

Figure 11: Losses for motoring, $\beta=20\%$; Energy dissipation (left), Leakages (right)
Although the trends are again very similar to that as shown in pumping mode for partial displacement operation the surface profiles are more likely to fail with smaller clearances at higher pressures (curves marked with ■ and ▲).

**Figure 12:** Multi-plots 3600rpm, 400bar, 20%, motoring mode; Baseline, 1.64‰ (left), Waved Barrel, 0.72‰ (right)

In **Figure 12** it can be seen that right away in the simulation in the high pressure stroke (φ=10°) that the piston with a surface profile does not build up the fluid support required in which the piston then drags and bounces along the cylinder bore leading to failure.

### 6.3. Conclusion

A comprehensive simulation investigation has been performed for the various novel piston designs presented in combination with decreased clearances at the corner operating conditions in both pumping and motoring mode. It was shown that the addition of a surface profile allowed for a reduction in clearance due to improved load carrying abilities of the interface leading to reduced leakages and an overall reduction in energy dissipation. In general, the reduced clearances are best, especially at higher pressure operating conditions, as long as the load can be supported. More specifically, this study concludes that the waved barrel does best at the further reduced clearances in pumping mode, but since this combination fails at partial displacements in motoring mode, the barrel surface profile at 0.96‰ is the best overall combination for the complete corner operating conditions investigated and presented in this study.

### 7. References


