ANALYSIS OF THE PISTON/CYLINDER PAIR OF AXIAL PISTON PUMP BASED ON VIRTUAL PROTOTYPE

Zhang Junhui  
Mechanical Department  
Zhejiang University  
Hangzhou, Zhejiang, 310058, China  
Email: benzjh@gmail.com

Xu Bing  
Mechanical Department  
Zhejiang University  
Hangzhou, Zhejiang, 310058, China  
Email: bxu@zju.edu.cn

Piston/cylinder pair is one of the three most important friction pairs of axial piston pump. The reciprocation of piston in the cylinder bushing realizes suction and discharge processes. Improvement of the load carrying ability of piston/cylinder pair contributes to the limit pressure improvement of axial piston pump. Meanwhile, the reduction of friction and leakage of piston/cylinder pair can improve the mechanical efficiency and volume efficiency of axial piston pump. Analysis of axial piston pump, as in other complex fluid power and mechanical systems, requires appropriate insight into three multidisciplinary domains, i.e., mechanics, hydraulics and tribology. In order to analyze and optimize the piston/cylinder pair precisely, we establish a virtual prototype of axial piston pump which is simulated in multibody environment.

The first part in this paper introduces the simulation model which can be divided into three sub-models. The first sub-model built by AMESim is hydraulic model which simulates the pressure in piston chamber. The second sub-model is oil film model which is built in Matlab, and the simulation results include pressure distribution of oil film, friction and leakage of piston/cylinder pair. The forces of hydraulic and oil film model are applied to the mechanical model which is established by software ADAMS. The mechanical model can simulate the realistic motion of piston and analyze the stress and strain of piston. And the piston can be chosen as rigid or flexible as needed. The co-simulation of the three sub-models provides plenty of simulation data which will be used in the characteristic analysis of piston/cylinder pair. And software ADAMS is selected as operation interface of the co-simulation model because of its advantage in visualization.

The second part introduces the relevant experiment rigs briefly. Not only the external performance of axial piston pump such as the delivery pressure but also the pressure distribution inside the piston/cylinder pair can be measured through the test rigs. The comparison of simulation and experiment results proves that the virtual prototype can be used in the prediction of characteristic of piston/cylinder pair precisely.

The last part of the paper conducts some simulation researches about piston/cylinder pair based on the virtual prototype of axial piston pump. Through comparisons of maximum value of pressure distribution, mixed friction area, axial friction force and leakage, some conclusions are drawn. Properties of piston/cylinder pair are the key factors in determining limit load pressure of pump and maximum inclined angle of swash plate. And within a certain range, a smaller clearance is beneficial to the improvement the carrying ability of piston/cylinder pair and the efficiency of pump.
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Zhang Bin
Mechanical Department
Zhejiang University
Hangzhou, Zhejiang, 310058, China
Email: bxu@zju.edu.cn

ABSTRACT

The virtual prototype of axial piston pump is discussed in detailed through the application in investigation of piston/cylinder pair. Three sub-models are introduced firstly. The data is transferred between three sub-models through software interface. The liquid-solid coupling and rigid-flexible coupling of piston/cylinder pair model is achieved through the co-simulation model which is called virtual prototype. Then several related test rigs are mentioned. And the comparisons of simulation results and experimental results demonstrate that the virtual prototype of axial piston pump has a great potential in axial piston pump design. At last, the influence of load pressure, inclined angle of swash plate and clearance between piston and bushing on properties of piston/cylinder pair is analyzed. The simulation results indicate that the design of piston/cylinder pair is the key factor in determining the limit pressure and maximum swash plate angle, and reduction of clearance between piston and cylinder contributes to reduction of leakage and friction force of piston/cylinder pair, and improvement of the carrying ability of the oil film.

1. INTRODUCTION

Axial piston pump is the most widely used type in hydraulic systems because of its advantages, such as higher power density, higher efficiency, higher pressure limit, compared with other pump types. The sealing and bearing gaps between piston and cylinder, cylinder and valve plate, and slipper and swash plate are the most important and difficult parts in axial piston pumps design. Especially, the optimization of piston/cylinder pair can increase the limit inclined angle of swash plate, which is helpful to improve the power density without requirement for higher quality material. And improvement of the load carrying ability of piston/cylinder pair contributes to the limit pressure improvement of axial piston pump. Meanwhile, the reduction of friction and leakage of piston/cylinder pair can improve the mechanical efficiency and volume efficiency of axial piston pump. Therefore, there has been lots of experimental and simulation researches on piston/cylinder pair design conducted around the world.

Renius (1974) is the first one to investigate the cylinder/piston pair by building a simplified single-piston test bench measuring the axial friction forces. A helpful theoretical simulation tool CASPAR is established by Wieczorek (2000). Olems (2000) designed a test bench to measuring the temperature distribution of oil film between cylinder and piston and compared the results with simulation results from CASPAR. Lasaar (2004) firstly investigated contoured pistons using CASPAR, which showed improvement concerning leakage and friction. He designed a friction tribo pump to verify the simulation results. Huang (2006) made great progress with CASPAR by considering Elastohydrodynamic effect.

With the progress of computer computation and multibody dynamic, the virtual prototype technology is applied in research on axial piston pump. Deeken (2005) built a simulation tool for axial piston pump using commercial software. DSHplus is used to calculate the pressure changeover in piston chamber. Tribological properties are calculated in MATLAB while mechanical properties in MSC.ADMAS. Fatemi (2008) improved this simulation tool by taking elastic deformation into consideration. He obtained good results in piston/cylinder analysis using this simulation tool which is testified by experiment. Zhang (2005) built a virtual prototype of pump with analysis tools such as Pro/Engineer, ADAMS, NASTRAN and EASY5. The analysis of the virtual pump provides an accurate estimation of load and stress of critical components. Roccatello (2007) modeled a variable displacement axial piston pump in a multibody simulation environment using ADAMS and AMESim. He analyzed the shaft torque and compared the co-simulation results of ADAMS and AMESim with Experimental results. Zhang (2009) analyzed the dynamic stress of piston and shaft using a virtual prototype of axial piston pump.

This paper introduces a virtual prototype of axial piston pump built with ADAMS, AMESim and MATLAB, which is applied in analysis and optimization of piston/cylinder pair. And some related experiments are conducted to testify the simulation results.

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2. MODEL CONSTRUCTION

2.1 Motion and Force Analysis of Piston

Figure 1. FORCES EXERTED ON PISTON

To predict the behavior of piston, the forces applied on piston must be analyzed first. Figure 1 shows the all the forces exerted on piston. Due to the pressure force $F_p$ and the spring force $F_s$, piston is pressed on the surface of swash plate. With the driving torque $M_s$, piston rotates around the axis of shaft along with cylinder block, and reciprocates in the piston chamber in the meanwhile. The inertia force $F_{ak}$ and the friction force $F_{fk}$ are related with the translational motion. The reaction force of the swash plate $F_{sk}$ is perpendicular to its surface and thus a lateral force $F_{sky}$ is exerted on the piston which is balanced by the pressure force of oil film. And piston is inclined due to these lateral forces as shown in figure 2. In addition, piston spins about its own axis.

Figure 2. INCLINED PISTON POSITION

In order to calculate the forces applied on piston accurately which is crucial to the analysis of piston/cylinder pair, the virtual prototype of axial piston pump is built including three sub-models constructed in different software according to characteristic of the forces, i.e., mechanical model in MSC.ADAMS, hydraulic model in AMESim and oil-film model in MATLAB described as below.

2.2 Mechanical Model

As Deeken (2003) described, the mechanical model of hydrostatic displacement unit can be portrayed as a multibody system. Mechanical propertied inertia, elasticity and force are assigned to individual discrete element. And individual massy bodies are joined to one another via links or forces.

The mechanical model is built using the commercial software MSC.ADAMS because of its advantage on multibody dynamics simulation and co-simulation capability with AMESim. Firstly, the necessary components of axial piston pump are built in three-dimensional modeling software Solidworks which is used widely in the industrial design. Through the COSMOSMotion interface, the built three-dimensional model is imported into ADAMS with all the properties about material, joint, forces and color assigned in Solidworks environment reserved. Then, with some necessary settings, the three-dimensional model turns into a physical multibody system which can simulate the actual relative motion and force transmission between inner components of piston pump.

Usually, for the analysis of piston/cylinder pair, it is deemed that the piston possesses only two degree of freedom with respect to the cylinder. One is the translational motion along the axis of piston chamber, while the other is the rotation about the same axis. However, the motion of piston is much complicated in reality as depicted in Figure 2. Therefore, piston and cylinder bore (or bushing) are coupled through contact instead of cylindrical joint.

Due to the high pressure of oil film between piston and cylinder chamber, there is a deformation of 12 to 14 um of piston which is small in comparison to the piston diameter but large compared with the clearance width of lubricating film, which is usually 15 to 20 um (Gel2 (2010)). In order to take the deformation of piston into consideration, the mechanical model takes the piston as flexible body, which increases the computation time dramatically. Moreover, it is necessary to chose regular hexahedral finite element mesh rather than irregular tetrahedral mesh for the transmission of pressure force of oil film calculated from oil-film model. And the density of meshing grids must be in accordance with oil-film model. Thus, the solver of ADAMS can simulate the dynamic properties of stress and deformation distribution of piston.
2.3 Hydraulic Model

The pressure force $F_p$ is very important in the analysis of piston/cylinder pair. And the hydraulic model is used to calculate the changeover process of an axial piston pump. The pressure pulsations arising from the kinematic characteristics have a direct influence on the dynamic property of piston.

AMESim is chosen as the simulation environment for hydraulic model. Firstly, the single piston chamber model is built using the components from AMESim libraries and packaged as an independent unit which is depicted as Figure 3. The three important leakages, i.e., $q_{v1}$ between piston and cylinder, $q_{v2}$ between slipper and swash plate and $q_{v3}$ between cylinder block and valve plate are included. Leakage $q_{v1}$ is calculated using the leakage and viscous with variable length and eccentricity model while $q_{v2}$ and $q_{v3}$ are simplified as fixed hydraulic orifices (Nař and Murrenhoff (2008)) as shown in Figure 3. The hydraulic model of an axial piston pump consists of $N$ piston models. $N$ is the number of piston.

The variation of discharge area of piston bore is the key factor determining the pressure in piston chamber, especially the transition region where the geometrical cross-sectional area changes fast due to the relief grooves as depicted in Figure 4. And the variation of discharge area, which is integrated in the hydraulic pump model as valve plate model as shown in Figure 5, may be described in a piecewise function divided into more than ten sections.

The hydraulic model of axial piston pump also includes the displacement-varying mechanism as shown in Figure 5. Therefore, the hydraulic model has the abilities of optimization of valve plate and displacement-varying mechanism, in addition to simulating the pressure in piston chamber and flow ripple of pump.

2.4 Oil-film Model

The oil film between piston and cylinder has two main functions. It works as a hydrodynamic bearing and as a sealing of the displacement chamber. The oil-film model is to calculate the pressure distribution which is the core in analysis of piston/cylinder pair, the leakage and internal friction which is related to the power loss of pump.

$$\frac{\partial}{\partial x}\left(\frac{h' \frac{\partial p}{\partial x}}{12\eta \frac{\partial h}{\partial x}}\right) + \frac{\partial}{\partial y}\left(\frac{h' \frac{\partial p}{\partial y}}{12\eta \frac{\partial h}{\partial y}}\right) = 6\left(\frac{\partial h}{\partial x} + \frac{\partial h}{\partial y} + \frac{\partial h}{\partial t}\right)$$

The depth of oil film is much less than the other two dimensions, therefore the pressure distribution can be computed in a plain coordinate system with the Reynolds Equation (1) which is solved numerically with the well-known finite volume method described in Patankar (1980). Then, the calculated pressure distribution is applied in the calculation of friction force and leakage. When solving the Reynolds Equation mathematically, pressure values below zero are set equal to saturation pressure.
As mentioned above, the piston is inclined in cylinder bore. When the inclined angle is large to some extent, the oil film at some point is so thin that solid contact of the roughness peaks occurs due to the surface roughness, which means the carrying ability is not enough and mixed friction occurs. The fraction of solid contact pressure between roughness peaks is calculated with the model described in Olems (2000), which yields the value of pressure as the function of gap height, roughness, elastic module of materials and radius of piston. The superposition of the fluid pressure fraction and the solid-contact pressure fraction gives the punctual value of pressure which is exerted over the surface of piston. Because of the reciprocation of piston in the cylinder bore, the length and position of the oil film relative to piston varies to time. For the calculation of deformation of piston, the pressure distribution matrix transferred to mechanical model can be divided into three parts as shown in Figure 6 (with bushing). The oil film refers to gap between piston and bushing. The pressure of parts exceeds bushing are set to pressure in case or chamber. And their lengths vary with the position of oil film relative to piston.

2.4 Co-simulation Model
The establishment of virtual prototype of axial piston pump has to solve the real-time data transfer between three sub-models. The co-simulation model can realize the fluid-solid coupling and rigid-flexible coupling. Considering the advantage of ADAMS in visualization during modeling and simulation, ADAMS is chosen as the tool to build the user operation interface.

The data flows between three sub-models are shown in Figure 7. The mechanical model transfers the velocity, displacement and other kinematic properties to hydraulic model and oil-film model. Besides, the mechanical model also provides the deformation of piston surface to oil-film model. The hydraulic model provides pressure in piston chamber to mechanical model. And this pressure is also the boundary condition in oil-film model simulation. The oil-film model provides pressure distribution and friction force to mechanical model. Therefore, there are five dataflow among three sub-models. In order to simplify the co-simulation model, the oil-film model is converted to one component of AMESim library through the SL2AME toolbox. Thus, the oil-film model is integrated into hydraulic model. And there exist only two dataflow between ADAMS and AMESim in the virtual prototype of axial piston pump.

However, there are more than one thousand nods in oil-film pressure distribution matrix. The number of nods of piston is much large. And the numbering of the two matrixes is different. How to apply these pressure forces to the corresponding nods of piston automatically is still under investigation. So the deformation of piston can’t be calculated dynamically. And the oil-film pressure can only be transferred as concentrated force instead of distributed force.

3. EXPERIMENTAL INVESTIGATION
To verify the basic simulation model as presented in this research, some test rigs are designed to measure to the microscopic properties of piston/cylinder pair. Renius (1974) was the first one investigating the piston/cylinder pair experimentally with a single-piston test bench. After that, most of the relative researches are based on similar inverse kinematical structure which neglected the centrifugal force of piston, such as Scharf (2005) and Manring (1999). Olems (2000) was the first to conduct experimental research on a standard axial piston pump whose measurement results were more reliable than single-piston test rig theoretically. In this research, all the relative
measurements are based on standard axial piston pumps, including pressure distribution of oil film, friction force between piston and cylinder bore, and pressure in displacement chamber under normal operating conditions. However, only the pressure-distribution measurement pump has been built so far. The other two have been designed and are still under construction. More details can refer to Zhang (2009).

The structure of axial piston pump for measurement of pressure distribution is illustrated in Figure 8. The cylinder block is modified for the installation of pressure sensors which should be very small. The wires of the pressure sensors are placed in the hole of the specially manufactured shaft, and then are connected to the telemetry which works as a measuring amplifier and gives the values to the data acquisition system. This structure successfully avoids problems associated with data transfer from rotating parts.

Figure 9 shows the comparison of measured and simulated results of the pressure distribution of oil film between piston and cylinder at the same time and position. Both the simulation and experimental results show the pressure increase due to micro motion. And the predicted position of pressure peak from simulation confirms with measurement. However the values of pressure peaks of simulation are a little larger than measurements. One reason is that the pressure sensors are limit. So the maximum value may be missed. This is also the reason why the experimental result is less smooth than simulation result. The other reason is that the current model hasn’t implemented the coupling of rigid and flexible. Gels (2010) and Huang (2006) both pointed out that the pressure distribution of rigid model was large than flexible model. This is also the key problems we strive to solve in the next step.

4. SIMULATION AND ANALYSIS

Based on the current virtual prototype of axial piston pump, some simulation researches about piston/cylinder pair are conducted. During the analysis, four parameters are selected as the evaluation criterion. The first one is the maximum value of pressure distribution. The higher is the peak value, the easier it cause fatigue damage to piston. The second one is the mixed friction area, where the roughness peaks contact. The larger is the area, the shorter is the working life of piston because of the abrasion. The other two are axial friction and leakage, which are related to power loss of pump. And some conclusions are introduced with the simulation.

4.1 Influence of Load Pressure

![Figure 10. PRESSURE IN CHAMBER](image)
Four load pressures of axial piston pump are chosen in the simulation as depicted in Figure 10. There exist pressure overshoots during the changeover processes. And the pressure ripple is very clear due to the flow ripple of piston pump. The rotation speed is 1500 rpm. And the inclined angle of swash plate is 12 degree.

Figure 11(a) demonstrates that the maximum values of pressure distribution increase exponentially with the increase of load pressure. The peak value of oil film is more than five times of the pressure in chamber, which proposes higher requirement for used material. The mixed friction areas increase dramatically too with the increase of load pressure as shown in Figure 11(b), which means the abrasion increases with the increasing load pressure. And it stays complete oil lubrication when the load pressure is less than 12Mpa. In addition, it shows that the most severe abrasion occurs around the outer and inner dead centers, which is consistent with what the well-known Stribeck-curve describes. The power loss also increases because the axial friction force which is related to mechanical efficiency and leakage which is related to volumetric efficiency both increase with the increase of load pressure. This is because the velocity and velocity gradient of oil film during the discharge process increase synchronously with the increase of load pressure.

Therefore, the load pressure has a great influence on the properties of piston/cylinder pair. On the one hand, the efficiency decreases due to the increase of load pressure.

On the other hand, the life of piston decreases due to increasing abrasion and fatigue. The property of piston/cylinder pair is one of the reasons why the limit pressure of current axial piston pump couldn’t be higher than forty MPa.

4.1 Influence of Inclined Angle of Swash Plate

As shown in Figure 12, four different inclined angles of swash plate are chosen in the simulation. And the load pressure is set as 28MPa which is the rating pressure, while the rotation speed is 1500 rpm.

Figure 12(a) shows that the maximum value of pressure distribution increase sharply when the inclined angle of swash plate exceed 12 degree. And under the rating pressure, the chosen axial piston pump shouldn’t works for a long time at the maximum inclined angle because the pressure peaks are much larger than allowable pressure of the practically used material, which will cause damage to axial piston pump easily. In the meanwhile, 12 degree is also the dividing line between oil lubrication and mixed friction. And the area of mixed friction has a great increase when the inclined angle of swash plate exceeds 12 degree as depicted in Figure 12(b). Figure 12(c) shows that the axial friction force increases with the increase of inclined angle. The one hand, this is because the displacement of piston during the same while increases with the increase of inclined angle of swash plate, which results in the increase of velocity gradient of oil film. The
other hand is because of increase of mixed friction area. The axial friction force is calculated with the Newton’s friction law, in which the friction is proportional to velocity gradient, instead of mixed friction model. Therefore the calculated friction force will be less than the practical value at the mixed friction state. However, the oil film where mixed friction occurs is much thinner. And the velocity gradient is much larger. So, the increase the mixed friction area accompanies with the increase of axial friction force although the calculated force is theoretically smaller. The leakage almost stays the same because the increase of velocity causes the increase on the flows through the gap between piston and cylinder both at suction process and discharge process.

To sum up, the dramatic increase of values of pressure peak under rating pressure determines the maximum inclined angle of swash plate of axial piston pump. And the larger is the inclined angle, the larger is the power loss of piston pump.

### 4.1 Influence of Clearance

![Figure 13. INFLUENCE OF CLEARANCE](image)

The average height of clearance of piston/cylinder pair is related to the assembly error. Three different clearance heights are selected in the simulation. And the load pressure is set as 28MPa, while rotation speed is 1500rpm. The inclined angle of swash plate is 12 degree.

The maximum value of distribution and the mixed friction are both decrease with the decrease of clearance height as shown in Figure 13. It means the decrease of clearance height will improve the carrying ability and decrease the abrasion. As explained above, the lateral force from swash plate is balanced by the squeeze effect resulting from the micro motion of the inclined piston in cylinder bore. And the squeezing effect refers to the division of the variation of depth to the average piston bore, which will decrease the area of mixed friction. Nevertheless the clearance height shouldn’t becomes too small because of surface roughness.

The smaller is the clearance height, the smaller is the velocity gradient. So the decrease of clearance height will increase the axial friction force. However, Figure 13(c) gives the opposite simulation results. This is because the effect of mixed friction on axial friction force is dominant. The leakage decreases clearly with the decrease of clearance height as shown if Figure 13(d). This is because the average section area of oil film decrease with the clearance height becomes smaller.

![Figure 14. EXPERIMENTAL RESULT ON INFLUENCE OF CLEARANCE](image)

The experiments of Bartelt (2003) and Brückelmann (2010) give the same results. Figure 14 is the experimental results of Bartelt. The clearance is 0.7‰ or 1.7‰ of piston diameter, with other parameters stay the same. $\eta_m$ is the mechanical efficiency of the smaller clearance. While $\eta_{or}$ is the mechanical efficiency of original clearance. Four lines show the experimental results under four different rotation speeds. It shows that the smaller clearance height will improve the mechanical efficiency of axial piston pump especially under low rotation speed. This experiment verifies the simulation about the influence of clearance and the virtual prototype model to some extent. Some similar results will also be conducted with the designed test rigs of author in the near future.
Therefore, within a certain range, a smaller clearance is beneficial to the improvement the carrying ability of piston/cylinder pair and the efficiency of pump. However, the smaller clearance calls for improvement of assembly error, which will raise the manufacture cost.

5. CONCLUSIONS

(1) On the basis of sub-models built in different domains and the interfaces, it is feasible to make construct the virtual prototype of axial piston pump, which can realize the liquid-solid coupling and rigid-flexible coupling. This simulation model has a great potential in the analysis and design of axial piston pump, not only about piston/cylinder pair.

(2) The structure of piston/cylinder pair is crucial to improve the power density of axial piston pump because the properties of piston/cylinder pair are key factors in determining the limit load pressure and inclined angle of swash plate.

(3) Within a certain range, a smaller clearance is beneficial to the improvement the carrying ability of piston/cylinder pair and the efficiency of pump.

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