Fluid-Structure Coupling Analysis of Non-linear Reciprocating Piston Seals in Series

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Abstract. The reciprocating piston seals are crucial parts in hydraulic system which are widely used in aerospace and military industry. A direct fluid-structure coupling method with high efficiency is proposed for solving the transient Elasto-Hydrodynamic-Lubrication (EHL) problem in hydropneumatic suspension reciprocating piston seal system. Detailed 3D fluid-structure coupling model is built using finite element discretization. Material tests are carried out to obtain the parameters of the 3rd-oder Ogden constitutive model of the rubber O-ring. The sealing performance and friction force of the sealing system are analyzed in different piston speed. The critical speed from mixed lubrication to full film lubrication is obtained.

Keywords: Fluid-structure coupling; Reciprocating piston seal; Elasto-hydrodynamics.

1. Introduction

The reciprocating seal is small but very important hydraulic component which determine the reliability and performance of the whole system. They are widely used in industries such as military vehicle, aerospace and so on. The working environment is usually severe for the sealing component such as high pressure, high reciprocating velocity and extreme temperature. In order to improve the performance of reciprocating seals, many innovative sealing structures have been proposed [1–3]. Efforts have been made by White and Denny since 193s to reduce the leakage and friction force in piston reciprocating movement [4]. Polypac PHD seals are proposed to be used in high pressure condition for piston sealing. VL seal are designed to replace step-seal, which has premium sealing performance and much lower friction than step-seal [5]. Seals with metal wire frame are proposed by Russia for use in extreme high pressure and high temperature condition in aerospace.

B Wennehorst and GWG Poll use 3D sinusoidal geometry to model seal lip surface roughness at micro-level [6]. Fluid-structure coupling simulation is made to obtain the friction and lubrication performance of the roughness model. The measured friction force coincides well with viscous lubricant friction force in soft EHL fluid-structure computation. Y Öngün makes a research on the fluid film thickness of piston rubber seals using FSI method [7]. Philip C. Hadinata makes an elasto-hydrodynamic analysis of radial sealing with different micro-asperity patterns on rod surface. The result shows that proper rod surface micro-asperity geometry shape improves both the sealing and load capacity performance.

Reynolds equation is widely used to obtain the sealing film thickness and pressure distribution in dynamic lubrication. However, it is not suitable for series sealings in mixed lubrication state. Firstly, the
whole fluid domain for series sealing is complicated. There is no oil film between adjacent seals. It should be artificially divided into the hydrostatic pressure zone without oil film and the EHL zone if the Reynolds equation is used because it is only applicable to the EHL zone. Accurate definition of the boundaries of the EHL zone is a difficult task because the dynamic deformation of the sealing zone in the piston moving process. The poor boundary definition will bring inaccuracy to the numerical solutions. Secondly, the Reynolds equation solution is based on iterative method. The errors caused by time lag is inevitable. The transient development processes of the lubricating films have important influence on the sealing performance especially in high pressure and high velocity condition. Direct fluid-structure coupling method has better accuracy in this condition. So, The Navie- Stokes equations are necessary to be used to consider the interaction between seals and fluid. Fluid-Structure Interface (FSI) coupling dynamic simulation has great advantages in solving the complicate transient Elastic-Hydrodynamic-Lubrication (EHL) problem. Until now, only simple seal such as step-seal or U-cup seal has been researched using FSI method. In reality, seals are used in series especially in high pressure and high velocity condition. The fluid-structure coupling process and sealing mechanism for complicated series seals used in severe condition has not been researched.

In this paper, a direct fluid-structure coupling method using finite element model with high solving efficiency is proposed. The 3D fluid-structure coupling model of series seals used in hydropneumatic suspension is built. The Elasto-hydrodynamic simulation in mixed and dynamic lubrication state is carried out using proposed fluid-structure coupling method. Suggestions are proposed for improving sealing and lubrication performance based on FSI analysis results.

2. Direct fluid-structure Coupling Method

There are two different methods available for fluid-structure coupling dynamics simulation. One is iterative method in which the fluid Navier-Stokes equation and structure dynamic equation are solved individually in succession. The fluid variables solved in the latest iteration are used as initial condition for the next iteration solving structure equation and vice versa. When the tolerance of each variable specified by user is satisfied and the solution time is reached, the solution process is complete. Another is direct method solving the fluid-structure coupling equation simultaneously. All the conservation equations are satisfied in this method. Direct method is proposed using finite element discretization of the fluid domain in this paper.

Fluid-structure Coupling Algorithm. The dynamic coupling of fluid and structure is based on the equilibrium of displacement and stress on the fluid-structure interface.

\[ d_f = d_s \]

\[ n \cdot \tau_f = n \cdot \tau_s \]

Where \( d_f, d_s \): fluid and solid displacements on the interface. \( \tau_f, \tau_s \): fluid and structure stresses.

The fluid velocity on the interface is resulted from derivative of coupling structure displacement.

\[ v_f = d_s \]

On the other hand, the fluid pressure (nodal stress) is integrated into fluid force and exerted onto the structure node

\[ F(t) = \int h^i \cdot \tau f dS \]

Where \( h^i \) is the virtual quantity of the solid displacement.

In this way, the fluid pressure affects the structural deformations and the solid displacement affects the flow pattern. The coupled fluid-structure equation is a nonlinear system regardless of the solid model used, since the fluid equations are always nonlinear.

The fluid and the solid equations are combined into one matrix system through displacement and stress coupling equations on interface proposed above. Variables in both fluid and solid equations are solved simultaneously in one matrix. Newton-Raphson method is used for solving the non-linear fluid-structure dynamic coupling equations at specific time. The coupling dynamic equations are linearized in a matrix system at time \( t \) and at iteration number \( k \) as follows.
\[
\begin{bmatrix}
A_{ff} & A_{fs} \\
A_{sf} & A_{ss}
\end{bmatrix}
\begin{bmatrix}
\Delta X^k_S \\
\Delta X^k_f
\end{bmatrix} =
\begin{bmatrix}
-F_f X_f^k + \lambda_d d_{is}^k + (1 - \lambda_d) d_{is}^{k-1} \\
-F_s X_s^k + \lambda_t \tau_{ij}^k + (1 - \lambda_t) \tau_{ij}^{k-1}
\end{bmatrix} = [F]
\]

And

\[
X^k = X^{k-1} + \Delta X^{k-1}
\]

\[
A_{ij} = \frac{\partial F_i^k}{\partial X_j^k} \quad (i, j = f, s)
\]

\[
X = [u, v, \omega, p, d, T \ldots \ldots]
\]

\(A_{ij}\) are Jacobian matrixes of fluid and structure system, \(\lambda_d, \lambda_t\) are displacement and stress relaxation factors which are used to promote convergence in difficult cases. \(X\) is the solution vector that includes all basic variables in the coupling system.

**Fig. 1** The computation process of direct fluid-structure dynamics coupling

### 3. Model Description

**Geometry and fluid-structure Coupling Model.** To satisfy the requirement of high-pressure sealing, the piston sealing system is designed with three different combination seals in series. The sealing rings in outside piston grooves are made of poly-tetra-fluoroethylene (PTFE) material which are supported with Nitrile Butadiene Rubber (NBR) O-rings. The seals in the middle is composed of \(
\square
\)-shape rubber ring, PTFE step ring and O-rings.

The dimension of the sealing system is listed in table 1. The cylinder and piston is made of steel while the step seals are made of PTFE combined with 30% bronze powder in volume fraction. The PTFE seals are assembled into piston groove in high temperature condition. They are in interference fit with cylinder and O-ring in normal working condition.

**Table 1. Dimension of the piston seals (unit: mm)**

| Component           | Outer Diameter | Inner Diameter | Thickness |
|---------------------|----------------|----------------|-----------|
| Middle Sealing Ring | 79.6           | 65.7           | 8.0       |
| Middle-Ring         | 80.5           | 78.0           | 2.37      |
| Side Sealing Ring   | 80.0           | 74.0           | 5.9       |
| Middle O_Ring       | 72.5           | 67.1           | /         |
| Side O_ring         | 72.4           | 62.1           | /         |

Hexahedron elements are used for the 3-D finite element model of the piston seals and fluid. In the normal working condition, the oil film between the cylinder and seal lip is generally as thin as 1 micrometer. Considering the balance of computation accuracy and cost, 0.01 mm is chosen as the basic
element size in axial and circumferential direction for the fluid model. Three layers of elements are built for oil film in thickness direction with initial thickness of 0.01mm. The local element size for structures near the fluid-structure coupling surface is 0.1mm. The element size increases to 1mm gradually when it is far away from the coupling surface as shown in figure 2. The friction coefficient are obtained by friction measurement. 0.04 is used for PTFE-steel contact and 0.1 for rubber-steel contact in friction force computation. There is 0.5 mm interference between supporting rubber rings and step seals at initial state in assembly condition. Initial penetration is established in the finite element model. It is gradually eliminated in the process of contact computation. The contact pressure is built between contact parts in the assembly. The total element number of the structure and fluid model is 0.215 million and 0.765 million separately.

**Initial and Boundary Condition.** The fluid and structure finite element models for the piston sealing system are built individually. They are coupled through fluid-structure interfaces which share the same geometry position. No slip boundary is defined on the fluid-structure coupling surfaces. Arbitrary Lagrange Euler (ALE) method is used to make the fluid mesh move synchronously with structure. The initial fluid pressure is 6 MPa which is imposed on the inlet and outlet surfaces. The Reynolds number of the flow in the sealing gap is less than 1000 at the maximum piston speed 1.5m/s. As a result, the fluid is modeled as laminar flow. The cylinder is fixed on end surfaces. The compression ratio is 15% between O-ring and PTFE step seal. Different uniform velocities is imposed on the float piston along axial direction.

The 3rd-order Ogden constitutive model is used for the Nitrile Butadiene Rubber (NBR) rubber material. The initial circumferential strain is allowed for consideration in the model. Stress-strain tests are performed to gain the model parameters at 60°C including uniaxial tension test, equibiaxial tension test and planar shear test. The engineering stress-engineering strain data from different tests are shown in figure 3.

![Fig. 2 3D Finite Element Model of the piston seal and fluid](image)

| Material                  | Elastic/Bulk Modulus (MPa) | Poisson Ratio | Density (Ton/mm3) | Viscosity (MPa·s) |
|---------------------------|----------------------------|---------------|-------------------|-------------------|
| Steel                     | 2E5                        | 0.3           | 7.89E-9           | /                 |
| PTFE with 30% Bronze      | 5200                       | 0.4           | 3.9E-9            | /                 |
| Nitrile Rubber            | 2400                       | 0.498         | 9.6E-10           | /                 |
| 10# Aviation Hydraulic Oil| 1E20                       | /             | 8.5E-10           | 8.7E-9 (60°C)     |

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**Table 2. Material Properties for the Structural Parts and Oil of the Sealing System**
Fig. 3 Rubber material stress-strain test result at temperature 60°C (Left: Equibiaxial tension test; Bottom: Planar shear test)

4. Simulation Result

Piston Sealing System Fluid-Structure Coupling Analysis Result. The fluid-structure coupling simulation of the cylinder-seal-piston system using direct coupling method proposed above is performed. At first the assembling process in which the seal rings are put into the piston grooves is simulated. The deformation and contact pressure of the series seals in assembly state is shown in figure 4.

Fig. 4 The deformation and pressure of fluid-structure coupling model in assembly state (Left: VonMises Stress; Middle: Cylinder Contact pressure; Right: O-ring Contact Pressure)

The maximum deformation is 0.52mm which lies on top of the O-ring. The maximum Von Mises stress is 56.0 MPa. It concentrates on the seal lip which contacts with cylinder inner surface. The maximum contact pressure for the O-ring is 3.47MPa while for the cylinder inner surface it is 4.2MPa. The contact pressure combined with piston speed determines the seals lubrication state and oil film thickness. It is a key factor for balancing the piston sealing performance and friction force.

The fluid pressure distribution in the sealing gap is shown in figure 5 in the piston axial direction.

Fig. 5 The fluid pressure distribution along piston axis
The fluid pressure on the inlet and outlet surface is 6MPa, which equals to the working pressure in the cylinder. The piston-cylinder clearance is 0.01mm while the cylinder-sealing gap is 1μm. The pressure inside the sealing gap is 2.6~2.7 MPa. It is much lower than the inlet and outlet pressure. The possible reason is that the fluid velocity increases dramatically in the gap between piston and cylinder. Thus the fluid pressure decreases according to the Bernoulli equation. It is favorable for reverse pumping in instroke process which improves the sealing performance.

When the piston moves from left to right at 0.5 m/s, the deformation and stress distribution of the piston seals fluid-structure coupling model is shown in figure 6.

![Fig. 6 The deformation and stress of fluid-structure coupling model in moving state](image)

In the piston rightward moving process, the O-rings and PTFE seals are pressed onto the left side of piston grooves. The left PTFE step seal has a trend to turn anti-clockwise. The seal outer surface is pressed onto the cylinder inner surface which tends to increase the total friction force. The maximum Von Mises stress for the O-ring is 2.9 MPa as shown in figure 8. The fluid mesh moves with piston at the same speed. It is compressed in the thickness direction with the seal movement and deformation. The inside fluid elements deform uniformly. As is seen there is no severe fluid element distortion in the moving process.

The piston friction force and fluid viscous force at piston speed from 0 to 1.5m/s is shown in figure 7. At static state, there is no hydrodynamic pressure in the sealing gap. The cylinder inner surface and piston seal lip are in direct contact. When the piston starts to move, the friction force is 1512 N which is very large. It decreases sharply with speed below 0.2 m/s. The friction force between cylinder and piston seals becomes zero at the piston speed of 0.4m/s. It means the dynamic fluid pressure lifts the seals fully and the dynamic oil film is formed. It is considered to be in dynamic lubrication above speed of 0.4 m/s. The piston seals are in mixed lubrication state at piston speed less than 0.4 m/s. The viscous shear force between cylinder and seals increases gradually with speed. When the piston speed is 1.5m/s the viscous shear force reaches the maximum value 410N. The total resistance force decreases first and reaches the minimum value at speed of 0.35 m/s. It increases slowly at high speed because of the increasing fluid viscous force.
5. Conclusion
A direct fluid-structure coupling method is proposed to solve the transient EHL problem in the reciprocating piston seal system. Detailed 3D fluid-structure coupling finite element models are built for piston seals and oil film in the sealing gap. The sealing performance and friction force is analyzed at different piston velocities. Different micro-asperity geometries on cylinder surface are researched for their influences on sealing performance. The results are as follows:

In the assembly state, the maximum Von Mises stress of the piston seals is 56.0 MPa. The contact pressure for the supporting O-ring is 3.47MPa and for the cylinder inner surface it is 4.2MPa. The fluid pressure in the piston sealing gap is 2.6~2.7 MPa while it is 6.0 MPa near the inlet and outlet surface.

In the piston rightward moving process, the left PTFE seal has a trend to turn anti-clockwise. The maximum Von Mises stress for the O-ring is 2.9 MPa. The friction force caused by direct contact decreased to zero at the piston speed of 0.4m/s. it is considered to be the critical speed for mixed lubrication and full film lubrication. The viscous shear force increases slowly with speed and reaches the maximum value 410N at speed of 1.5 m/s. The leakage rate increases with cylinder oil pressure.

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