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Research on thermal-fluid-structure coupling of valve plate pair in an axial piston pump with high pressure and high speed

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Abstract

Purpose – High pressure and high speed of the axial piston pump can improve its power density, but they also deteriorate the thermal-fluid-structure coupling effect of the friction pairs. This paper aims to reveal the coupling mechanism of the pump, for example, valve plate pair, by carrying out research on multi-physics field coupling.

Design/methodology/approach – Considering the influences of temperature on material properties and thermal fluid on structure, the thermal-fluid elastic mechanics model is established. A complete set of fast and effective thermal-fluid-structure coupling method is presented, by which the numerical analysis is conducted for the valve plate pair.

Findings – According to calculations, it is revealed that the temperature and pressure evolution laws of oil film with time, the pressure distribution law of the fluid, stress and displacement distribution laws of the solid in the valve plate pair. In addition, the forming history of the wedge-shaped oil film and mating clearance change law with rotational speed and outlet pressure in the valve plate pair are presented.

Originality/value – For an axial piston pump operating under high speed, high pressure and wide temperature range, the multi-physics field coupling analysis is an indispensable means and method. This paper provides theoretical evidence for the development of the pump and lays a solid foundation for the research of the same kind of problem.

Keywords Axial piston pump with high-pressure and high-speed, Thermal-fluid elastic deformation, Thermal-fluid-structure coupling, Valve plate pair

Paper type Research paper

Nomenclature

dp_r, dp_θ, dp_z	= radial, tangential and axial volume force increment, N/m^3;
$d\bar{p}_r, d\bar{p}_\theta, d\bar{p}_z$	= radial, tangential and axial surface force increment, N/m^2;
du^*, dv^*, dw^*	= tiny virtual radial, tangential and axial displacement increment, m/rad/m;
u_r, u_θ, u_z	= radial, tangential and axial speed, rad/s, m/s;
$\boldsymbol{u}$	= speed vector;
$\{f\}$	= displacement vector at some point, rad, m;
r, θ, z	= coordinate axes;
N_i	= shape function;
$[N]$	= shape function matrix;
$\{T\}^e$	= temperature vector, K;
dP_r, dP_θ, dP_z	= radial, tangential, axial concentrated force increment, N;
$[B]$	= geometry matrix;
$[D]$	= elastic matrix;
R', T', Z'	= radial, tangential, axial unit mass force, N/kg;

T	= temperature, K;
T_0	= reference temperature, K;
R_p	= piston distribution radius, m;
ρ	= density, kg/m^3;
η	= oil viscosity, Pa.s;
η_0	= oil viscosity at temperature T_0', Pa.s;
$d\{\varepsilon\}$	= total strain increment;
$d\{\varepsilon\}_e$	= elastic strain increment;
$d\{\varepsilon\}_T$	= thermal strain increment;
$d\sigma_r, d\sigma_\theta, d\sigma_z$	= radial, tangential, axial stress increment, Pa;
$d\varepsilon_r^*, d\varepsilon_\theta^*, d\varepsilon_z^*$	= virtual radial, tangential, axial strain increment;
$d\tau_{r\theta}, d\tau_{\theta z}, d\tau_{zr}$	= shear stress increment in $r\theta$, θz, zr planes, Pa;
$d\gamma_{r\theta}^*, d\gamma_{\theta z}^*, d\gamma_{zr}^*$	= virtual shear strain increment in $r\theta$, θz, zr planes;
$\{\delta^*\}^e$	= virtual nodal displacement vector of an element, m;
$d\{\delta^*\}^e$	= virtual nodal displacement increment vector of an element, m;

$d\{\delta\}^e$	=	nodal displacement vector of an element, m;
$\{\varepsilon\}$	=	strain vector;
$d\{\tilde{\varepsilon}\}_T$	=	initial strain at each iteration;
$d\{\tilde{\sigma}\}_T$	=	initial stress at each iteration, Pa;
$\{\alpha\}$	=	thermal expansion array, $10^{-6}/K$;
α	=	thermal expansion coefficient, $10^{-6}/K$; and
ϖ	=	relative angular speed, rad/s.

Introduction

A higher rotational speed of the axial piston pump is required in aircraft for achieving adequate power density. The speed can reach 9,000 rpm, even 16,000 rpm. Thus, the heat induced by high pressure and high speed increases during the operating process. This heat makes the oil viscosity change drastically, thereby changing the bearing, sealing and lubricating characteristics. In addition, the heat and high pressure increase the local deformation in the core components. The inadequate lubrication and inconsistent deformation make the friction pairs seriously wear. These further increase the oil leakage and generated heat and make the oil film formation more difficult. Therefore, as one of the key friction pairs, there is an urgent need to study valve plate pair from the perspective of multi-physics field coupling.

Many scholars have studied the axial piston pump from many aspects. For the oil film of the valve plate pair, Frnaeo (1961) used the proposed method under constant film thickness assumption, and Yamaguchi first used the numerical method by considering the influence of the wedge-shaped film (Yamaguchi, 1966) and conducted further research by adding an unsteady item in Reynolds equation (Yamaguchi, 1984). Pan *et al.* (1989) carried out some research studies by using the finite difference method. Mandal *et al.* (2012) studied the effects of flow inertia and valve-plate geometry on the axial piston pump. Bergada *et al.* (2012) discussed how barrel film thickness and barrel dynamics of an axial piston pump depend on oil pressure and temperature. Richardson *et al.* (2017) pointed out that the motion of the floating valve plate directly affected the lubricating pressures between valve plate and cylinder block. For the sea water axial piston, Yang *et al.* (2015) studied the influence of pre-compression angle, friction coefficient and clearance on the torque. Xu *et al.* (2017a, 2017b) analyzed the influence of dimensional and geometrical errors on the cylinder block tilt (Xu *et al.*, 2017a, 2017b) and studied hydro-mechanical losses of various friction pairs of the pump.

Ivantysynova and team carried out numerous studies on the axial piston pump. For the piston pair, Ivantysynova and Huang (2002, 2002) first considered the elasto-hydro-dynamic effect, and Pelosi and Ivantysynova (2009) first presented the thermal-fluid-structure coupled model. In 2016, Shang and Ivantysynova (2016) used the fluid structure and thermal interaction model to investigate the complex fluid behavior of the cylinder block/valve plate interface, and Chacon and Ivantysynova (2016) studied the impact of the elastic deformation due to pressure and thermal loadings of the end case/housing on the performance of the cylinder block/valve plate interface, but for the fluid, only oil film was considered. In addition, for slipper pair or external gear machine, research on

thermos-elasto-hydro-dynamic lubrication has been conducted (Tang *et al.*, 2017; Hashemi *et al.*, 2017; Thiagrajan *et al.*, 2015; Dhar and Vacca, 2015).

By analyzing and summarizing the literature, for the valve plate pair of the axial piston pump with high pressure and high speed, with the increase in rotational speed and operating pressure, high-speed shear flow and high-pressure difference flow make transient flow and viscous heating more obvious. Therefore, future research should be focused on the transient analysis and thermal-fluid-structure coupling.

Therefore, we consider the effects of temperature, high speed and high pressure for conducting transient thermal-fluid-structure coupling research for the valve plate pair.

Problem formulation

The flow distributor mechanisms are mainly composed of valve plate, cylinder block and center spring, as shown in Figure 1. Valve plate is used to insulate oil, distribute oil and bear the load from the cylinder block, which is fixed at the rear end of the pump housing by a side pin. Cylinder block rotates at a constant speed, which is driven by a spline of the rotational shaft. When the pump starts, the relative motion exists between cylinder block and valve plate, and the oil in the piston chambers together with the spring presses the cylinder block toward the valve plate. Lubrication film is immediately formed and a certain gap is maintained between the two, thereby avoiding direct contact friction.

In the valve plate pair, under high speed and high pressure, temperature rises of valve plate and cylinder block and oil in them, and deformations of valve plate and cylinder block are larger. They are not completely the same at different locations, resulting in higher stress and wedge-shaped clearance, and interactions exist among them. These usually cause a series of consequences, involving unsteady operation of the pump, insufficient pumping oil pressure and partial wear. Moreover, the consequences determine efficiency, operation performances, reliability and service life of the pump. Therefore, research on thermal-fluid-structure coupling in the valve plate pair is imperative.

Thermal-fluid elastic mechanics model

Suppose that the valve plate pair is loaded by volume force (such as temperature load), surface force (such as fluid pressure) and concentrated force and its research region is

Figure 1 Axial piston pump with high pressure and high speed
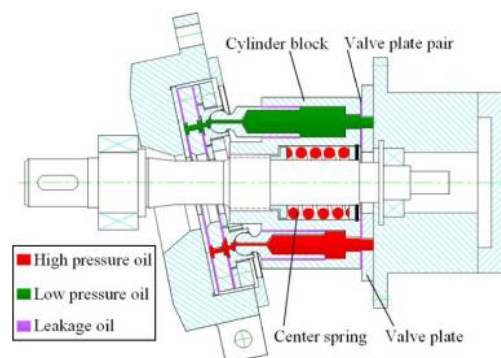
discretized as finite elements. For each element e, according to the incremental virtual work principle, the equation is established as follows:

$$\sum_e \int_V (d\sigma_r d\varepsilon_r{}^* + d\sigma_\theta d\varepsilon_\theta{}^* + d\sigma_z d\varepsilon_z{}^* + d\tau_{r\theta} d\gamma_{r\theta}{}^*$$
$$+ d\tau_{\theta z} d\gamma_{\theta z}{}^* + d\tau_{zr} d\gamma_{zr}{}^*)dV$$
$$- \sum_e \int_V (dp_r du^* + dp_\theta dv^*)dV - \sum_e \int_V (dp_z dw^*)dV$$
$$- \sum_e \int_{S_\sigma} (d\overline{p}_r du^* + d\overline{p}_\theta dv^* + d\overline{p}_z dw^*)dS$$
$$- \sum_e (dP_r du^* + dP_\theta dv^* + dP_z dw^*) = 0 \qquad (1)$$

In an element, using the virtual nodal displacement $\{\delta^*\}^e$, the virtual displacement $\{f^*\}$ at any point is expressed as:

$$\{f^*\}^e = [N]\{\delta^*\}^e \qquad (2)$$

The virtual strain $\{\varepsilon^*\}$ is defined as:

$$\{\varepsilon^*\} = [B]\{\delta^*\}^e \qquad (3)$$

In the elastic regions, the total strain increment is decomposed as:

$$d\{\varepsilon\} = d\{\varepsilon\}_e + d\{\varepsilon\}_T \qquad (4)$$

Taking the derivative of $\{\varepsilon\}_e = [D]^{-1}\{\sigma\}$ yields:

$$d\{\varepsilon\}_e = \frac{d[D]^{-1}}{dT}\{\sigma\}dT + [D]^{-1}d\{\sigma\} \qquad (5)$$

Taking the derivative of $\{\varepsilon\}_T = \{\alpha\}\{T-T_0\}$ yields:

$$d\{\varepsilon\}_T = \frac{d\{\alpha\}}{dT}\{T-T_0\}dT + \{\alpha\}dT \qquad (6)$$

Substituting equations (5) and (6) into equation (4) yields:

$$d\{\varepsilon\} = \frac{d[D]^{-1}}{dT}\{\sigma\}dT + [D]^{-1}d\{\sigma\} + \frac{d\{\alpha\}}{dT}\{T-T_0\}dT + \{\alpha\}dT$$
$$(7)$$

Multiplying equation (7) by [D] at the two ends and then arranging it yields:

$$d\{\sigma\} = [D](d\{\varepsilon\} - d\{\tilde{\varepsilon}\}_T) \qquad (8)$$

where $d\{\tilde{\varepsilon}\}_T = \left(\{\alpha\} + \frac{d[D]^{-1}}{dT}\{\sigma\} + \frac{d\{\alpha\}}{dT}\{T-T_0\}\right)dT$.

The deduction of equations (1)-(8) can be referred to Rao's works (Rao, 1982).

For every element, substituting equations (2), (3) and (8) into equation (1) yields:

$$\left(\int_V d\{\delta\}^{eT}[B]^T[D]^T[B]d\{\delta^*\}^e dV - \int_V (d\{\tilde{\varepsilon}\}_T)^T[D]^T[B]d\{\delta^*\}^e dV\right)$$
$$- \int_V \{dp\}^T[N]d\{\delta^*\}^e dV - \int_{S_\sigma} \{d\overline{p}\}^T[N]d\{\delta^*\}^e dS - \{dP\}^T[N]d\{\delta^*\}^e = 0$$
$$(9)$$

Transposing and arranging equation (9) yields:

$$[k]^e d\{\delta\}^e = \{dF\}_T^e + \{dF\}_p^e + \{dF\}_{\overline{p}}^e + \{P\}^e \qquad (10)$$

where:

$$[k]^e = [B]^T[D][B],$$
$$\{dF\}_T^e = \int_V [B]^T[D](d\{\tilde{\varepsilon}\}_T)dV,$$
$$\{dF\}_p^e = \int_V [N]^T\{dp\}dV,$$
$$\{dF\}_{\overline{p}}^e = \int_{S_\sigma} [N]^T\{d\overline{p}\}dS,$$
$$\{P\}^e = [N]^T\{dP\}.$$

The global stiffness equation is modeled according to element stiffness equation (10), thereby achieving the corresponding displacements and stresses.

## Numerical calculation and result discussion

Research case

To study the influence of different outlet pressures and rotational speeds on coupling characteristics of valve plate pair, outlet pressure $p = 30$ MPa and speed $n = 12,000$ r/min are selected as the basic case. Other cases studied are as follows: outlet pressure $p = 25$ MPa and speed $n = 9,000$ r/min; outlet pressure $p = 30$ MPa and speed $n = 9,000$ r/min; outlet pressure $p = 32$ MPa and speed $n = 9,000$ r/min; and outlet pressure $p = 30$ MPa and speed $n = 12,000$ r/min.

Thermal-fluid-structure coupling flow

To carry out the coupling analysis for the valve plate pair, a complete set of fast and effective thermal-fluid-structure coupling method is put forward. Its flowchart is shown in Figure 2, and the detailed steps are as follows:

- The calculations of temperature and pressure on the fluid-structure coupling interfaces are accomplished by Fluent, and then the batch processing for the temperature result is accomplished by VC++ and APDL.
- The boundary nodes are extracted on the fluid-structure coupling interfaces by APDL, and temperature of the nodes is interpolated with that of the corresponding nodes in the previous step.
- The transient thermal analysis in the structure is carried out by APDL.
- The equivalent elements are obtained on the fluid-structure coupling interfaces by APDL.

Figure 2 Thermal-fluid-structure coupling flowchart
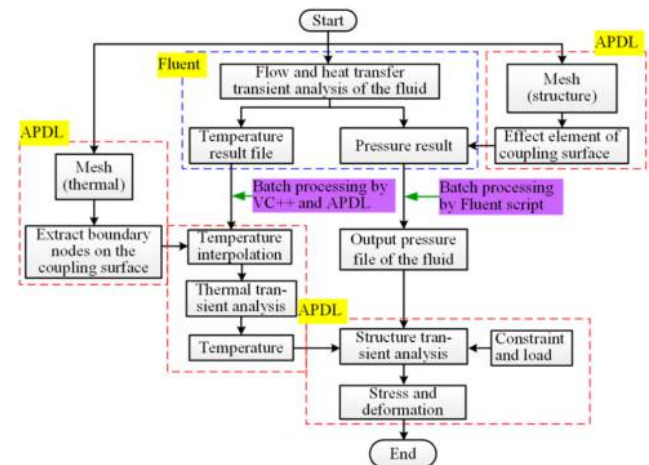
- Pressure of the equivalent elements in the previous step are processed in batches by Fluent script.
- The temperature in the third step and the pressure in the fifth step are loaded into the structure model according to load step, constraints and other loads are added, and the transient structure analysis is accomplished.

In the method, Fluent with user-defined function is applied for fluid calculation, and ANSYS Parametric Design Language is adopted for solid calculation. Therefore, compared with the system coupling module of ANSYS Workbench platform and mesh-based parallel code coupling interface, it has the incomparable advantages, such as no special restrictions for the research objects, flexible solution and other settings and high computation efficiency.

Results and discussion

Fluid field analysis

In the valve plate pair, pressure distribution and temperature distribution of the fluid at 0.125 s are shown in Figures 3 and 4, respectively. In Figure 3, the pressure is higher at regions connected with the oil discharge port; when the fluid in a piston passes through Damping Groove 1, oil pressure instantaneously rises at the location. In Figure 4, the temperature of the entire oil film and part of the piston coupling walls is higher.

Figure 5 shows the pressure distribution of the oil film in the valve plate pair when the pump works for different time. By the analysis for Figure 5, the following conclusions can be drawn:

- The highest pressure of the oil film sometimes exceeds the rated pressure. The value is closely related to the relative position among pistons and damping grooves. For example, when a piston passes through Damping Groove

Figure 3 Pressure distribution of the fluid at 0.125 s in the valve plate pair (30 MPa and 12,000 r/min)
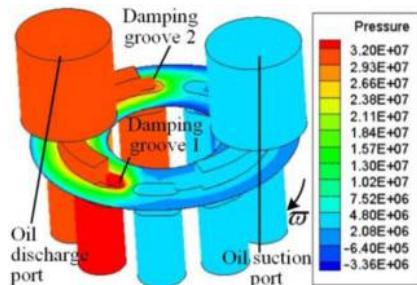
Figure 4 Temperature distribution of the fluid at 0.125 s in the valve plate pair (30 MPa and 12,000 r/min)
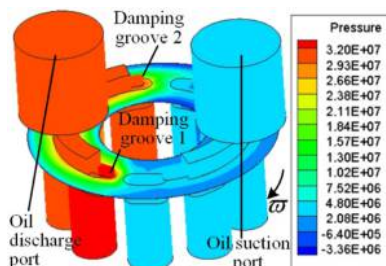
1, because of the outflow oil blocked by the groove, instantaneous pressure ripple occurs.
- When a piston passes through Damping Groove 2, because of high-pressure fluid in the piston chamber at the time, the pressure at part of Transition Region 2 is also high.
- In the oil discharge region, a higher pressure gradient exists along the radial direction.
- Compared to the other oil suction region, at Damping Groove 3, because of the transition from high pressure to low pressure and increasing outflow area, with the rotation of the cylinder block, the pressure greatly varies.
- With the rotation of the cylinder block, the position among cylinder holes and oil discharge port is constantly changing, and high pressure distribution is also slightly different and varies between Damping Grooves 1 and 3.

When the pump works for different time, temperature distributions of the oil film in the valve plate pair are shown in Figure 6. The time history of temperature rise of oil film can be seen in Figure 6.

- At the initial working stage, the temperature near Damping Groove 1 increases sharply because of the damping effect. The temperature is higher near the outer edges of the oil film and the oil discharge port in the oil discharge region and Damping Groove 3.
- With the rotation of the cylinder block, because of the high-temperature oil leaving from the oil discharge region, temperature value and relative high temperature area near Damping Groove 3 even exceed those near Damping Groove 1.
- After a period of time, the temperatures of oil film among adjacent cylinder holes and valve plate gradually increase. But the locations of relatively high temperature region always change with the rotation of the cylinder block. And their numbers also increase from 1 to 3. After that, even in the oil suction region, relative high temperature occurs at the corresponding locations, and their temperatures gradually decrease along the rotation direction of the cylinder block.
- Because of the decreasing pressure and the rotation of the cylinder block, near Damping Groove 2, a heart-shape lower temperature region occurs, whose area gradually shrinks.
- With the increasing operation time, along the rotation direction of the cylinder block, the temperature of oil film continuously increases until the temperature of the entire oil film is almost high.
- After 0.01 s, the temperature distribution shapes are basically stable, i.e. except the locations near oil suction port, oil discharge port and Damping Groove 1, the temperature of oil film is high. Because of the low-temperature oil entering from the oil suction region, the temperature near Damping Groove 1 is relatively lower.

Thermal analysis for the solid

When the pump works until 0.125 s, the temperature distributions of the valve plate and the cylinder block are shown in Figures 7 and 8, respectively. From Figures 7 and 8, it is seen that the high temperature area is larger, and the temperature

Figure 5 Pressure distribution of the oil film in the valve plate pair when the pump works for different time
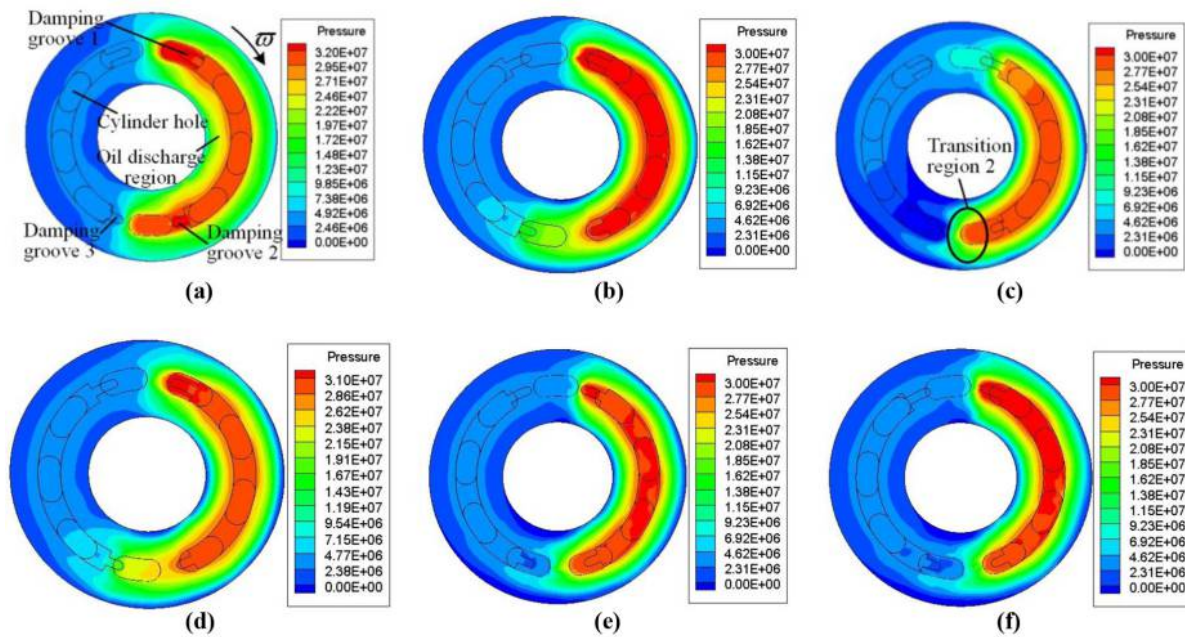
Notes: (a) 0.00025 s; (b) 0.001 s; (c) 0.00125 s; (d) 0.0015 s; (e) 0.005 s; (f) 0.125 s (30 MPa and 12,000 r/min)

Figure 6 Temperature distribution of the oil film in the valve plate pair when the pump works for different time
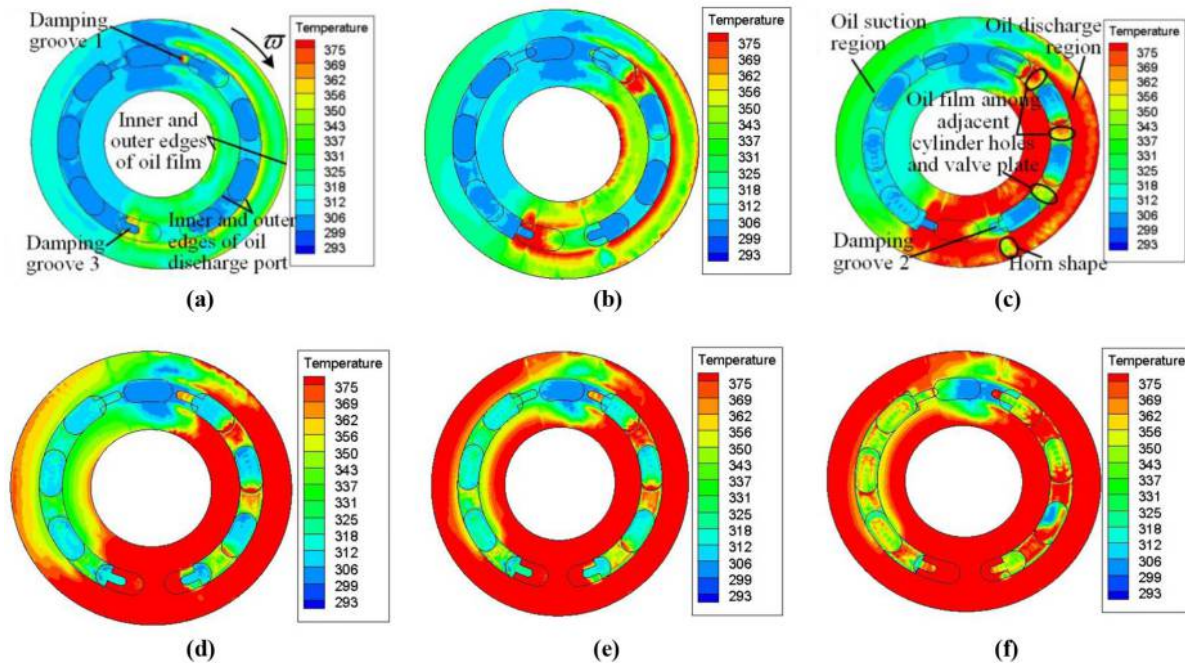

Notes: (a) 0.0005 s; (b) 0.001 s; (c) 0.002 s; (d) 0.005 s; (e) 0.01 s; (f) 0.125 s (30 MPa and 12,000 r/min)

distribution laws of the valve plate and cylinder block are in accordance with those of the oil film, which suggests that the temperature interpolation from the fluid to the structure is right.

Thermal-fluid elastic analysis for the solid

When the pump works until 0.125 s, the von Mises stress distributions of the valve plate and the cylinder block are shown in Figures 9 and 10, respectively. In Figure 9, the higher stress
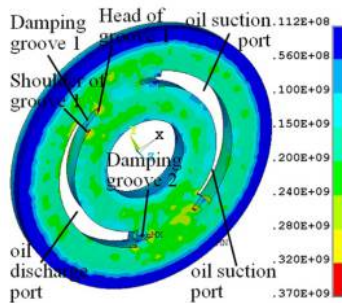
Figure 7 Temperature distribution of the valve plate at 0.125 s (30 MPa and 12,000 r/min)
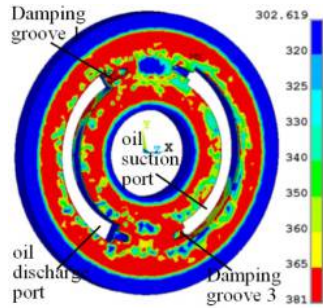
**Figure 8** Temperature distribution of the cylinder block at 0.125 s (30 MPa and 12,000 r/min)
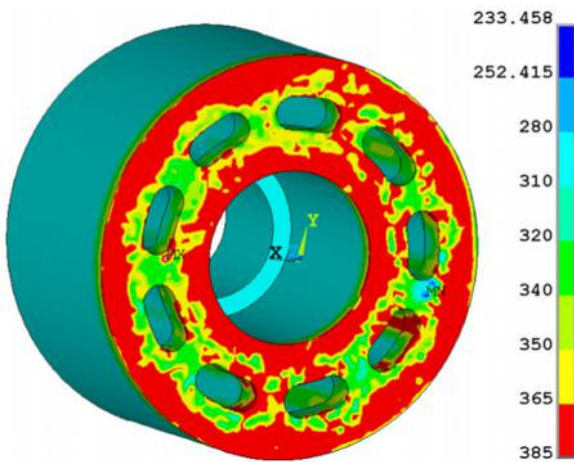
Figure 9 von Mises stress of the valve plate at 0.125 s (30 MPa and 12,000 r/min)

is located at the heads of Damping Grooves 1, 2 and 3, some shoulders of oil discharge port and oil suction port. The maximum stress is over 370 MPa. At this time, it is noted whether the value exceeds the yield limit or the fracture limit of the material. In Figure 10, The maximum stress occurs at the changed cross section of the internal surface of the cylinder block, which also is the dividing line of the constraint. The maximum stress is 157 MPa.

When the pump works until 0.125 s, the axial deformation contours of the valve plate and the cylinder block are shown

Figure 10 von Mises stress of the cylinder block at 0.125 s (30 MPa and 12,000 r/min (30 MPa and 12,000 r/min)
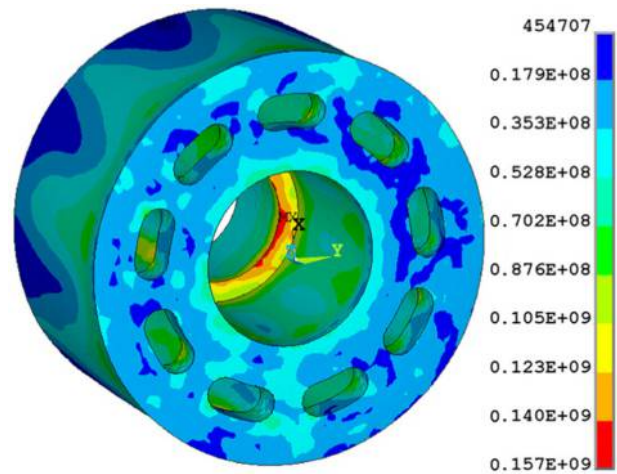

in Figures 11 and 12, respectively. Overall, the axial deformation of the valve plate is smaller than that of the cylinder block. The axial deformation sizes are different at the oil discharge side and the oil suction side of the cylinder block. Between the cylinder block and the valve plate, the clearance increases at the oil discharge side, whereas the clearance changes little or decreases at the oil suction side. The wedge-shaped clearance forms are shown in Figure 13.
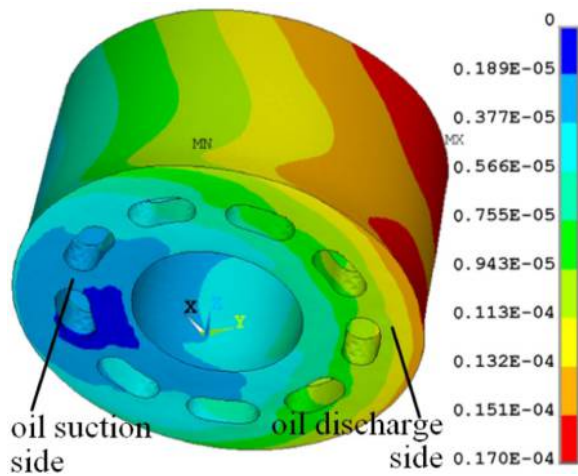
The mating clearance
To investigate the mating clearance changing with time, two pairs of nodes on the front surface of the valve plate and the bottom surface of the cylinder block are selected. Under different rotational speeds and different outlet pressures, at these locations, the curves of the mating clearance change versus time are shown as Figures 14 and 15, respectively. By the analysis for them, the following conclusions are drawn:
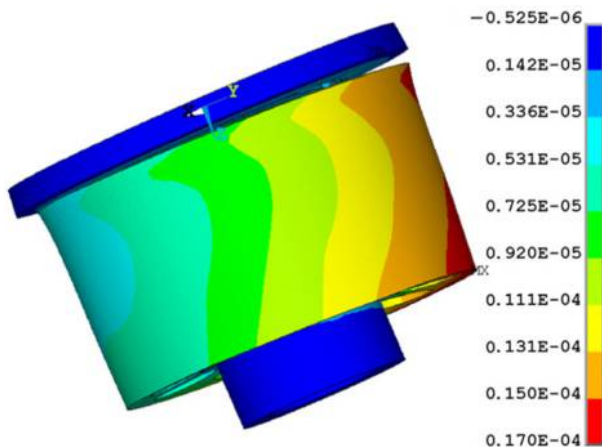
Figure 11 Axial deformation of the valve plate when the pump works until 0.125 s
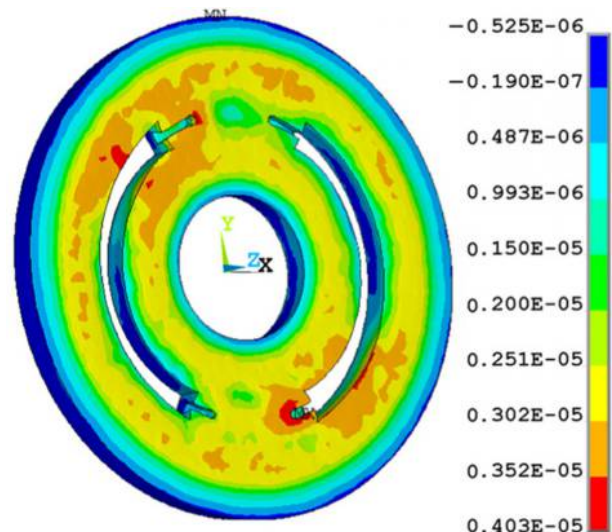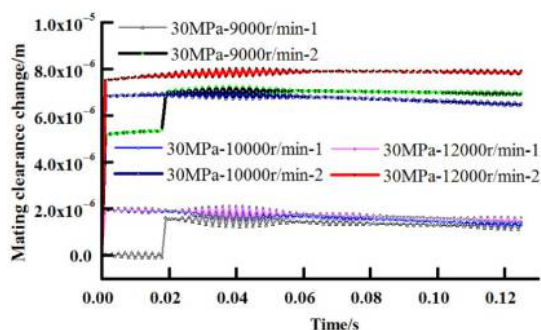
Figure 12 Axial deformation of the cylinder block when the pump works until 0.125 s
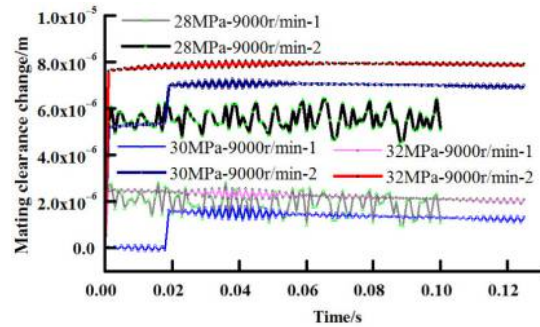

Figure 13 Wedge-shaped clearance between cylinder block and valve plate (×150)

Figure 14 Mating clearance change curves versus time under different rotational speed

Figure 15 Mating clearance change versus time curves under different outlet pressures

- Compared with rotational speed, the influence of outlet pressure on mating clearance is larger, such as 28 and 30MPa (Figure 15).
- Under high pressure and high speed, the mating clearances at the majority places increase, but the increasing sizes are different and sometimes also decrease at some local regions, such as 30 MPa-9,000 r/min, thereby forming wedge-shaped clearance.
- Compared with other cases, under the case of 28 MPa-9,000 r/min, the changing magnitude of mating clearance is larger because the area of low temperature oil is larger under 28 MPa, whereas over 30 MPa, the entire oil film is almost a high temperature region, and the deformation induced by the pressure is also larger.
- In general, the higher the pressure, the higher the speed and the more the clearance increase.
- With the increase in operation time, the overall change trend of mating clearance is from rapidly increasing to smaller amplitude of oscillation.

Conclusions

Using the fast and effective multi-physics field coupling method proposed, the numerical analysis on the valve plate pair is conducted, and the main conclusions are as follows:

- When a piston passes through Damping Groove 1 or 2, instantaneous pressure ripple of the oil film and the fluid in the valve plate pair occurs. The value is closely related to the relative position among pistons and damping grooves. High pressure region of the oil film varies between Damping Grooves 1 and 3.
- After 0.01 s, the temperature distribution shapes of the oil film are basically stable, and the temperature of the entire oil film is almost high.
- Compared with other cases, under the case of 28 MPa-9,000 r/min, the changing magnitude of mating clearance is larger.
- The forming history of wedge-shaped clearance is from rapidly increasing to smaller amplitude of oscillation.

For research on the generated heat and the deformation of an axial piston pump working under high speed, high pressure and wide temperature range, the multi-physics field coupling analysis is an indispensable means and method, and it is extended to more application areas, such as the wet multidisc

brake of a heavy vehicle, the pantograph-catenary system and the vehicle-track system of a high-speed train.

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