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Research on two-dimensional fluid vibration axial transmission path model of axial piston pump[①]

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Abstract

High speed and high pressure can enhance the vibration of axial piston pump. A fluid vibration transmission law of axial piston pump is studied in this paper. According to harmonic response analysis results, a transmission path analysis is used to establish a two-dimensional fluid vibration transmission path model in the vertical plane, which has characteristics of multi excitation sources, multi-path and multi-receptors. Model parameters are obtained by experimental and numerical analysis. Matlab is used to solve the model, and acceleration vibration response of three shells is got. To reduce the effect of mechanical vibration, the surface acceleration of pump is tested under low speed condition. Results show that the model can accurately reveal transmission law of fluid vibration and the accuracy is more than 90%. The research lays a foundation for exploring vibration transmission law and vibration control.

Key words: axial piston pump, fluid vibration, vibration transmission path, vibration response

0 Introduction

High speed and high pressure can significantly improve the power weight ratio of hydraulic system, so it is the main development direction of high-end equipment hydraulic system[1]. However, with the increase of velocity and pressure, the system vibration will be greatly enhanced. That will bring harm to dynamic performance and reliability of the system, and make the vibration control more difficult. The main reasons for this phenomenon are as follows. High speed and high pressure can greatly increase pressure in unit volume oil. When the mechanical energy is converted to hydraulic energy, the coupling effect of solid-fluid-heat-sound is enhanced. The alternating load and high-frequency impact caused by high speed and high pressure will make the vibration more complex and the vibration control more difficult. The hydraulic pump is one of the main vibration sources, and its high speed movement will produce complex mechanical vibration and fluid vibration[2].

The fluid vibration is produced by the fluid-structure interaction between the fluid in plunger chamber and the plunger or the cylinder block when the plunger moves back and forth. Early researches on the fluid vibration mechanism started in the 1960s – 1970s, and has attracted wide attention since then. In the early model, the effect of nonlinear factors was ignored, such as the leakage, backward flow and multi-field interaction. The model was imprecise. With the increasing of speed and pressure, the fluid vibration of pump is characterized by wide frequency range and strong intensity. In recent years, a series of researches have been carried out to improve the model precision. Considering the movement flow, leakage flow of three friction pairs, backward flow and multi-field interaction, a relatively complete flow pulsation model was established. Moreover, the CASPAR software was developed to calculate flow pulsation accurately[3,4]. Considering temperature field, pressure model and oil film characteristics, a virtual prototype simulation platform of pump was built by Yang Huayong to obtain fluid vibration information. The unsteady flow field in the pump was numerically simulated based on the computational fluid dynamics (CFD) method, obtaining the flow pulsation at the pump port and the pressure distribution at the oil film[5,6].

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The axial piston pump is one of the most complex hydraulic components. The fluid vibration is transferred to the pump shells along the parts of the pump and the transmission law is quite complicated. In recent years, research on vibration transmission of pump is mainly concerned with the vibration response of key parts. For this method, mechanical vibration and fluid vibration are considered as excitation, and the constraints of the structural parts are taken into account. So, it is a good contribution to explore the vibration response characteristics and the vibration transmission law. Ozdemir et al.[7], Kim and Ivantysynova[8], Zhang et al.[9], Chao et al.[10], Li et al.[11] and Quan et al.[12] have carried out lots of researches in this respect. The vibration transmission path method was used to analyze the vibration transmission law from the system point of view, which was put forward for the first time. The mechanical vibration transmission path model was established and the main transmission path was obtained[13].

Aiming at the fluid vibration generated by axial piston pump, the dynamic transmission characteristics are analyzed. The two-dimensional transmission path model of fluid vibration is established by means of transmission path analysis. The transmission law of fluid vibration is studied. The results will lay a foundation for the investigation of the fluid vibration transmission law and the realization of the precise vibration control.

1 The transmission mechanism of fluid vibration

The flow pulsation is formed by the high and low pressure conversion in the distribution region. When it is coupled with the structural parts of the pump, the fluid vibration will be produced. The transmission process of fluid vibration has the characteristics of multi-excitation points, multi-path and multi-receptors. Its transmission rule is influenced by the interaction relationship between parts, the dynamic topological structure and the pump working condition. Therefore, the transmission mechanism is very complex.

The 25PCY14-1B axial piston pump is used as the research object. The pressure is 31.5 MPa and the displacement is 25 mL/r. The fluid vibration transmission path model is established based on the following assumptions. The pump operates at a low speed of 1 500 r/min, so the mechanical vibration is ignored. The excitation force is produced in the high pressure region of the plunger chamber. The inertia effect of oil is ignored. Only the rotation and movement of plunger are considered. The mass of the connecting elements is equivalent to the vibration bodies. The oil film has the characteristics of stiffness and damping. The axial damping of the bearing is neglected. The three fluid vibration transmission paths are shown in Fig. 1.

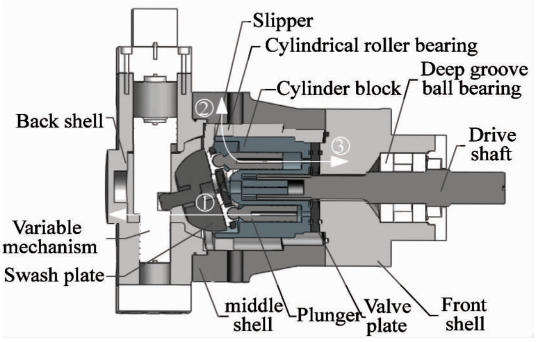

Fig. 1 Fluid vibration transmission paths of pump

The three transmission paths are described separately as follows.

Path 1: the plunger →slipper assembly →swash plate →variable mechanism →the back shell;

Path 2: the plunger →cylinder block and drive shaft →cylindrical roller bearing →the middle shell;

Path 3: the plunger →cylinder block and drive shaft →the front shell.

2 Dynamic model of fluid vibration transmission path

2.1 Mechanical model of fluid vibration transmission path

The axial piston pump is connected with the motor output by the bell-shaped hood. The main shaft and the bell-shaped hood form a cantilever beam structure. The preliminary research results show that the first order vibration model of the pump is in the vertical plane[14]. Therefore, the fluid vibration transmission law under the first order vibration mode is focused. The parts and their interactions are simplified to mass, stiffness and damping. In the vertical plane, the mechanics model of fluid vibration two-dimensional transmission path is shown in Fig. 2.

In Fig. 2, F is the excitation force of the oil in the plunger chamber; x_b, x_m, x_f, x_{vm}, x_{sp}, x_p, x_o and x_{cb} respectively mean the vibration displacement of the back shell, the middle shell, the front shell, the variable mechanism, the swash plate, the slipper assembly, the oil in plunger chamber, the cylinder block and drive shaft, and valve plate under the excitation force F_w; m_b, m_m, m_f, m_{vm}, m_{sp}, m_p, m_o and m_{cb} are respectively the real mass of the back shell, the middle shell, the front shell, the variable mechanism, the swash plate, the slipper, the plunger, the oil in the plunger

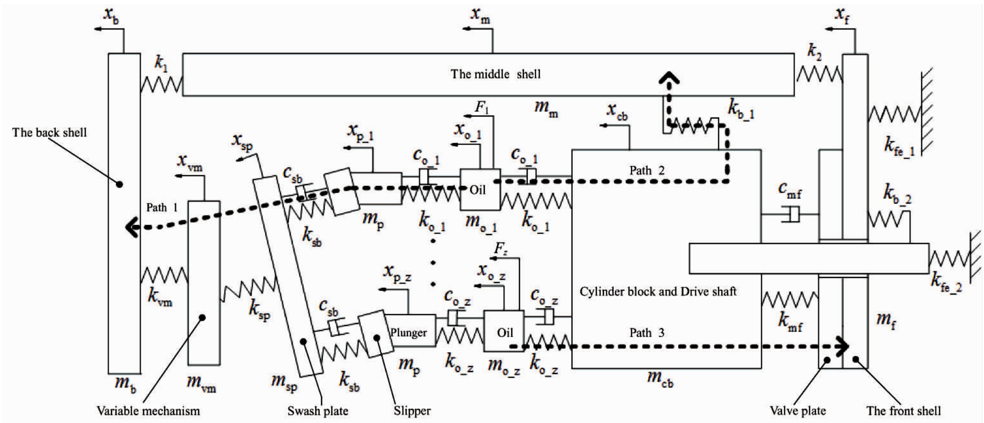

Fig. 2 The mechanical model of fluid vibration transmission path of the axial piston pump

chamber, cylinder block and drive shaft; k_1 is the axial contact stiffness between the back shell and the middle shell; k_2 is the axial contact stiffness between the middle shell and the front shell; k_{b-1} is the axial stiffness of the cylindrical roller bearing; k_{b-2} is the axial stiffness of the deep groove ball bearing; k_{vm} is the axial stiffness between the back shell and the variable mechanism; k_{sp} is the axial stiffness between the variable mechanism and the swash plate; k_{sb} and c_{sb} mean the stiffness and damping of the oil film supported by the slipper friction pair; k_{o-w} and c_{o-w} mean the stiffness and damping of the oil in plunger chamber; k_{mf} and c_{mf} are the stiffness and damping of the oil film supported by the port platefriction pair; k_{fe-1} is the axial stiffness between the front shell and the fixed end; k_{fe-2} is the axial stiffness between the cylinder block and drive shaft, and the coupling.

2.2 Mathematical model of fluid vibration transmission path

The energy conversion relation of each particle in analytical mechanics method is used instead of mechanical relationship in the classical Newtonian mechanics, which will avoid many equations brought by the constraint condition and simplify the solving process. The analysis of the vibration transmission path focuses mainly on the transmission law of vibration energy in the system. So the mathematical model of the fluid vibration transmission path is established by analytical mechanics method.

In the analytical mechanics method, the elementary work $\mathrm{d}W$ made by the external force is equal to the sum of the kinetic energy, potential energy and dissipation energy of the system, which can be expressed as

$$\mathrm{d}W = \mathrm{d}(T + U + D) \tag{1}$$

The mechanical vibration system is defined in the generalized coordinate systems, which can be described as $\boldsymbol{q}_i = [q_1, q_2, \cdots, q_z]^{\mathrm{T}}$. The generalized force of the system is $\boldsymbol{\Omega}_i = [\Omega_1, \Omega_2, \cdots, \Omega_z]^{\mathrm{T}}$. The elementary work of the mechanical vibration system is $\mathrm{d}W = \boldsymbol{\Omega}^{\mathrm{T}}\mathrm{d}q = \sum_{j=1}^{z} \Omega_j \mathrm{d}q_j$. The kinetic energy function is $T = T(q_1, q_2, \cdots, q_z, \dot{q}_1, \dot{q}_2, \cdots, \dot{q}_z)$; the potential energy function is $U = U(q_1, q_2, \cdots, q_z)$; and the dissipation energy function is $D = D(\dot{q}_1, \dot{q}_2, \cdots, \dot{q}_z)$. Under this definition, the relationships can be described as

$$\mathrm{d}T = \sum_{j=1}^{z} \frac{\partial T}{\partial q_j}\mathrm{d}q_j + \sum_{j=1}^{z} \frac{\partial T}{\partial \dot{q}_j}\mathrm{d}\dot{q}_j \tag{2}$$

$$\mathrm{d}U = \sum_{j=1}^{z} \frac{\partial U}{\partial q_j}\mathrm{d}q_j \tag{3}$$

$$\mathrm{d}D = \sum_{j=1}^{z} \frac{\partial D}{\partial \dot{q}_j}\mathrm{d}\dot{q}_j \tag{4}$$

In the fluid vibration transmission path, the generalized fluid excitation force is produced by flow pressure pulsation of plunger and cylinder block. The generalized fluid excitation force is defined as F, which can be expressed as $\boldsymbol{\Omega}_0 = F$.

Eqs(2), (3) and (4) are substituted into Eq. (1), the energy relationship under the generalized coordinate can be obtained after simplifying.

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial T}{\partial \dot{q}}\right) - \frac{\partial T}{\partial q} + \frac{\partial U}{\partial q} + \frac{\partial D}{\partial \dot{q}} = \boldsymbol{\Omega} \tag{5}$$

The kinetic energy, potential energy, dissipation energy and generalized force of each component in the axial piston pump are substituted into Eq. (5) respectively, and the Lagrange differential equation of each vibration body can be obtained.

For the seven-plunger axial piston pump, the number of plunger in the drainage area is alternating between 3 and 4 in the period of 25.7 °. When the number is 3, there are 16 vibration bodies, which are the back shell, the front shell, the middle shell, the variable mechanism, the swash plate, three plungers in the drainage area, three slippers, three plunger chambers, a port plate, a cylinder block and drive shaft. When the number is 4, vibration bodies are increased by 3, which are the plunger in the drainage area, the slipper and the plunger chambers. The differential equations of fluid vibration transmission are as follows.

$$
\left\{
\begin{aligned}
& m_b\ddot{x}_b - k_{vm}(x_{vm} - x_b) + k_1(x_b - x_m) = 0 \\
& m_m\ddot{x}_m - k_1(x_b - x_m) - k_{b-1}(x_{cb} - x_m) + k_2(x_m - x_f) = 0 \\
& m_f\ddot{x}_f - k_{b-2}(x_{cb} - x_f) + k_{fe-1}x_f - c_{mf}(\dot{x}_{cb} - \dot{x}_f) - k_{mf}(x_{cb} - x_f) = 0 \\
& m_{vm}\ddot{x}_{vm} - k_{vm}(x_b - x_{vm}) - k_{sp}\cos\beta(x_{sp}\cos\beta - x_{vm}) = 0 \\
& m_{sp}\ddot{x}_{sp} - \sum_{w=1}^{7} c_{sb}(\dot{x}_{p-w}\cos\beta - \dot{x}_{sp}) + k_{sp}(x_{sp} - x_{vm}\cos\beta) - \sum_{w=1}^{7} k_{sb}(x_{p-w}\cos\beta - x_{sp}) = 0 \\
& m_p x_{p-1} - c_{o-1}(\dot{x}_{o-1} - \dot{x}_{p-1}) - k_{o-1}(x_{o-1} - x_{p-1}) + c_{sb}\cos\beta(\dot{x}_{p-1} - \dot{x}_{sp}\cos\beta) + k_{sb}\cos\beta(x_{p-1} - x_{sp}\cos\beta) = 0 \\
& \vdots \\
& m_p x_{p-7} - c_{o-7}(\dot{x}_{o-7} - \dot{x}_{p-7}) - k_{o-7}(x_{o-7} - x_{p-7}) + c_{sb}\cos\beta(\dot{x}_{p-7} - \dot{x}_{sp}\cos\beta) + k_{sb}\cos\beta(x_{p-7} - x_{sp}\cos\beta) = 0 \\
& m_{o-1}\ddot{x}_{o-1} + c_{o\_1}(\dot{x}_{o-1} - \dot{x}_{p-1}) + c_{o-1}(\dot{x}_{o-1} - \dot{x}_{cb}) + k_{o-1}(x_{o-1} - x_{p-1}) + k_{o-1}(x_{o-1} - x_{cb}) = F_1 \\
& \vdots \\
& m_{o-7}\ddot{x}_{o-7} + c_{o-7}(\dot{x}_{o-7} - \dot{x}_{p-7}) + c_{o-7}(\dot{x}_{o-} - \dot{x}_{cb}) + k_{o-7}(x_{o-7} - x_{p-7}) + k_{o-7}(x_{o-7} - x_{cb}) = F_7 \\
& m_{cb}\ddot{x}_{cb} - \sum_{w=1}^{7} c_{o-w}(\dot{x}_{o-w} - \dot{x}_{cb}) + c_{mf}(\dot{x}_{cb} - \dot{x}_f) - \sum_{w=1}^{7} k_{o-w}(x_{o-w} - x_{cb}) - k_{b-1}(x_m - x_{cb}) - k_{b-2}(x_f - x_{cb}) \\
& \qquad + k_{mf}(x_{cb} - x_f) + k_{fe-2}x_{cb} = 0
\end{aligned}
\right.
\tag{6}
$$

In Eq. (6), β is the inclining angle of swash plate. According to the number of plunger in the drainage area, main vibratory components are identified and the corresponding vibration differential equations are listed.

2.3 Determination of parameters in fluid vibration transmission path model

The main parameters in the dynamics model of fluid vibration transmission path include stiffness and damping of the interaction between parts, and the fluid excitation force generated by the fluid vibration. The mass parameters can be obtained directly from the material density and three-dimensional model. The stiffness, damping and excitation force can be determined by mathematical calculation or finite element analysis. The material properties of some components are shown in Table 1.

Table 1 The material properties of components in the pump

Component name	Material	Elastic modulus (Pa)	Poisson ratio	Density (kg/m^3)
Front shell	HT300	1.43×10^{11}	0.270	7 300
Middle shell	HT200	1.48×10^{11}	0.310	7 200
Back shell	HT300	1.43×10^{11}	0.270	7 300
Bolt	Q235A	2.11×10^{11}	0.288	7 860
Bearing	GCr15SiMn	2.06×10^{11}	0.300	7 820
Valve plate	38CrMoAl	2.12×10^{11}	0.286	7 850
Variable mechanism	40Cr	2.11×10^{11}	0.277	7 870

2.3.1 Determination of stiffness and damping parameters

(1) The axial stiffness of deep groove ball bearing

The deep groove ball bearing is located in the front shell to support the pump main shaft and the front shell. The outer ring and the shell are connected under an interference fit. In this study, the axial movement of bearings is mainly considered. The inner and outer ring will produce axial relative displacement due to rolling deformation. The outer ring is set as the fixed support, and the axial static load is applied to the tangential direction of the inner ring. The relative displacements of the inner ring and outer ring under different static loads are analyzed. When the axial static load is 100 N, the static displacement of deep groove ball bearing is shown in Fig. 3.

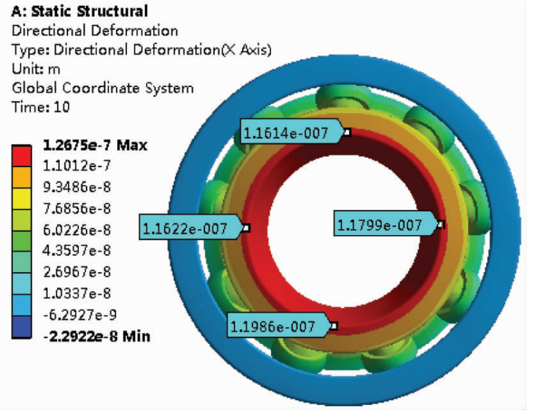

Fig. 3 Static displacement of deep groove ball bearing

The axial relative displacement $\triangle x$ under different static loads F is shown in Table 2.

Table 2 The relative displacement of inner ring and outer ring under different static loads

F(N)	100	200	300	400	500
$\triangle x \times 10^{-6}$(m)	0.82	1.65	2.51	3.24	4.18

Based on the calculation results of Table 2, the relationship between the static load and the axial relative displacement is obtained by curve fitting. The axial contact stiffness of the rolling and the ring is $\dfrac{F}{\Delta x} = 1.2 \times 10^8$ N/m. Considering the influence of various factors on the contact stiffness and oil stiffness, the axial stiffness of the deep groove ball bearing is approximately $k_{b,2} = 8.0 \times 10^6$ N/m using the series stiffness calculation method.

Similarly, the axial stiffness of the cylindrical roller bearing located in the middle shell is $k_{b,1} = 4.0 \times 10^7$ N/m.

(2) The axial contact stiffness between shells

Three shells of the pump are connected by bolts. The axial contact stiffness between shells is composed of the stiffness of bolt and the axial stiffness of the shell joint surface. The contact surface type between shells is set to 'No Separation', and the remaining contact surface type is set to 'Bonded'. The clamped constraint is set to the bolt end face.

(3) Axial stiffness between the back shell and the variable mechanism

The contact surface type between the back shell and the variable mechanism is set to 'Rough' and the clamped constraint is applied to the back shell.

(4) Axial stiffness between the variable mechanism and the swash plate

The contact surface type between the variable mechanism and the swash plate is 'Bonded'. The clamped constraint is applied to variable mechanism.

(5) Stiffness and damping of the supporting oil film in the slipper friction pair

Under the assumption, the flow field of the supporting oil film in the slipper friction pair can be considered as a parallel disc slot flow field. According to Ref. [15], the stiffness of the oil film is

$$k_{sb} = -\frac{\partial F_{sl}}{\partial h} = \frac{576\pi(r_2^2 - r_1^2)d_d^4 l_d h^2}{\left(3\ln(\frac{r_2}{r_1})d_d^4 + 64 l_d h^3\right)^2}p \qquad (7)$$

where, r_1 is the inner radius of the bottom sealing belt of the slipper, r_2 is the outer radius of the bottom sealing belt of the slipper, l_d is the length of damping hole in the slipper, d_d is the diameter of damping hole, h is the thickness of the supporting oil film and p is the pressure in plunger chamber.

The damping of supporting oil film is

$$c_{sb} = \frac{3\pi\mu}{2h^3}(r_2^2 - r_1^2)^2 \qquad (8)$$

here, μ is the dynamic viscosity of the oil.

(6) Stiffness and damping of the oil in plunger chamber

According to Ref. [15], the stiffness of oil in plunger chamber can be expressed as

$$k_o = \frac{\mathrm{d}F}{\mathrm{d}x} = E\frac{A^2}{V} \qquad (9)$$

The damping of the oil in plunger chamber is caused by the oil leakage from the annular gap between the plunger and the cylinder block. The damping coefficient of the oil in plunger chamber can be expressed as

$$c_o = \frac{3\pi\mu l_p d^3 \alpha_L}{4h_p^3} \qquad (10)$$

In Eq. (10), d is the plunger diameter, h_p is the thickness of the oil in plunger chamber, l_p is the plunger length and α_L is the flow leakage coefficient of annular gap.

(7) Stiffness and damping of the oil film supported by the port plate friction pair

According to Ref. [16], the comprehensive stiffness of the oil film supported by the port plate friction pair is[15]

$$k_{mf} = \sqrt{k_m^2 + k_a^2} \qquad (11)$$

Where, k_m is the mean of oil film stiffness and k_a is the amplitude of oil film stiffness.

$$k_m = \frac{3}{h_m(1/p_s - f_1/Q_s)} \qquad (12)$$

$$k_a = \frac{2\eta_2}{A_{n1}t_n + A_{f1}t_f} \tag{13}$$

Where, h_m is the thickness of the oil film supported by the port plate friction pair, Q_s is the total oil of the port plate friction pair, t_n is the time to supply oil and t_f is the time to stop oil.

The damping of the oil film supported by the port plate friction pair is described as

$$c_{mf} = \frac{k_{mf}h_m}{Q_{\text{total}}} \tag{14}$$

In Eq. (14), k_{mf} is the polar radius of any point in the sealing strip flow fieldand, Q_{total} is the total oil leakage flow at the gap between the inner and outer sealing strip.

$$\begin{aligned} Q_{\text{total}} &= |Q_{in}| + |Q_{\text{out}}| \\ &= \frac{\alpha h_m^3}{6\mu c_e}\frac{p_s - p_0}{\ln(R_2/R_1)} + \frac{\alpha h_m^3}{6\mu c_e}\frac{p_s - p_0}{\ln(\frac{R_4}{R_3})} \end{aligned} \tag{15}$$

Among them, α is the angle of waist groove, R_1 and R_2 mean radius of the inner sealing strip, R_3 and R_4 are radius of the outer sealing strip, c_e is the flow correction coefficient, p_s and p_0 are the boundary pressure of the sealing strip.

In summary, by finite element analysis or theoretical calculation, the stiffness and damping parameters in the model are shown in Table 3.

Table 3 The stiffness and damping parameters in the model

Parameters	Value	Parameters	Value
$k_{b,2}$ (N/m)	8.0×10^6	$k_{b,1}$ (N/m)	4.0×10^7
k_1 (N/m)	1.5×10^9	k_2 (N/m)	1.5×10^9
k_{vm} (N/m)	4.0×10^7	k_{sp} (N/m)	7.58×10^8
k_{sb} (N/m)	4.7×10^5	c_{sb} (N·s/m)	9.5×10^4
k_o (N/m)	9.4×10^6	c_o (N·s/m)	300
k_{mt} (N/m)	1.1×10^7	c_{mt} (N·s/m)	1.0×10^3

2.3.2 The excitation force of fluid vibration

The main reasons for fluid vibration of axial piston pump include flow pressure pulsation, oil intrusion and cavitation. The fluid vibration caused by flow pressure pulsation and oil intrusion is studied.

The plunger chamber is regarded as the control volume. The volume of plunger chamber changes periodically with the plunger movement. So, the oil pressure in plunger chamber is decreased periodically. Eq. (16) is a mathematical model for the pressure gradient change in the plunger chamber.

$$\frac{dP_f}{dt} = \frac{K_e}{V_p}(q_n - q_t - q_l - \frac{dV_p}{dt}) \tag{16}$$

In the equation, P_f is the oil pressure in plunger chamber, K_e is the oil bulk elastic modulus, V_p is the oil volume in plunger chamber, q_n is the flow flowing into the plunger chamber through the oil sucking waist groove, q_t is the flow flowing out of the plunger chamber through the oil drain waist groove, q_l is the leakage flow of the plunger chamber. According to the flow formula of thin wall orifice, q_n and q_t can be expressed as

$$q_n = CA_{lp}\sqrt{\frac{2|p_f - p_l|}{\rho}} \cdot \text{sign}(p_l - p_f) \tag{17}$$

$$q_t = CA_{hp}\sqrt{\frac{2|p_h - p_f|}{\rho}} \cdot \text{sign}(p_f - p_h) \tag{18}$$

The leakage flow includes the leakage of the plunger friction pair q_{SK}, the leakage of the slipper friction pair q_{SG} and the leakage of the port plate friction pair q_{SB}. So the relationship can be described as $q_l = q_{SK} + q_{SG} + q_{SB}$. According to the structure of plunger friction pair and the leakage theory of eccentric annular gap, q_{SK} is

$$q_{SK} = \frac{\pi d\delta_1^3}{12\mu l_1}(1 + 1.5\varepsilon^2)(p_f - p_0) - \frac{\pi d\delta_1 v_p}{2} \tag{19}$$

According to the laminar flow theory for parallel disk gap, q_{SG} and q_{SB} are described as

$$q_{SG} = \frac{\pi d_d^4\delta_2^3}{\mu[6d_d^4 In(r_2/r_1) + 128\delta_2^3 l_d]}(p_f - p_0) \tag{20}$$

$$q_{SB} = \frac{\alpha_f\delta_3^3}{12\mu}\left[\frac{1}{In(R_2/R_1)} + \frac{1}{In(R_4/R_3)}\right](p_f - p_0) \tag{21}$$

The L-HM46# wear-resistant mineral hydraulic oil is used as the medium. When the operating temperature is 40 ℃, the elastic modulus of oil is 1.69×10^9 Pa. According to Ref. [17], Matlab software is applied to analyze the oil pressure in single plunger chamber. The pressure change curve of single plunger chamber is shown in Fig. 4.

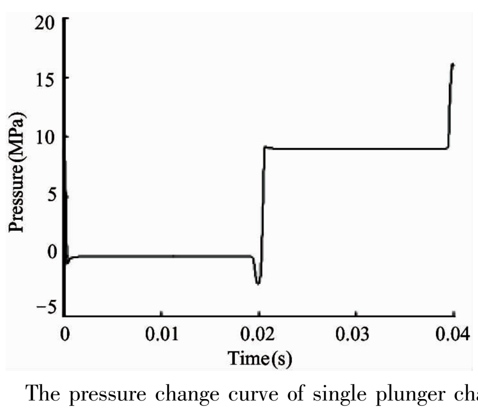

Fig. 4 The pressure change curve of single plunger chamber

3 Solution of mathematical model of fluid vibration transmission path

The fluid vibration of pump has periodic characteristic. The mathematical model of the fluid vibration transmission path is the nonlinear partial differential equations. The fluid vibration transmission law can be obtained by solving the equations.

The Newton iteration method is usually used to solve the equations, but the initial value should be given. However, the real solution of the equations is unpredictable. So, the Runge-kutta method is used, which has higher precision and lower requirements for initial value. According to Eq. (6), the differential equations are defined and piston. m function is obtained. The fluid vibration transmission path model is solved in Matlab software, and the programming flow chart is shown in Fig. 5.

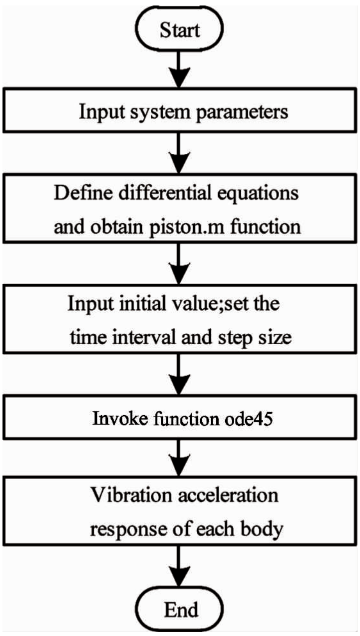

Fig. 5 Theprogramming flow chartfor vibration system

To reduce the effect of mechanical vibration, the pressure is 3 MPa; the rotational speed is 1500 r/min and the frequency of fluid vibration excitation force is 175 Hz. Then the two-dimensional transmission path model of fluid vibration is solved. Fig. 6, Fig. 7 and Fig. 8 are the vibration acceleration frequency domain response curves of three shells respectively.

From Fig. 6 to Fig. 8, the conclusions are as follows.

By vibration amplitude analysis, the back shell has largest amplitude; the middle shell is next and the front shell has the smallest amplitude.

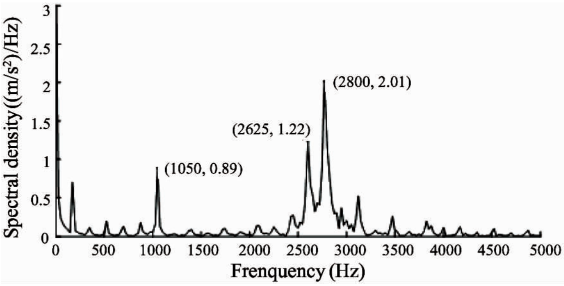

Fig. 6 The vibration acceleration curve of the front shell

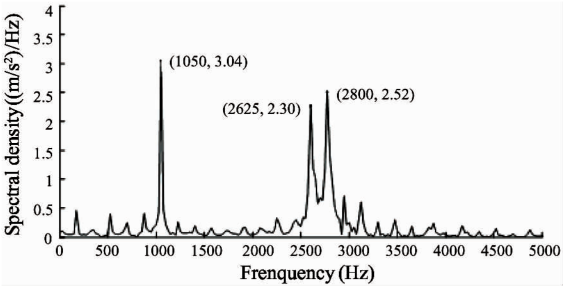

Fig. 7 The vibration acceleration curve of the middle shell

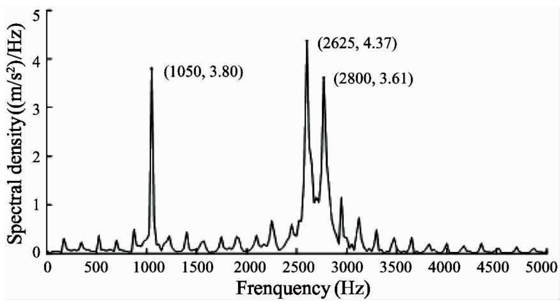

Fig. 8 The vibration acceleration curve of the back shell

The base frequency of the vibration response is 175 Hz. The frequency of other order harmonics is consistent with the excitation frequency.

The first-order harmonic frequency of the vibration response is 1 050 Hz; the second-order harmonic frequency is 2 625 Hz and the third-order harmonic frequency is 2 800 Hz.

The frequency threshold of maximum vibration acceleration peak is 2 500 – 3 000 Hz with maximum power density.

4 Experimental research

The vibration test bench is shown in Fig. 9. The front shell is fixedly connected with bell-shaped hood and the back shell is hanging to form the whole structure of the cantilever beam. The acceleration sensors are placed on the strongest vibration position of three shells. The rotational speed is 1 500 r/min and the pressure is 3 MPa. The sampling frequency is 10 Hz. The acceleration data of three shells are obtained.

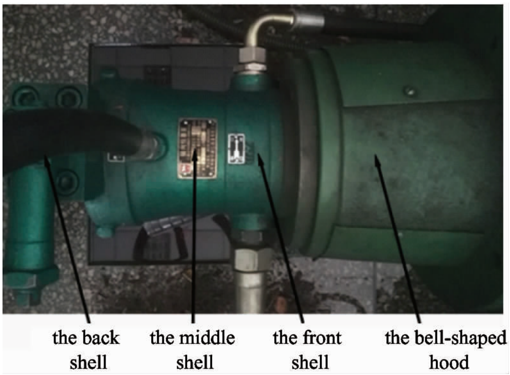

Fig. 9 The vibration test bench for axial piston pump

The vibration acceleration frequency domain response curves of three shells obtained by Fourier transform of sampled vibration data are shown in Fig. 10, Fig. 11 and Fig. 12, respectively.

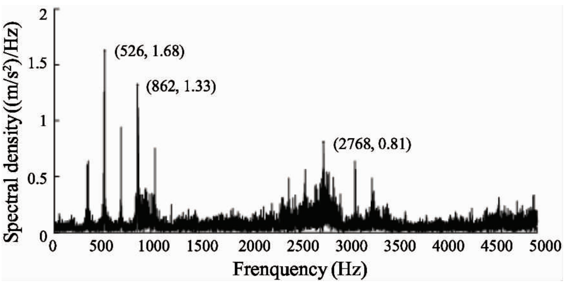

Fig. 10 The vibration acceleration curve of the front shell

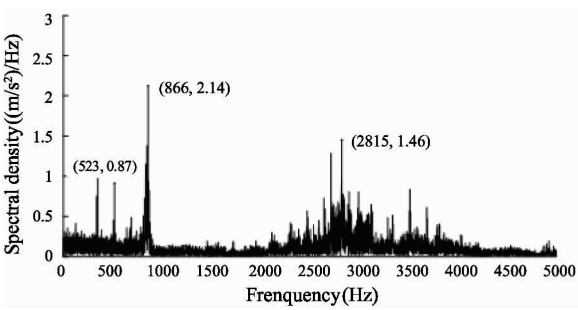

Fig. 11 The vibration acceleration curve of the middle shell

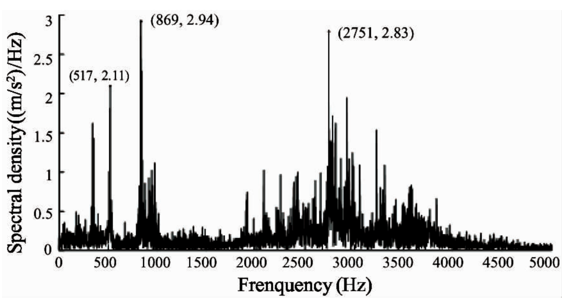

Fig. 12 The vibration acceleration curve of the back shell

From Fig. 10 to Fig. 12, the conclusions are as follows.

Through the vibration amplitude analysis, the back shell has the largest amplitude; the middle shell is next and the front shell has smallest amplitude.

The base frequency of the vibration response is 175 Hz. The frequency of other order harmonics is consistent with the excitation frequency.

5 Conclusions

The axial piston pump is one of the main vibration sources in hydraulic system, and it is the most complex hydraulic component. The fluid vibration transmission law of axial piston pump is analyzed in this work. The mathematical model of fluid vibration two-dimensional transmission path is established by means of transmission path method. The analytical solution of the model is obtained by Matlab programming. The results of theoretical analysis are verified by experiment. The conclusions are as follows.

When the fluid vibration in the plunger chamber is transmitted to the pump shells along the parts, there are three transmission paths. The main transmission path is 'Path 1: the plunger → slipper assembly → swash plate → variable mechanism → the back shell'.

In the future, the time-varying nonlinear parameters and the simultaneous interaction of multi excitation sources will be considered to further improve the model and obtain better simulation results.

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