Research on the Distribution Characteristic of the Double Acting Axial Piston Pump

Professor PhD. Long Quan

PhD. Jiahai Huang

Associate Professor Bin Li

Senior Engineer Jinsheng Wang

Institute of Macaronis Engineering, Taiyuan University of Technology, 53# Xikuang Street
030024 Taiyuan, P.R.China, E-mail: quanlong@tyut.edu.cn

Abstract

In order to use only one hydraulic pump as two independent power source, to control the differential cylinder with only one pump, the scheme of double acting axial piston pump is put forward. There are two types of valve plate of the new double acting axial piston pump. One is asymmetrical structure with three flowing distributing windows and the other one is symmetrical structure with four flowing distributing windows. In paper, the structure and parameters are deteremined for prototype by digital simulation. And then two prototypes are tested. Although the results of the theoretical analysis are similar, the testing results show that the pressure fluctuation and the noise level of the symmetrical double acting axial piston pump with four flow distributing windows are much lower than the asymmetrical one. The research has laid a theoretical foundation for further promoting the using of the new double acting axial piston pump.

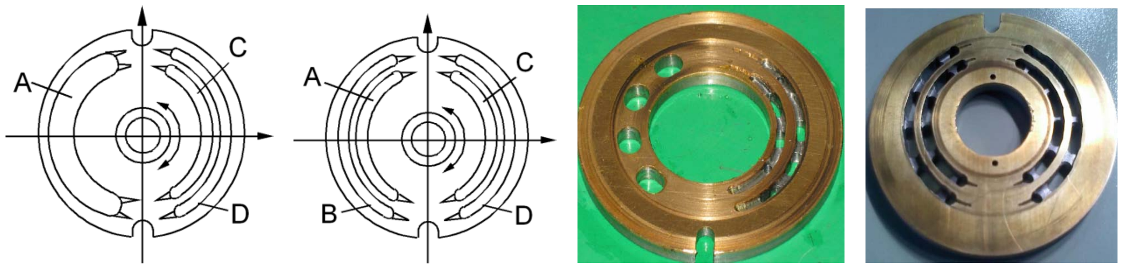
KEYWORDS: double-action axial piston pump, control loop, pressure fluctuation, noise level

1. Introduction

The electric hydrostatic actuator adopts the axial piston pump with two flow distributing windows to control the symmetrical double-rod hydraulic cylinders based on the close circuit principle /1-3/. It has some disadvantages as follows: a larger axial space is needed; the use of series redundant configuration will lengthen the axial dimension; the use of differential cylinder may reduce the axial size and weight, but the two chambers of a single-rod hydraulic actuator have different areas, so the flow needed by two chambers are different, and some supplementary measures must be taken to balance this flow difference, such as using a hydraulic transformer to compensate the asymmetric volume flow of the single rod hydraulic cylinder /4/, using pilot-operated check valve or solenoid valve to balance the flow/5-6/, and using two hydraulic pumps, etc. /7/. These schemes mentioned above not only increase the complexity of the hydraulic system but also decrease the system energy efficiency because of the introduction of control valve. Thus, through changing the valve plate and cylinder block structure of the existing axial piston pump, a double-acted axial piston pump that can balance the asymmetric flow of a differential cylinder and can act as two independent power sources is suggested.

2. Double-acting flow distributing principle

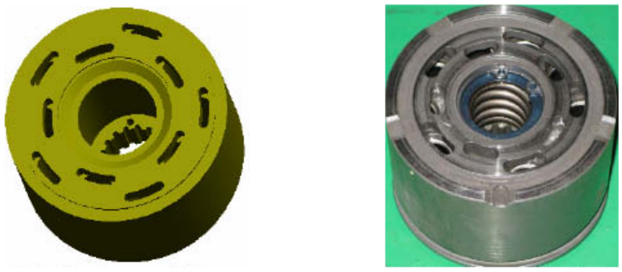
We know that double-acting vane pump has four flow distributing windows and every circle pump suction and discharge twice. By referring to the double-acting vane pump working principle, it can be extended to the axial piston pump, that make the axial piston pump sucking and discharging oil twice during one circle. The valve plate of double-acting axial piston pump has two configurations, one is the asymmetric structure equipped with three flow distributing windows, and the other is the symmetric structure equipped with four flow distributing windows. The structural principles and the prototypes of two newly designed valve plates are shown in **figure 1**.



(a) Valve plate with 3 ports (b) Valve plate with 4 ports (c) Prototype photo with 3 ports (d) Prototype photo with 4 ports

Figure 1: Valve plate structure and photos of double-acting pump

The kidney ports situated on the same side of the valve plate are in parallel arrangement. Both ends of each kidney ports are installed with a damping triangular slot, so this pump can work reversible. In order to achieve parallel flow distributing and balanced flow rate, the cylinder block of the pump must also be modified accordingly. In order to reduce the flow ripple, the double-acting pump is designed with 10 pistons, and the interval between two adjacent pistons is 36°. During each circle of cylinder block, separately 5 pistons are connected to the flow ports C and D totally. To make the cylinder block evenly stressed, the every two adjacent kidney slot at the bottom of cylinder block, which corresponds to each cylinder bore, are staggered as shown in **figure 2**. This configuration not only makes the cylinder block uniformly stressed, but also avoid mutual interference between the suction port and discharge port. By reference to the theories and methods in the literature /8/, simulation model for double-acting axial piston pump can be established. Then the prototype can be manufactured with the reasonable structural parameters calculated through simulation.



(a) Design model (b) Prototype photo

Figure 2: Cylinder three-dimensional design model and its picture

When the double-acting axial piston pump is working, the pistons rotate with the cylinder block. In each circle, the outer pistons pass over the outer kidney ports on the valve plate and absorb and discharge oil once, similarly, the inner pistons pass over the inner kidney ports on the valve plate and absorb and discharge oil once too. In addition, the pistons connected to inner and outer flow distributing windows can work independently. Therefore, in each circle of the pump, there are two separate suction and discharge processes, so is coaled double-acting piston pump. By adjusting the number of pistons connected to inner and outer ports or the pitch circle diameter of the inner and outer pistons, the discharge and suction flow rate of the inner and outer ports can be regulated. In the case of three flow distributing windows, when the pistons pass over the flow distributing port A, the flow within the chamber of inner and outer pistons will be connected to each other, merging into one port.

3. Modelling and simulation

The flow ripple and pressure fluctuation are the two main factors causing the undesirable noise and vibration of the pump, and they are critical to the application of the designed pump. The double-acting axial piston pump has a different flow distributing mode compared with the existing axial piston pump, especially for the flow ports connected to the ports C and D, and only five plungers are used for oil suction and discharge, so the flow fluctuations will be increased and the pump output characteristics will be affected. Therefore, before manufacturing the prototypes, the output characteristics of whole pump are modeled and simulated by simulation package ITI SimulationX®. The structure of the valve plate, especially the structure and size of damping slot are designed. The flow ripple and pressure fluctuation of piston pump are predicted and analyzed through simulation.

3.1. Single piston flow distributing area

The overlapped part between the piston chamber and flow port in valve plate defined as the flow area of an individual cylinder. Four kidney ports in the valve plate are symmetrical, so the calculation methods for the flow area are the same. The radiuses of pitch circles for two kidney ports on the same side are different. Ports A and B, and C and D have the same phase positions respectively, while the phase difference between ports A and C is π. The process of an individual piston passing over a flow port can be divided into seven stages, as shown in **figure 3**.

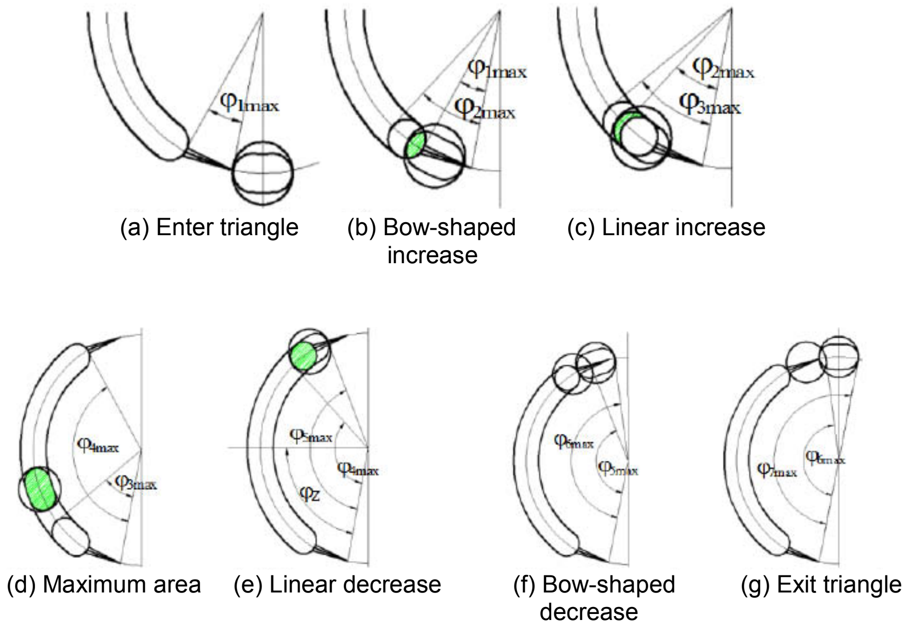


Figure 3: Variation of the flow area of an individual cylinder

The pitch circle radius of pistons movement is *R*, and the radius of kidney slot is *r*. The structure of transition triangular slot at the end of kidney slot is shown in **figure 4**, the bilge length is *l*, and the maximum cross section has a height of *h*, and a length of *L*.

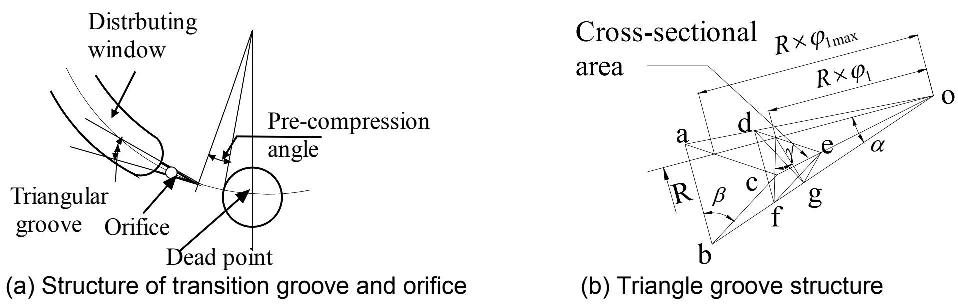


Figure 4: Position relationship the transition groove comparatively to rotor angle

The flow area *A(φ)* is a function dependent to the rotating angle of the cylinder block *φ*, and the formulas for *A(φ)* and *φ* are derived as following:

a) As shown in figure 3(a), the cylinder bore is connected to the damping triangular slot. The flow area $A_1(\varphi)$ is the cross-section triangle area, as shown in figure 4.

$$A_1(\varphi) = R^2\varphi_1^2 lh / L^2 \qquad 0 \le \varphi_1 < \varphi_{1\max} \tag{1}$$

b) As shown in figure 3(b), the cylinder bore starts to be connected to the semi-circular metering edge of the flow port. The flow area $A_2(\varphi)$ is equal to the largest cross-sectional area of the transition triangular slot plus the bow-shaped area S_{G1}.

$$\begin{aligned} A_2(\varphi) &= A_1(\varphi_{1\max}) + S_{\mathrm{G1}} = A_1(\varphi_{1\max}) + 2(r^2\cos^{-1}(1 - R_2(\varphi_2 - \varphi_{1\max})/(2r))) \\ &- \sin\cos^{-1}(1 - R_2(\varphi_2 - \varphi_{1\max})/(2r)) \times r(r - R_2(\varphi_2 - \varphi_{1\max})/2) \quad \varphi_{1\max} \le \varphi_2 < \varphi_{2\max} \end{aligned} \tag{2}$$

c) As shown in figure 3(c), the cylinder bore enters the linear growth area through the semi-circular metering edge of flow distributing port. The flow area $A_3(\varphi)$ is equal to the newly formed bow-shaped area S_{G2} plus $A_2(\varphi_{2\max})$.

$$A_3(\varphi) = A_2(\varphi_{2\max}) + S_{G2} = A_2(\varphi_{2\max}) + 2rR(\varphi_3 - \varphi_{2\max}) \qquad \varphi_{2\max} \le \varphi_3 < \varphi_{3\max} \tag{3}$$

d) As shown in figure 3(d), the cylinder bore completely enters the flow port, and the flow area $A_4(\varphi)$ is $A_3(\varphi_{3\max})$.

$$A_4(\varphi) = A_3(\varphi_{3\max}) = A_2(\varphi_{3\max}) + 2rR(\varphi_{3\max} - \varphi_{2\max}) \qquad \varphi_{3\max} \le \varphi_4 < \varphi_{4\max} \tag{4}$$

e) As shown in figure 3(e), the cylinder bore gradually retreats from the semi-circular metering edge of the flow port and enters the linear reduction area. Z is the symmetrical axis of the flow port, $\varphi_z = (\varphi_{1\max} + \varphi_{4\max})/2$, the flow area $A_5(\varphi)$ and $A_3(\varphi)$ are symmetrical about the Z-axis. So $A_5(\varphi)$ can be calculated as,

$$A_5(\varphi) = A_3(2\varphi_Z - \varphi_5) = A_2(\varphi_{2\max}) + 2rR(2\varphi_Z - \varphi_5 - \varphi_{2\max}) \qquad \varphi_{4\max} \le \varphi_5 < \varphi_{5\max} \tag{5}$$

f) As shown in figure 3(f), the cylinder bore gradually retreats from the semi-circular metering edge of the flow slot and enters the bow-shaped reduction area. The flow area $A_6(\varphi)$ and $A_2(\varphi)$ are symmetrical about the Z-axis, too. So $A_6(\varphi)$ is calculated as,

$$\begin{aligned} A_6(\varphi) &= A_2(2\varphi_Z - \varphi_6) = 2(r^2\cos^{-1}(1 - R_2(2\varphi_Z - \varphi_6 - \varphi_{1\max})/(2r))) + A_1(\varphi_{1\max}) \\ &- \sin\cos^{-1}(1 - R_2(2\varphi_Z - \varphi_6 - \varphi_{1\max})/(2r)) \times r(r - R_2(2\varphi_Z - \varphi_6 - \varphi_{1\max})/2) \end{aligned}$$

$$\varphi_{5\max} \le \varphi_6 < \varphi_{6\max} \tag{6}$$

g) As shown in figure 3(g), the cylinder gradually retreats from the metering edge of transition triangular slot. The flow area $A_7(\varphi)$ and $A_1(\varphi)$ are symmetrical about the *Z*-axis, and $A_7(\varphi)$ is,

$$A_7(\varphi) = A_1(2\varphi_Z - \varphi_7) = R^2(2\varphi_Z - \varphi_7)^2 lh / L^2 \qquad \varphi_{6\max} \le \varphi_7 < \pi \tag{7}$$

According to above formulas, the relationship between $A(\varphi)$ (four flow ports in valve plate) and φ are shown in **figure 5**. Piston periodically passes over the flow ports A and C or B and D. When the piston passes over the triangular slot, $A(\varphi)$ is very small. With the rotation of the cylinder block, the piston bore will gradually enter the bow area and the linear increasing area. After the $A(\varphi)$ reaches a maximum value, it will maintain at a certain angle and then gradually be reduced to zero, and then enter the next flow port. At the same time, each cylinder bore can only be connected with one flow port.

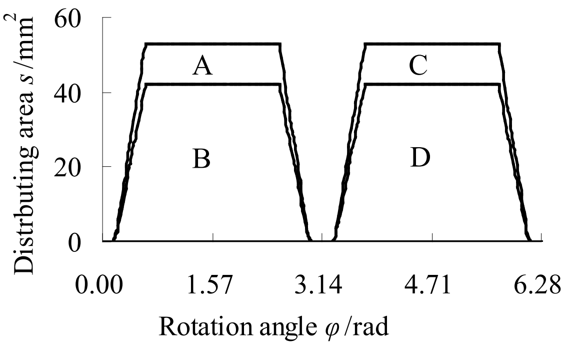


Figure 5: Flow distributing area varying curves with rotation angle

3.2. Leakage calculations of pump

Leakage in the piston pump mainly includes the gap leakage q_{S1} between the cylinder bore and the piston, the volume loss q_{S2} between the slipper and swash plate, and the flow loss q_{S3} between the valve plate and cylinder block. The total leakage of the axial piston pump is $q_1 = q_{S1} + q_{S2} + q_{S3}$, which will affect the discharge flow rate of the pump. The calculating formulas are

$$q_{S1} = \pi d\delta_1^3(1+1.5\varepsilon^2)(p_z - p_p)/(12\mu l_1) - \pi d\delta_1 v_p / 2 \tag{8}$$

$$q_{S2} = \pi d_d^4 \delta_2^3 (p_z - p_p) / \left(\mu\left(6d_d^4 \ln(r_2/r_1) + 128 l_d \delta_2^3\right)\right) \tag{9}$$

$$q_{S3} = \frac{\alpha_f \delta_3^3 (p_z - p_p)}{12\mu}\left[\frac{1}{\ln(R_{r2}/R_{r1})} + \frac{1}{\ln(R_{r4}/R_{r3})}\right] \tag{10}$$

3.3. Single piston model

Based on the above analysis, a complete set of single-piston model and packaging model can be established by using simulation package ITI SimulationX®, as shown in **figure 6**.

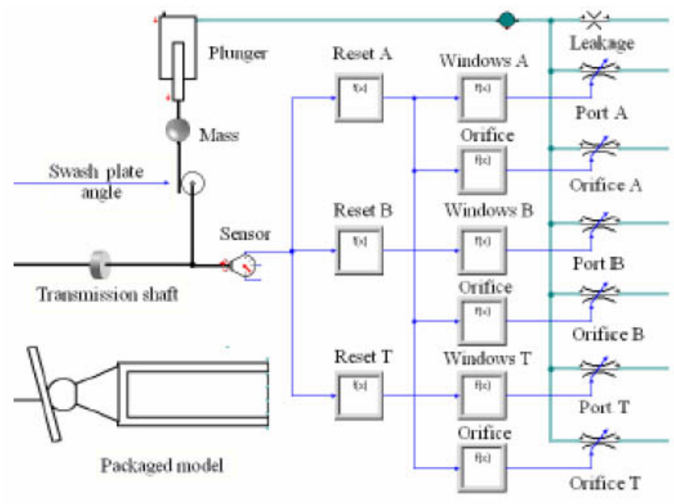


Figure 6: Single piton simulation and packaged model

In the simulation model, the opening area of a variable throttle valve is substituted for the flow area of an individual cylinder, and the opening area of the valve can be calculated through Eq.(1-7).The resetting module is used to achieve the periodical rotation of piston around the transmission shaft. The inner parameters of piston model are used to determine q_{S1}, and the leakage module is used to determine q_{S2} and q_{S3}.

3.4. Analysis of simulation results

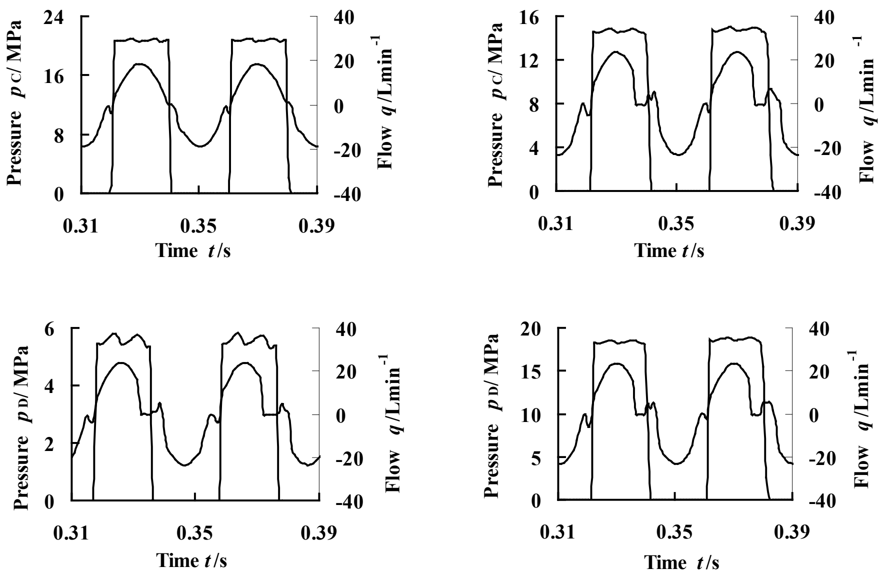
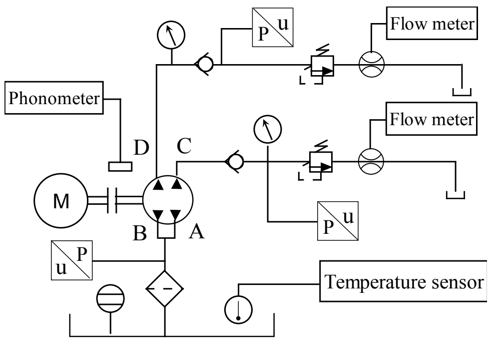


Figure 7: Flow and pressure characteristics windows C and D

The rotational speed of pump is 1500 r/min, the ports A and B suck oil, while ports C and D discharge oil, respectively. As the piston passes over the flow ports C and D, the pressure and flow characteristics within a single piston chamber are shown in **figure 7**. As shown in figure 7, It can be found that the discharge flow of the ports C and D are independent, and the pressure and flow rate are stable when P_C=P_D=21 MPa. In addition, q_C+q_D= q_A holds. It is obvious that port C and D can be loaded independently. So the double-acting axial piston pump can be applied on two different hydraulic circuits. On the basis of simulation results, the reasonable structure parameters can be determined and prototypes can be created.

4. Experimental test

In theory, if the two ports on one side of the pump with four ports are unified as one general port, then its performance should be equivalent to that of the pump with three ports. But in fact, the experimental results show that the pressure and flow characteristics of the pump with four ports are different from that of pump with three ports. In the paper, the noise, pressure and pulse spectrum of each working port of pump are tested. The testing principle and photo of the testing system are shown respectively in **figure 8**. In the testing system, the servo motor is used to control the conversion between suction and discharge of the pump. Each port of the pump is installed with a pressure sensor. A pilot relief valve is installed respectively at ports C and D to load the pump. When port C is loaded, ports A and B directly return to the tank, and the retracting of piston rod of the differential cylinder can be simulated. When ports C and D are loaded, a pump can be simulated as a double-use pump.



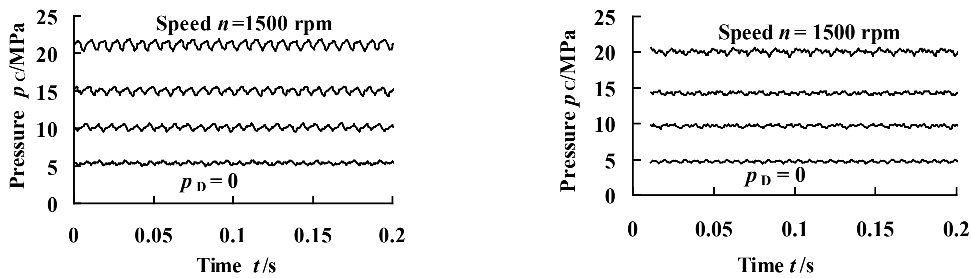
(a) Principle of testing system


(b) Photo of the testing system

Figure 8: Principle and photo of the testing system

4.1. Pressure test results

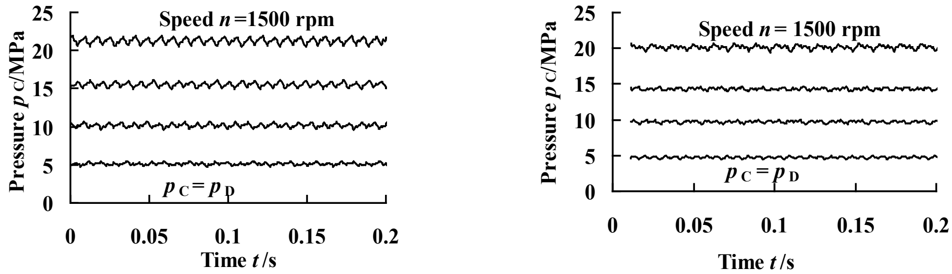
a) When the rotational speed is 1500r/min, port C is applied with different loads, and port D is idle, the pressure response of port C of the two kinds of pumps is tested and the results are shown in **figure 9**.



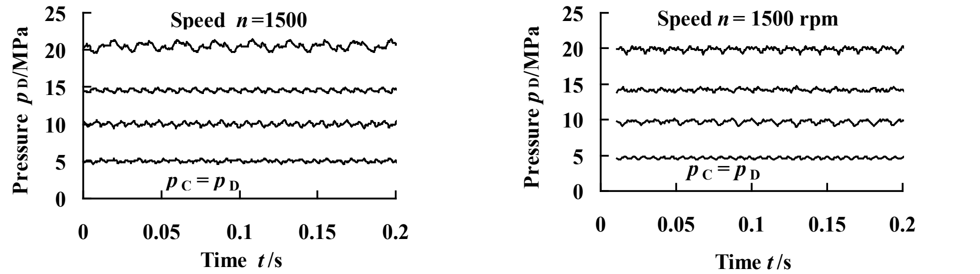
(a) Pressure in port C of 3 ports pump (b) Pressure in port C of 4 ports pump

Figure 9: Pressure characteristic curves tested in port C

a) When the rotational speed is 1500r/min, both port C and D are loaded. The pressure response of the two kinds of pumps is tested and the results are shown in **figure 10**.



(a) Pressure in port C of 3 ports pump (b) Pressure in port C of 4 ports pump



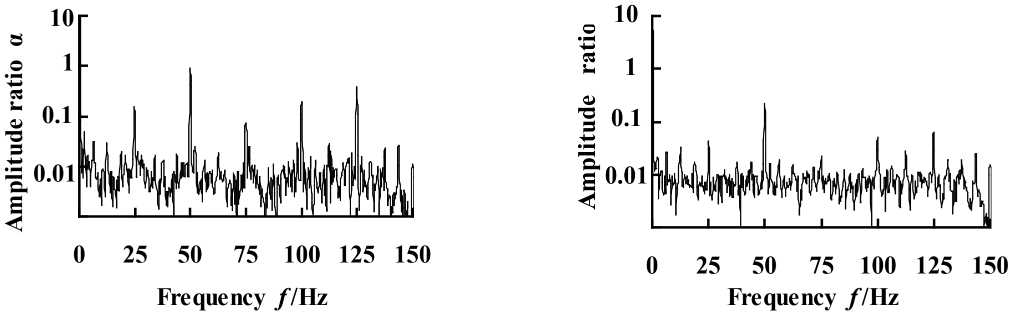
(c) Pressure in port D of 3 ports pump (d) Pressure in port D of 4 ports pump

Figure 10: Pressure characteristic curves tested in outlet C

From figure 10, it can be found that the pressure characteristics of ports C and D of the two kinds of pump are relatively stable on the whole, but there are still small fluctuations. When the loading pressure is low, the pressure fluctuation is relatively small. The pressure fluctuation amplitude increases as the load increases, but the pressure fluctuation amplitude of the pump with three ports is greater than that of the pump with four ports. By comparing the results of figure 9 and figure 10, it can be found that the load at port D has no influences on the pressure pulsation of the port C.

4.2. Pressure spectrum test results

To facilitate the study of pressure spectral characteristics of the pump, the rotational speed is set to be 1500 r/min, and port C is applied with pressure 21 MPa, the pressure pulsation spectrum curve of two prototype pumps is measured and shown in **figure 11**.



(a) Pump with 3 windows (b) Pump with 4 windows

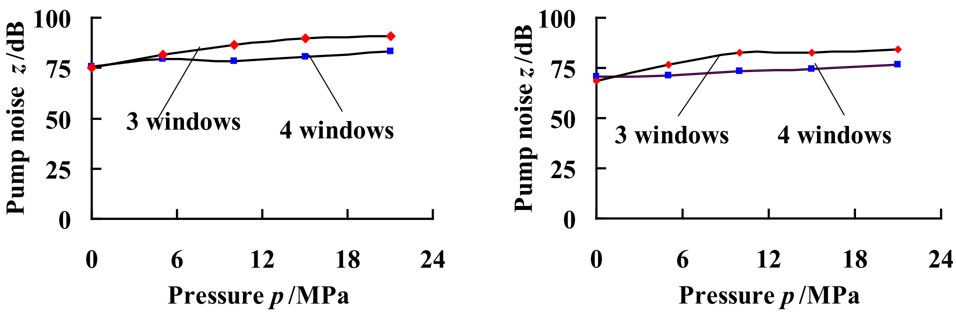
Figure 11: Spectrum curves of the prototype pump at speed 1500 rpm

According to the theoretical calculations, the pressure (flow) pulse frequency of a single cylinder is the number of revolutions per second. The pump has ten pistons, when the rotational speed is 1500 r/min, the pressure pulsation frequency of a single cylinder is 25 Hz; according to the function of pump pressure pulsation, a large pulsation amplitude will occur under the integer multiples of the pump pressure pulsation frequency. The test results shown in figure 11 also illustrate this rule. The pressure pulsation frequency points of two pumps are basically the same, and the pressure pulsation amplitude in the integer multiples of 25 Hz is significantly large. In the case of applying the same load, the difference of pressure pulse amplitude between the two pumps is large; the pressure pulsation amplitude of double-acting piston pump with three flow distributing windows is much higher than that of the double-acting pump with four flow distributing windows; and the pressure pulsation amplitude of the former one is about 10 times higher than that of the latter one on the average.

Through the comparison and analysis of the time-domain pressure pulsation in figure 9, it can be seen that the frequency range with low amplitude in the frequency domain diagram reflects small amplitude pulsations in the time domain response. The pressure pulsation of the pump with three flow ports is larger than that of the double-acting pump with four ports, which has something to do with the isolation structure and symmetrical structure between the flow distribution ports on the same side of the double-acting piston pump with four flow distributing windows. In addition, being affected by the noise of AC power supply, the pulse amplitude of two pumps at 50 Hz reaches the maximum value. The pressure process and the spectrum are tested in the case of two kinds of pump speeds, i.e., 1000 r/min and 500 r/min. The pump pulsation rule is consistent with that at the speed of 1500 r/min.

4.3. Noise testing results

During the test, the sound level meter is used to record the noise of two pumps with different numbers of revolutions. **Figure 12** shows that when the rotational speed is 1500 r/min and 1000 r/min, port D has no load, and the noise of two pumps changes with the load applied at the port C.



(a) Speed1500 r/min

(b) Speed 1000 r/min

Figure 12: Noise characteristics of the prototype pump

It can be seen from figure 12 that, in the case of different rotational speed, the noise increases with the increase of pump loading pressure. In the case of the same speed and small load, the noise of two pumps has few differences. As the load increases, the noise of the piston pump with four ports is essentially the same, and that of pump with three flow ports is increased by about 10 dB; the noise level is higher than that of the pump with four flow ports, which shows similar changes consistent with pressure pulsation characteristics changes in the previous tests. From the comparative analysis of the results both from time domain and frequency domain in figure 9 and figure 11, it can be obtained that the pressure and flow pulsation of double-acting axial piston are the main factors causing the noise of piston pump; if the pump outlet load is greater, the pump pressure fluctuation is greater, and the noise is increased, and the results of the pump with three ports are even more significant, which indicates that the performance of the symmetrical double-acting piston pump with four flow distributing windows is superior to that of the pump with three flow distributing windows.

5. Conclusion

1) The test results verify the practical feasibility of the double-acting axial piston pump which can be used as a single pump and a double pump to drive two different loads, or to direct control the differential cylinder in closed circuit.

2) The outlet pressure pulsation of double-acting axial piston pump increases with the increase of the outlet load. In some pressure situation a minor pressure fluctuations is overlapped, but the overall pressure response performance is stable, so the pump can

be used in the actual system.

3) From the time domain, frequency domain and noise level, it is shown that the performance of symmetrical double-acting axial piston pump with four flow distributing windows is better than that of the pump with three flow distributing windows.

6. References

/1/ SHA Nansheng, LI Jun. Research on airborne power-by-wire integrated electrical actuation and control systems[J]. Journal of Beijing University of Aeronautics and Astronautics, 2004，Vol.30 (9): 909-912. (in Chinese)

/2/ Kang Rongjie, Jiao Zongxia, Wang Shaoping et al. Design and Simulation of Electro-hydrostatic Actuator with a Built-in Power Regulator [J]. Chinese Journal of Aeronautics, 2009, Vol.22 (6): 700-706.(in Chinese)

/3/ GAO Bo, FU Yong-ling, PEI Zhongcai et al. Research on Dual-Variable Integrated Electro-Hydrostatic Actuator[J]. Chinese Journal of Aeronautics, 2006, Vol.19 (1):77-82. (in Chinese)

/4/ LODEWYKS J. Variable displacement pumps closed loop controlled differential cylinder system [D]. Aachen: RWTH，1994

/5/ Vickers Inc. Electro-hydraulic system and apparatus with bidirectional electric motor hydraulic unit [P]. USA, 98/113581998[P].1998-03-19

/6/ FEUSER A, DANTLGRABER J, SPATH D. Servopumpenantriebe für Differentialzylinder [J], O+P Oelhydraulik und Pneumatik, 1995, 39(7): 540-544

/7/ LI Jianguo, QUAN Long. New strategy on differential cylinder controlled by electro-hydraulic pump system [J]. Fluid Power Transmission and Control, 2009, 35 (4):5-7. (in Chinese)

/8/ YANG Huayong, MA Jien, XU Bing. Research status of axial piston pumps fluid-borne noise [J]. Chinese Journal of Mechanical Engineering, 2009, 45(8): 71-78. (in Chinese)

7. Nomenclature

R_1, R_2	Pitch circle radius of outer and internal flow windows	mm
δ_1, δ_2, δ_3	Gap piston to cylinder, slipper to swash plate, valve plate to cylinder	mm
r_1, r_2	Internal and external radius of slipper sealing band	mm
R_{r1}, R_{r2}, R_{r3}, R_{r4}	Internal and external radius of valve plate sealing band	mm