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DOI: 10.1177/0954406212462336
The online version of this article can be found at:
http://pic.sagepub.com/content/227/3/459
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Abstract
Noise reduction in axial piston pumps has been attempted by many researchers with different design approaches and techniques. However, most traditional structures on valve plate for noise reduction are at the cost of efficiency to different extent. In this article, a new distribution method with pressure equalization mechanism composed of check valve and pressure recuperation chamber is discussed. A simulation model for the analysis of noise excitation sources is developed, and is verified by comparison of flow ripple between simulation and experiment. The working principle of pressure equalization mechanism is analyzed in detail. Compared with reference commercial axial piston pump, the simulation results indicate that the flow ripple and the torque pulsation are sharply reduced with the pressure equalization mechanism. Moreover, the volumetric efficiency of axial piston pump is also improved. The power of variable-displacement control mechanism will be reduced and the control accuracy can be improved easily due to the swash-plate torque reduction. The analysis shows that the check-valve frequency and the pressure recuperation chamber volume are vital for the pressure equalization mechanism. The optimal pressure recuperation chamber volume is about three times the size of the minimum piston chamber dead volume. The optimal maximum displacement of check valve is about 1 mm. The pressure equalization mechanism is promising in the design of high-performance axial piston pump with low noise emission.

Keywords
Axial piston pump, flow ripple, noise reduction, efficiency

Introduction
Axial piston pumps are widely used in hydraulic driving systems for their advantages such as high power-to-mass ratio, high limit load pressure and excellent controllability. However, it also has obvious disadvantages such as high noise emission, especially with the increase of operating parameters. Pumps are the main noise sources in hydraulic systems. On one hand, the force pulsation inside the pump causes vibrations of the pump shell. On the other hand, the ripple of the delivery flow interacts with the downstream load components, which creates pressure pulsation that causes the vibrations of hydraulic components and pipelines. The magnitude and frequency of both vibrations can be large, resulting in not only high noise emission, but also looseness and fatigue failure of components. In order to realize the overall noise reduction of axial piston pump, the flow ripple of the delivery flow and swash-plate torque pulsation should be reduced at the same time, and it should not be at the cost of efficiency. Design methods based on multi-objective optimization were proposed in the noise reduction researches.1,2

In the past several decades, many methods were proposed to reduce the noise emission of axial piston pump. Pettersson et al.,3 Nafz et al.,4 and Ivantysynova et al.5 made a comprehensive summarization about these researches. The traditional researches mostly focused on the optimization of valve-plate transition region, including relief grooves and timing.6–8 The valve plate timing is quite sensitive to the pressure level and swash-plate angle. Therefore, the commercial axial piston pumps applied in open-loop systems usually have both relief grooves and timing. However, during the pressure transition in
piston chamber from high delivery pressure to low suction pressure, a small part of the high-pressure oil in piston chamber flows into suction kidney through the pressure relief groove. This inner leakage will reduce the pump efficiency.

Some other methods such as pre-compression filter chamber,9–11 relief check valve,12 cross angle of swash-plate13,14 and some adjustable systems also proved to have satisfying results in noise reduction and were tried to be applied in the commercial pumps. However, most of these methods focused on the pressure transition from suction to delivery pressure to reduce the backflow from delivery kidney into piston chamber. The pressure transition from delivery to suction pressure was still at the cost of efficiency, especially at high pressure and small swash-plate angle.

During the transition region, the pressure variations in piston chamber at outer dead center (ODC) where the volume of piston chamber is the largest and inner dead center (IDC) where the volume of piston chamber is the smallest are contrary. Therefore, the hydraulic energy stored in the dead volume at IDC can be used to pressurize the low-pressure piston chamber at ODC. Kahrs15 proposed the pressure equalization between piston chambers at ODC and IDC firstly. Lemmen and Schmitt16 proposed a pressure equalization system with additional hydraulic chamber. Nafz et al.4 analyzed a pressure equalization system with valve-controlled pressure recuperation chamber (PRC), and it showed that both the efficiency and the noise emission reduction could be improved apparently.

This article analyzes the noise reduction method of axial piston pump based on pressure equalization principle with check valve and PRC. The effect of this method on delivery flow ripple, swash-plate torque pulsation and pump efficiency are investigated in detail based on a simulation model of axial piston pump verified by flow-ripple experiment. The structural parameters of pressure equalization mechanism are optimized to have better noise reduction and efficiency improvement effect. Some encouraging results are obtained, which can be used in the design of low-noise and high-efficiency axial piston pump.

**Simulation model and experiment verification**

In order to analyze the noise excitation source, a general simulation model of an axial piston pump is developed with the AMESim software. The schematic is mainly composed of the cylinder module and the valve-plate module as shown in Figure 1.17 The cylinder module calculates the flow rate and pressure variation in each piston chamber. \( n \) is the number of piston chamber in the cylinder module which could be odd or even as needed. The flow rate into and out of the piston chamber are modeled using the classical orifice equation. The actual pressure in piston chamber is calculated by integrating the pressure-rise-rate defined by

\[
\frac{dP}{dt} = \frac{K_{c}}{V_{pc}} \left( Q_{s} - Q_{d} - Q_{l} - \frac{dV_{pc}}{dt} \right)
\]

\[
Q_{s} = CA_{lp} \sqrt{\frac{2|P - P_{l}|}{\rho}} \cdot \text{sign} \left( P_{l} - P \right)
\]

\[
Q_{d} = CA_{hp} \sqrt{\frac{2|P_{h} - P_{l}|}{\rho}} \cdot \text{sign} \left( P_{h} - P \right)
\]

\[
Q_{l} = Q_{lp} + Q_{ls} + Q_{lv}
\]

This equation is used to calculate the actual pressure in piston chamber.

**Figure 1.** Schematic diagram of pump simulation model.
The leakage of each piston chamber is composed of three parts. The piston/cylinder pair is not an ideal cylindrical joint. The piston axis is eccentric relative to the cylinder-bore axis due to the lateral forces, and the piston guidance length varies with the rotation angle. Therefore, the piston leakage $Q_{lp}$ employs the variable length and eccentricity model as depicted by

$$Q_{lp} = \frac{\pi d_p \delta_l^2}{12\mu_l} \left(1 + 1.5\varepsilon^2\right)(P - P_c)$$

(2)

The fixed damping orifices in the piston and slipper plus the varying gap height of the slipper/swash-plate pair form a hydrostatic bearing mechanism. Assuming an average constant gap height, the slipper leakage $Q_{ls}$ can be calculated by

$$Q_{ls} = \frac{C_{pb}C_{sb}C_{sp}}{C_{pb}C_{sb} + C_{pb}C_{sp} + C_{sb}C_{sp}}(P - P_c)$$

$$C_{pb} = \frac{\pi d_{pb}^2}{128\mu_{pb}}$$

$$C_{sb} = \frac{\pi d_{sb}^2}{128\mu_{sb}}$$

$$C_{sp} = \frac{6\mu \ln(R_{op}/r_{op})}{\pi \delta_l^3}$$

(3)

The valve plate leakage is the flow from piston chamber into the pump shell through the inner and outer sealing belts of valve plate. Assuming an average constant gap height between the valve plate and cylinder, it can be calculated by employing the gap flow model between two parallel discs as depicted by

$$Q_{lv} = \frac{\alpha \delta_l^3}{12\mu_l} \left[\frac{1}{\ln(R_2/R_1)} + \frac{1}{\ln(R_3/R_2)}\right](P - P_c)$$

(4)

The valve plate module calculates the flow area variation for each piston chamber in a cycle. The flow area variation of a piston chamber is the key factor in determining its pressure transition, especially during the transition region where the geometrical cross-sectional area changes sharply due to the relief grooves. The flow area variations $A_{hp}$ and $A_{lp}$ are calculated precisely by a piecewise function as show in Ma et al.6

Because the axial friction force of a piston is much smaller than the axial force applied by the pressure in the piston chamber, it is neglected in the calculation of the swash-plate torque as depicted by equation (5). The force $F_{AK}$ is determined by the pressure in piston chamber and the piston inertia.

$$M_{sx} = \frac{R_p}{\cos^2 \beta} \sum_{i=1}^{n} F_{AKi} \cdot \cos \phi_i$$

(5)

The simulation model for a piston chamber is shown in Figure 2. The variable length and eccentricity model for the piston leakage is a component from the AMESim libraries. The leakages of slipper/swash-plate and cylinder/valve-plate, the suction and delivery flows are all modeled as damping orifices. The flow area $A_{hp}$ and $A_{lp}$ for the delivery kidney and suction kidney are integrated in the form of two ASCII files.

The accurate bulk modulus is important for the calculation of pressure overshoots during the transition region.18 The hydraulic oil in the commercial software has several different models. In this simulation model, the fluid bulk modulus is defined with a table of bulk modulus values against pressure and temperature from experimental results as shown in Figure 3. Because of the presence of air, with the variation of pressure, the air can be dissolved in the liquid or can be free as bubbles, which is related to cavitations of the valve plate.

The simulation model of axial piston pump is composed of several piston chambers. In order to be in accordance with the experimental research, a 2-m hard pipeline is placed between the load throttle valve and the fixed chamber of the delivery passage-way. It can analyze the instantaneous pressure in piston chamber, the outflow ripple, the swash-plate torque pulsation and the leakages of each friction pairs in a cycle. Therefore, the noise excitation source and efficiency of axial piston pump utilizing different flow distribution methods can be analyzed based on this simulation model. The extra module is optional and could be different according to the

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**Figure 2.** Simulation model of one piston chamber.
investigated flow distribution method. In this research, it includes a mathematical model of check valve and a fixed chamber. The flow area between the piston chamber and extra module is also integrated into the simulation model in the form of ASCII files.

The pressure in the piston chamber and the swash-plate torque pulsation are inner dynamic characteristics of axial piston pump which are difficult to be tested. In order to verify the general simulation model, a test rig for flow ripple measurement at the delivery port of axial piston pump was designed. The flow ripple of axial piston pump is a kind of signal with high frequency that cannot be tested directly by the present commercial flow transducer. Therefore, the test rig was built based on the secondary source method as shown in Figure 4. Both the test pump and the secondary source pump can work at different rotational speed with the control of a frequency converter.

Figure 5 shows the flow ripple comparison between the simulation and experimental results of an axial piston pump whose displacement is 56 cm³. The load pressure is 10 MPa, and the rotation speed is 1000 r/min. The experimental flow ripple characteristic is obtained indirectly by processing the four tested pressure signals in the straight pipeline. The measurement error and the two transformations between time domain and frequency domain during the data processing result in the differences between the simulation and experimental results. The peak-to-peak value of the experimental result is 3.99 L/min, while the simulation result is 4.06 L/min. Comparing with the average flow rate, the flow ripple rate of experimental result is about 10.19%, while the simulation result is 10.21%. Therefore, the simulation model has a satisfying accuracy for the noise-excitation source evaluation of axial piston pump. It can be applied in the design and optimization of the flow-distribution structures.

**Pump with pressure equalization mechanism**

Using the hydraulic energy stored in the dead volume at the IDC to pressurize the piston chamber at the ODC directly, it requires an even number of pistons. However, because the flow ripple and swash-plate torque pulsation of the axial piston pump with an even number of pistons are much larger than those of pump with an odd number of pistons, the practical commercial axial piston pumps almost all have an odd number of pistons, usually nine or seven.

For pumps with odd pistons, there is no another piston chamber near the ODC when a piston chamber passes the IDC, and vice versa. Therefore, the hydraulic energy stored in the piston chamber near the IDC has to be transferred into an intermediate chamber first, as shown in Figure 6. The positions of the orifices ensure that the piston chamber connects with the PRC as soon as it disconnects from the kidney slots. The pressure transition from high delivery pressure to low suction pressure in the piston chamber near the IDC is a little ahead of the inverse pressure transition in the piston chamber near the ODC for the pressure equalization process, because the hydraulic energy stored in the piston chamber 1 can only pressurize...
the piston chamber 5. The check valve guarantees that the oil can only flow from the high-pressure piston chambers to the low-pressure ones.

**Working principle analysis**

In the design and analysis of axial piston pump utilizing pressure equalization mechanism, a commercial pump with nine pistons is selected as the reference pump whose displacement is 71 cm³. The main parameters of the pump employed in the simulation model are shown in Table 1. The sub-model of the pressure equalization mechanism is shown in Figure 7. The orifices connecting with the check valves are also integrated in the form of ASCII files. The model of the check valve takes the dynamic characteristics of opening and closing processes into consideration. A tiny high-frequency redesigned commercial check valve is chosen whose moving part weights only 0.2 g. Its nominal diameter is 2 mm, and the maximum displacement is only 1 mm. The volume of PRC is set as 30 cm³ at first. The diameter and length of the flow passages are respectively set as 2 mm and 80 mm. Moreover, the relief grooves and timing of the original valve plate are all reserved. The rotation speed is 1500 r/min, and the load pressure is about 20 MPa. The inclined angle of the swash plate is set as 12°.

The pressure transition process in the piston chamber at ODC is employed to analyze the working principle of pressure equalization mechanism. The pressure equalization process between the PRC and the piston chamber at the ODC is shown in Figure 8. It can be divided into three parts.

1. As shown in Figure 6, the pressure in the flow passage between the orifice and check valve near the ODC is the delivery pressure after the piston chamber 6 passes. That is why the check valve stays closed. When the piston chamber 5 finishes the suction process and connects with the orifice, the pressure in this passages reduces sharply to low suction pressure. The check valve opens and the displacement increases quickly because the pressure in the PRC is larger than the pressure in the piston chamber 5. The flow rate into this piston chamber increases with the increasing opening. Thus, the pressure in the PRC decreases, meanwhile the piston chamber is

**Table 1. Main parameters of axial piston pump.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston radius (mm)</td>
<td>10</td>
</tr>
<tr>
<td>Piston pitch radius (mm)</td>
<td>40.5</td>
</tr>
<tr>
<td>Minimum dead volume of piston chamber (cm³)</td>
<td>9.10</td>
</tr>
<tr>
<td>Mean clearance between piston and cylinder bore (μm)</td>
<td>15</td>
</tr>
<tr>
<td>Mean clearance between valve plate and cylinder (μm)</td>
<td>15</td>
</tr>
<tr>
<td>Mean clearance between slipper and swash plate (μm)</td>
<td>15</td>
</tr>
</tbody>
</table>
pressurized, and the pressure difference between the piston chamber and the PRC decreases quickly.

II The opening velocity of check valve slows down and then moves backwards when the thrust force due to the pressure difference is smaller than the spring force of the check valve. The flow rate from the PRC into the piston chamber decreases with the reduction of both the pressure difference and the valve opening. The flow decreases to zero when the pressures in the PRC and the piston chamber are equal.

III Although the pressures in the piston chamber and the PRC are already equal, the check valve is still open at this moment. The pressure in the piston chamber is still increasing through the axial motion of piston and the backflow from the delivery kidney, so a small amount of fluid oil flows back into the PRC due to the new inverse pressure difference until the check valve totally closes. The pressure in PRC increases a little due to this small backflow. The pressure gradient in piston chamber slows down apparently at this stage. Although the check valve closes, the passage is still connecting with the piston chamber, so the pressure in this passage will increases to the high delivery pressure and waits for the next pressure equalization process.

If the dynamic characteristic of check valve is neglected, the closing and opening are only determined by the pressure difference and take no time, so the small part of backflow at the third stage will not exist. Thus, more hydraulic energy stored in the PRC is used in the pressurization of low-pressure piston chambers. Hence, the check-valve frequency should be as high as possible to reduce their influence on the pressure equalization mechanism.

**Noise reduction analysis**

The comparisons of flow ripple and torque pulsation between axial piston pumps with and without pressure equalization mechanism are shown in Table 2. It shows that the pressure equalization mechanism with check valve and PRC is quite effective in noise reduction.

**Flow ripple reduction.** Figure 9 shows the effect of pressure equalization mechanism on flow ripple reduction. With the pressure equalization mechanism, the peak-to-peak value decreases from 8.95 to 6.50 L/min, reduced by 2.45 L/min, and the flow ripple decreases from 12.96% to 9.20%. Figure 10 shows the relationship between the out-flow rate at the pump delivery port and the delivery flow of one piston chamber. It shows that the value of the backflow from the delivery kidney to the piston chamber is crucial in determining the minimum value of outflow. For the axial piston pump with pressure equalization mechanism, the high-pressure oil in the piston chamber at the IDC contributes to the pressurization of the piston chamber at the ODC, so the backflow from the delivery kidney to pressurize the piston chamber at the ODC will decrease. That is why the minimum outflow of pump with pressure equalization mechanism increases. Because the relief grooves of the original valve plate are reserved, the pressure in the piston chamber is more over pressurized with the extra pressure equalization mechanism. Therefore, when the piston chamber connects with the delivery kidney, there is a larger delivery flow due to the larger pressure difference. This is why the outflow of pump with pressure equalization mechanism has a larger abrupt peak as shown in Figure 9. It can be decreased or eliminated by the optimization of the relief grooves.
based on the pump with pressure equalization mechanism.

**Torque pulsation reduction.** Figure 11 shows the effect of pressure equalization mechanism on swash-plate torque pulsation reduction. With the pressure equalization mechanism, the mean value of swash-plate torque decreases from 104.18 to 53.00 Nm, reduced by 49.12%. The peak-to-peak value decreases from 123.06 to 84.56 Nm, reduced by 31.29%. Therefore, the oscillating magnitude of the source for pump shell vibration decreases apparently. What’s more, it is the input for variable-displacement control mechanism, so the input power for the variable-displacement control mechanism of axial piston pump also decreases, and the control accuracy can be improved due to a smaller load.

Figure 12 compares the swash-plate torque and pressure in piston chamber between the reference pump and pump with pressure equalization mechanism. The positions where the phase angles are $0^\circ$ and $180^\circ$ present the ODC and IDC, respectively. The high-pressure piston chambers contribute to the swash-plate torque. Because the pressures in the adjacent piston chambers have the fixed phase difference, the thin continuous lines show the positions of the high-pressure piston chambers when the torque is the largest for the reference pump, meanwhile the thin dashed lines show the positions when the torque is the largest for pump with pressure equalization mechanism. Therefore, the difference of pressure gradient in the piston chamber at transition regions affects the maximum and minimum swash-plate torques. This is why the pressure equalization mechanism helps to reduce the swash-plate torque.

**Efficiency improvement.** The average outflow rate of pump increases from 71.00 to 71.86 L/min due to the pressure equalization mechanism at the same working parameters as shown in Table 2. One reason is that the backflow from delivery kidney into the piston chamber is reduced. Another reason is that the piston chamber realizes pressure transition in shorter time, which means that the actual suction and delivery periods increase as shown in Figure 10. The increase of output power lessens the harmful influence of the pressure equalization mechanism on the pump power-to-mass ratio.

The efficiency improvement of pressure equalization mechanism is determined by the amount of recuperated high-pressure fluid oil from the piston chamber at the IDC. Table 3 shows efficiency comparisons at different pressures and swash-plate angles between pumps with and without pressure equalization mechanism. Because only the leakages of three friction pairs are taken into consideration in the simulation model, the pump efficiency in Table 3 is approximately the volumetric efficiency of commercial axial piston pump. The pump efficiency reduces with the increases of the load pressure and swash-plate angle. The efficiency improvement is more evident at the high pressure and small swash plate, 3.27% at 28 MPa load pressure and 5° swash-plate angle.
That is because the dead volume of the piston chamber at the IDC is larger at smaller swash-plate angle, and the pressure difference is larger between the piston chamber and the PRC at higher load pressure. Therefore, more energy is recuperated during the pressure equalization process. This efficiency improvement is valuable because the lower efficiency at small displacement is an important drawback for axial piston pump.

Therefore, the pressure equalization mechanism with check valves and PRC has three advantages. Firstly, it can reduce the pump noise emission evidently because of its smaller magnitude of flow ripple and torque pulsation. Secondly, it can improve the pump efficiency slightly because of the energy recuperation. Thirdly, it can reduce the power needed for variable-displacement control mechanism and improve the control accuracy outstandingly because of swash-plate torque reduction.

**Parameters optimization**

The parameters of pressure equalization mechanism include the size of the flow passages, the position of the orifices, the parameters of the check valves and the volume of the PRC. In order to improve the noise reduction effect, these parameters should all be optimized.

To eliminate the cross-flow, the positions of the orifices should ensure that the piston chamber connects with the PRC as soon as it disconnects with the kidney slots, so the position is determined by the wrap angles of the piston-chamber out port and kidney slots. The diameter is equal to the nominal diameter of check valve.

As mentioned above, the volume of the flow passages should be as small as possible. The diameter of flow passage is equal to the nominal diameter of check valve, and the length should be as short as possible.

**PRC volume.** To select an optimal value for the PRC volume, simulations are carried out for the PRC sizes ranging from 20 to 60 cm³, and the outflow ripple at the delivery port and the swash-plate torque for pumps with PRC of different sizes are compared as shown in Table 4.

For the larger PRC, the pressure difference between the PRC and the piston chamber vanishes more slowly. Therefore, more hydraulic energy...
stored in the piston chamber at the IDC is transferred into the PRC, and then into the piston chamber at the ODC. As shown in Table 4, the flow ripple and the average swash-plate torque decreases gradually as the PRC volume increases, but the torque amplitude increases. Overall, the variations of the flow ripple and torque pulsation are quite small. What’s more, larger PRC will increase the pump mass, which will decrease the power-to-mass ratio. Therefore, taking all these factors into account, 30 cm³ may be the best choice for the PRC size, which is about three times the size of minimum dead volume of piston chamber shown in Table 1, because the summation of the four parameters in Table 4 is the smallest when the volume is 30 cm³. Accurate value calls for smaller step in the simulation analysis and experimental researches in the future.

Check valve. As mentioned above, a tiny high-frequency check valve is selected in the simulation model. However, the maximum displacement of the moving part has to be redesigned to improve the dynamic characteristics. Figure 13 compares the flow rates between the piston chamber at the ODC and the PRC for check valve with different maximum displacements. The minus value means the flow is into the PRC.

The closing time of check valve is much longer than the opening time. If the maximum displacement is smaller than 1 mm, the closing time almost just keeps the same. However, the maximum flow rate decreases for the smaller opening. When the maximum displacement is larger than 1 mm, the flow rate from the PRC into the piston chamber at ODC nearly stays the same. However, the backflow from the piston chamber into the PRC is larger because the closing time of the check valve becomes longer. As shown in Table 5, the net high-pressure oil transferred from the PRC into piston chamber at the ODC is largest if the maximum displacement is 1 mm, and the flow ripple and the swash-plate torque pulsation are the smallest. Therefore, 1 mm is the best choice for the maximum displacement of check valve.

A model pump for flowing experimental research designed according to the analysis and optimization in this article is being manufactured based on a commercial axial piston pump. The actual effect of the pressure equalization mechanism on noise reduction of axial piston pump will be investigated experimentally in details.

Conclusions

1. The simulation model developed with AMEsim can analyze the flow ripple, swash-plate torque and efficiency of axial piston pump. The experimental tests show that it has a satisfying accuracy. It can be used in the analysis and optimization of noise reduction for axial piston pump.

2. The pressure equalization mechanism with check valve and PRC is quite effective in noise reduction. Compared with the traditional commercial axial piston pump, the flow ripple amplitude can be reduced by more than 25%, and torque pulsation can be reduced by more than 30%. Moreover, the efficiency of axial piston pump is also improved, especially at high load pressure and small swash-plate angle. The pressure equalization mechanism is promising in the design of high-performance axial piston pump with low noise emission.

3. The dynamic characteristic of check valve and the PRC volume are vital for the noise reduction effect of the pressure equalization mechanism. The natural frequency of check valve can be improved by reducing the maximum displacement, and the optimal value is about 1 mm. The optimal PRC size is about three times the size of the minimum dead volume of piston chamber.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>20 cm³</th>
<th>30 cm³</th>
<th>40 cm³</th>
<th>50 cm³</th>
<th>60 cm³</th>
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<tbody>
<tr>
<td>Amplitude (L/min)</td>
<td>6.63</td>
<td>6.50</td>
<td>6.43</td>
<td>6.39</td>
<td>6.38</td>
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<tr>
<td>Flow ripple (%)</td>
<td>9.39</td>
<td>9.20</td>
<td>9.10</td>
<td>9.03</td>
<td>9.02</td>
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<tr>
<td>Amplitude (Nm)</td>
<td>83.49</td>
<td>84.58</td>
<td>86.58</td>
<td>87.87</td>
<td>88.77</td>
</tr>
<tr>
<td>Average torque (Nm)</td>
<td>56.24</td>
<td>53.00</td>
<td>51.29</td>
<td>50.24</td>
<td>49.53</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>0.5 mm</th>
<th>1.0 mm</th>
<th>1.5 mm</th>
<th>2.5 mm</th>
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<tbody>
<tr>
<td>Net flow (cm³)</td>
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<td>0.068</td>
<td>0.060</td>
<td>0.048</td>
</tr>
<tr>
<td>Flow ripple (L/min)</td>
<td>6.60</td>
<td>6.50</td>
<td>6.72</td>
<td>6.98</td>
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<tr>
<td>Amplitude (Nm)</td>
<td>85.63</td>
<td>84.58</td>
<td>85.11</td>
<td>87.47</td>
</tr>
<tr>
<td>Average torque (Nm)</td>
<td>58.51</td>
<td>53.00</td>
<td>58.56</td>
<td>62.55</td>
</tr>
</tbody>
</table>
Funding
This work is supported by the National Natural Science Foundation of China (no. 51075360), by the Fundamental Research Funds for the Central Universities, and the National Key Technology R&D Program of the twelfth Five-year Plan of China (no. 2011BAF09B05).

References

Appendix

Notation

\( d_p \) piston diameter (mm)
\( d_{pb} \) diameter of fixed orifices for hydrostatic bearing of piston bore (mm)
\( d_{sb} \) diameter of fixed orifices for hydrostatic bearing of slipper bore (mm)
\( l_p \) guidance length of piston (mm)
\( l_{pb} \) length of fixed orifices for hydrostatic bearing of piston bore (mm)
\( l_{sb} \) length of fixed orifices for hydrostatic bearing of slipper bore (mm)
\( n \) number of pistons
\( r_{sp} \) inner radius of slipper (mm)
\( A_{ex} \) flow area between piston chamber and extra module (mm²)
\( A_{hp} \) flow area between piston chamber and delivery kidney (mm²)
\( A_{lp} \) flow area between piston chamber and suction kidney (mm²)
\( C \) flow coefficient
\( F_{Ak} \) axial forces applied on swash plate by the pressure in the piston chamber (mm)
\( Ke \) bulk modulus of hydraulic oil (MPa)
\( M_{sx} \) swash-plate torque about x-axis (Nm)
\( M_t \) shaft torque due to one piston (N·m)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$P$</td>
<td>pressure in piston chamber (MPa)</td>
<td></td>
</tr>
<tr>
<td>$Pa$</td>
<td>oil viscosity (Pas)</td>
<td></td>
</tr>
<tr>
<td>$P_c$</td>
<td>pressure in pump shell (MPa)</td>
<td></td>
</tr>
<tr>
<td>$P_h$</td>
<td>pressure in delivery port (MPa)</td>
<td></td>
</tr>
<tr>
<td>$P_l$</td>
<td>pressure in suction port (MPa)</td>
<td></td>
</tr>
<tr>
<td>$Q_d$</td>
<td>delivery flow of piston chamber (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_l$</td>
<td>leakage of piston chamber (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_{lp}$</td>
<td>leakage of piston/cylinder pair (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_{ls}$</td>
<td>leakage of slipper/swash-plate (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_{iv}$</td>
<td>leakage of cylinder/valve-plate (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_n$</td>
<td>suction flow of pump (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_s$</td>
<td>suction flow into piston chamber (L/min)</td>
<td></td>
</tr>
<tr>
<td>$Q_t$</td>
<td>output flow of pump (L/min)</td>
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</tr>
<tr>
<td>$R_1$</td>
<td>inner radius of inner sealing belt (mm)</td>
<td></td>
</tr>
<tr>
<td>$R_2$</td>
<td>outer radius of inner sealing belt (mm)</td>
<td></td>
</tr>
<tr>
<td>$R_3$</td>
<td>inner radius of outer sealing belt (mm)</td>
<td></td>
</tr>
<tr>
<td>$R_4$</td>
<td>outer radius of outer sealing belt (mm)</td>
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</tr>
<tr>
<td>$R_c$</td>
<td>piston pitch radius (mm)</td>
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<tr>
<td>$R_{sp}$</td>
<td>outer radius of slipper (mm)</td>
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</tr>
<tr>
<td>$V_{pc}$</td>
<td>volume of piston chamber (cm$^3$)</td>
<td></td>
</tr>
<tr>
<td>$\alpha_f$</td>
<td>wrap angle of piston chamber (deg)</td>
<td></td>
</tr>
<tr>
<td>$\beta$</td>
<td>inclined angle of swash plate (deg)</td>
<td></td>
</tr>
<tr>
<td>$\delta_1$</td>
<td>average clearance height of piston pair (µm)</td>
<td></td>
</tr>
<tr>
<td>$\delta_2$</td>
<td>average gap height of slipper pair (µm)</td>
<td></td>
</tr>
<tr>
<td>$\delta_3$</td>
<td>average gap height of valve-plate pair (µm)</td>
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</tr>
<tr>
<td>$\varepsilon$</td>
<td>eccentricity of piston in cylinder bore</td>
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</tr>
<tr>
<td>$\rho$</td>
<td>density of hydraulic oil (kg/m$^3$)</td>
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</tr>
<tr>
<td>$\varphi$</td>
<td>rotation angle of shaft (rad)</td>
<td></td>
</tr>
<tr>
<td>$\omega$</td>
<td>angular velocity of shaft (rad/s)</td>
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</tr>
</tbody>
</table>