Design of A Novel Electro-Hydraulic Rotary Valve with Continuous Adjustable Rated Flow

Y. Sang*, X. Wang, & X. X. Li

School of Mechanical Engineering, Dalian University of Technology, Liaoning 116024, China, *Email: dlsdp@sina.com

ABSTRACT: Servo valve with smaller rated flow can guarantee the static control precision and servo valve with larger rated flow can guarantee the dynamic response in the electro-hydraulic servo control system. In order to obtain static high-precision control and good dynamic response, more than one servo valve with different rated flow must be used in the system. However, the extra spare servo valves are more expensive, frequent replacement of the servo valves may bring contaminating problem. A novel electro-hydraulic rotary valve with continuous adjustable rated flow (adjustable-area gradient) is proposed to solve this problem in this paper, which is easy to change flow gain and can realize the functions of throttling and directional control. The structure and operating principle of the novel electro-hydraulic rotary valve have been explained in detail. Its steady-state flow torque has been analyzed by establishing the mathematical model. The simulations for steady-state flow torque and flow characteristic have been carried out. The result shows that valve sleeve and spool structure have opposite effects on steady-state flow torque and flow. Four kinds of optimized scheme have been analyzed through simulation. This paper will provide a reference to develop the adjustable-area gradient rotary electro-hydraulic valve.

KEYWORDS: Rotary valve; Operating principle; Steady-state flow torque; Flow characteristic; Structure optimization.

INTRODUCTION

The servo valve is the most important component in the hydraulic servo system. The types of the hydraulic servo valves with different rated flows are the same, but their bandwidths are different. High dynamic response can be obtained by using the servo valve with large rated flow, but the static control precision obtained is usually low. This problem could be found in the common testing machines. Contradiction among rated flow, dynamic response and static control precision is not easy to be solved. A typical example happens in the geotechnical field. The hydraulic servo system is adopted in the dynamic tri-axial apparatus, which is a very important tool to test the strength and deformation characteristics of the soil. In order to study the deformation characteristics of soil samples, the dynamic tri-axial apparatus needs to load different amplitude forces/stresses on the soil samples [1]. Some samples are soft like sand, others are hard like clay. The deformation of the soft specimen is relative much bigger than that of the hard specimen under the same force/stress, which will need more flow. Servo valve with small rated flow could secure high-precision static loading but small motion range and vice versa. Different rated-flow servo valves need to be prepared for the geotechnical test in consideration of different samples. Only in this way, the ideal test data with high accuracy of static and dynamic force/stress could be obtained simultaneously.

However, frequent change of different servo valves is easy to contaminate the oil conduits, even causing malfunction of hydraulic servo valve. Some engineers figured out a possible way to solve this problem. Several different rated-flow servo valves in parallel are installed in the system and the servo valves with suitable rated flows are chosen by switching solenoid valve to the right position. But it is not worth applying this method in most occasions due to the larger structure, trivial switch, difficult control and higher cost.

Nowadays, there are two main structure forms to adjust flow in the hydraulic system: the slide valve and the rotary valve [2-4]. In contrast to the slide valve, the rotary valve is easier to obtain high frequency characteristic, which can realize the control of small flow and has high resolution. Besides, the motion of rotary valve spool has no acceleration zero drift and has high control precision. The rotary valve can be directly driven by the servo or step motor so that it has a compact structure. Furthermore, the rotary valve is less vulnerable to contamination [5-7].

Consequently, in combination with the characteristics and the advantages of the rotary valve, this paper proposes a novel electro-hydraulic rotary valve with continuous adjustable rated flow, which has an adjustable-area gradient.

DOI: 10.7508/jmerd.2016.03.005
First, the paper elaborates its structure principle and specific implementation plan. Then, the simulation analysis of steady-state flow torque and flow characteristic is conducted. Based on the results, four kinds of optimized structure scheme are proposed and further analyzed via simulation at last.

STRUCTURE AND OPERATING PRINCIPLE OF THE NOVEL ROTARY VALVE

Structure of the Novel Rotary Valve

The novel rotary valve is proposed in this paper, which is mainly composed of a valve body, a sleeve and a rotary spool. The essential components of the novel rotary valve are shown in Figure 1. There is a longitudinal bore that defines five axially spaced-apart annular chambers through the valve body. The chambers communicate respectively with the hydraulic tank via port T1 and port T2, the fluid input/output line via port A and port B, the pressurized fluid source via port P.

![Figure 1. The essential components of the novel rotary valve.](image)

The sleeve is mounted in the longitudinal bore, which is held in place by right end cap. There are five axially spaced-apart annular grooves that are aligned with the chambers in valve body on the outer wall of the sleeve. The sleeve is perforated by two groups of valve ports. Each group is comprised of the first, the second and the third set of valve ports. Each set has two ports, which are circumferentially distributed evenly around the wall of the sleeve at predetermined angular position. The ports of the second set of each group are designated as free-flow, always-open ports and the ports of the other two sets are named as metering ports. The solid structure of sleeve is shown in Figure 2.

![Figure 2. The solid structure of the valve sleeve.](image)
The spool is rotatably mounted inside the sleeve. The spool has four sets of axially spaced-apart arched grooves, each set of grooves communicating through the radial orifice to assure pressure equalization. The opposite arched sealing lands between two grooves of each set perform the gating capability. Three annular equalizer-slots are machined laterally into the wall of the spool. The solid structure of spool is shown in Figure 3.

![Figure 3. The solid structure of the rotary spool.](image)

Reciprocating the spool to the first, the second or the null position through selected valve ports can measure the fluid flow and change the flow direction into or out of the chambers filled with the fluid driven by device such as a cylinder. The concrete motion of the spool is shown in Figure 4. When the valve spool rotates counterclockwise to the first angular position viewed from the right end in Figure 4, the sealing lands open the port A to port T2 and vent the other port B to port P at the same time. On the contrary, the valve spool rotates clockwise to the second angular position, the reverse function takes place. The different peak amplitudes (rated flow) are dependent on adjustable-area gradient. The peak amplitude of the force/stress wave is controlled by the linear motions of the spool. When the spool moves towards the right end as shown in Figure 4, the actual throttling area formed between the arched grooves on the spool and the rectangular ports of the sleeve will decrease.

![Figure 4. The rotary and linear motions of the spool.](image)

Operating Principle of the Novel Rotary Valve

The intact rotary valve assembly as shown in Figure 5 mainly comprises step motor 1, left connecting tube 2, left sleeve end cap 3, bearing 4, valve body 5, valve spool 6, right connecting tube 7, spring reset device 8, servo motor 9, elastic coupling 10, right sleeve end cap 11, valve sleeve 12, piston 13, limit screw 14, micrometer screw 15, differential measurement cylinder 16, etc.

The specific implementation scheme is elaborated as follows. The left end of spool is equipped with a bearing embedded on the right end of piston, while the left end of piston is connected to the micrometer screw. The limit screw through the radial pore drilled in the left end cover is embedded in V groove machined axially on the micrometer screw to allow the micrometer screw to move axially only. The left end of the micrometer screw is equipped with differential measurement cylinder, which is coupled to the step motor shaft to boost its rotation. When
the micrometer screw axially moves to push the connected piston in axial motion, the axial movement of the valve spool occurs. The valve spool driven by servo motor rotates to change the fluid flow and the direction.

**Figure 5.** The overall cross-sectional assembly stereogram of the intact rotary valve.

ANALYSIS OF STEADY-STATE FLOW TORQUE OF ROTARY VALVE

State Flow Force

When the valve spool rotates to a certain position, flow is constant [8]. Owing to the change of the flow velocity and the direction, liquid flowing into or out of the ports will give rise to the reaction forces on the valve spool as Figure 6 (The cross-section along line C-C of Figure 4) shows.

**Figure 6.** The velocity of flow into or out of sleeve ports.

In Figure 6, the part surrounded by black thick border is defined as control volume. The free-flow port area is much bigger than the metering port area. Therefore, the flow velocity $v_0$ through free-flow port is smaller than velocity $v$ through metering port so that it can be neglected. According to Bernoulli’s law, the bigger the flow rate, the lower the pressure [9]. The flow force just arises from the speed difference between metering ports and free-flow ports on the valve spool as shown in Figure 7 (The cross-section along line C-C of Figure 4). Normally jet Angle $\alpha$ is always greater than 0, so the steady-state flow torque always has the tendency to close the metering ports, thus increasing the driving torque of the valve.

Mathematical Model [10]

In order to get the biggest driving torque, calculating the steady state flow torque needs to take the maximum area gradient into account. According to the theorem of momentum, flow force $F$ generated by a single metering port is

$$F = \rho Q(v - v_0) = \rho Qv$$

(1)
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

Figure 7. The steady state flow force on the spool.

The direction of the flow force \( F \) is as shown in Figure 7, which can be decomposed into the radial component \( F_r \) and the circumferential component \( F_s \). Due to the symmetrical distribution of the metering ports, the radial components offset each other. The jet angle of the metering port is \( \alpha \) as shown in Figure 7, so the circumferential flow force \( F_s \) is

\[
F_s = F \sin \alpha = \rho Q \nu \sin \alpha
\]

(2)

where \( \nu = \frac{Q}{C_v A} \); \( A = 2R(L-x_c)\sin \frac{\theta}{2} \); \( Q = C_j A \sqrt{\frac{2\Delta p}{\rho}} \); \( C_j = C_c C_v \).

Therefore, flow \( Q \) can be expressed as:

\[
Q = 2C_j R(L-x_c)\sin \theta \sqrt{\frac{2\Delta p}{\rho}}
\]

(3)

The circumferential flow force \( F_s \) can be written as:

\[
F_s = 4C_j C_v R(L-x_c)\Delta p \sin \theta \sin \alpha \sin \alpha
\]

(4)

where \( C_j \) is the flow coefficient; \( C_v \) is the contraction coefficient; \( C_v \) is the velocity coefficient; \( A \) is the throttling area; \( \theta \) is the spool rotation angle; \( R \) is the radius of the rotary spool; \( L \) is the width of the metering port cross section; \( x_c \) is the axial displacement of spool; \( \Delta p \) is the pressure difference between the upstream and the downstream of the arched groove.

Figure 8. The upstream and downstream pressure through arched groove.

As shown in Figure 8, \( \Delta p \) can be written as:
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

\[ \Delta p = p_s - p_b = \frac{p_s - p_L}{2} \]  

(5)

where \( p_s \) is the supply pressure; \( p_b \) is the pressure of the free-flow ports; \( p_L \) is the load pressure.

Therefore, circumferential flow force \( F_c \) can be written as:

\[ F_c = 2C_d C_l R (L - x_c)(p_s - p_L) \sin \frac{\theta}{2} \sin \alpha \]  

(6)

The steady-state flow torque \( T_{io} \) induced by the fluid through a single metering port can be expressed as:

\[ T_{io} = 2C_d C_l R^2 (L - x_c)(p_s - p_L) \sin \frac{\theta}{2} \sin \alpha \]  

(7)

The rotary valve with zero opening property will have four metering ports work at the same time accompanied by the rotation of spool, so the total steady-state flow torque \( T_c \) can be expressed as:

\[ T_c = 8C_d C_l R^2 (L - x_c)(p_s - p_L) \sin \frac{\theta}{2} \sin \alpha \]  

(8)

From equation (8), it can be seen that the steady-state flow torque strongly depends on the rotation angle \( \theta \) and the jet angle \( \alpha \).

According to equations (3), (5) and (8), the unloaded flow and the steady-state flow torque can be calculated as shown in Figure 9 and Figure 10, where we take \( C_d = 0.65 \), \( C_l = 0.98 \), \( R = 0.005m \), \( L = 0.003m \), \( p_s = 21MPa \), \( p_L = 0 \), \( \alpha = 69^\circ \).

Figure 9. Theoretical values of total unloaded flow.

Figure 10. Theoretical extreme values of total steady-state flow torque.
SIMULATION FOR INTERIOR FLOW FIELD OF THE NOVEL ROTARY VALVE

The steady-state flow torque is one of the most important forces affecting the valve performance [11]. In this paper, simulation results will be compared with the theoretical values, and the steady-state flow torque and the flow change law can be discovered by analysis of the simulation so as to identify the influence of valve structure shapes on them.

Geometric Model

The geometric model of interior flow field is set up by using Pro/e software. The sizes of model are given in Figure 11. The shape and sizes of the four sets of arched groove on the spool are exactly same. In order to simplify the model, three dimensional flow field formed by a set of metering ports can be selected to build its geometric model. The geometric model of interior flow field is as shown in Figure 12.

![Figure 11. The geometric parameters of interior flow field.](image1)

![Figure 12. The geometric model of interior flow field.](image2)

Meshing Models

The different models are respectively set up with spool rotation angle at the interval of 2.5° from 0° to 30°. One of the meshing models is shown in Figure 13.
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

Figure 13. Computational grid of simulation model with 15° spool rotation angle.

Figure 14. The flow field boundary conditions.

Specifying the Flow Field Boundary Conditions

With regards to setting boundary conditions [12], both entrance boundary inlet1 and inlet2 are defined as the pressure-inlet, exit boundary outlet1, exit boundary outlet2 and the pressure-outlet. The remaining boundary is defined as the Wall. The arched groove bottom is named as wall1 to monitor torque, the rest of the walls are the wall2, as shown in Figure 14.

Simulation Parameters and Conditions Assumption

Some software like FLUENT can be applied to accomplish simulation work [13, 14]. The actual three-dimensional movement of the fluid flow field is very complex, so it is difficult to accurately simulate. Thus in the actual calculation of the computational model we make the following assumptions [15, 16].

(1) The working medium is oil and the corresponding parameters are: density $\rho$ is $872 \text{kg}/\text{m}^3$; kinematic viscosity $\mu$ is $0.0279 \text{kg} / (\text{m} \cdot \text{s})$.

(2) The entrance pressure is set to 10.5 MPa and the exit pressure is 0 MPa. Since the compressibility of oil is very small, the impact of compression can be ignored.

(3) The state of flow within the valve chamber is steady and turbulent, so the control equations of the two $k-\varepsilon$ equation models are used.

(4) Adiabatic flow and the temperature are not transmitted to other substances except the fluid.

(5) There’s no sliding surface.

SIMULATION RESULTS AND ANALYSIS
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

Figure 15. The total flow torque on the spool of different rotation angles.

Torque Simulation Results

Simulation of different spool angles is carried out to monitor steady-state flow torque on the arched groove bottom and the results are obtained (define the spool rotation direction as reference direction to torque). Because the four metering ports work at the same time, the total torque as shown in Figure 15 should be the twice of the simulation value.

From Figure 15, the simulation values of flow torque are different from the theoretical values thoroughly. When the spool turns to 12.5° and 30° respectively, steady-state flow torque reaches the extreme value and the maximum value is about ±0.33 N/m. Steady-state flow torque is 0 N/m when the spool turns to about 24°. From 24° to 30°, the direction of flow torque is consistent with spool rotation, so the torque will make the metering ports open.

Flow Simulation Results

The quality flow of two outlets could be read from the simulation and the total volume flow could be calculated. The simulation flow values compared with the theoretical calculation values are shown in Figure 16. In Figure 16, simulation results and theoretical values of flow are consistent in general, but their deviation begins to increase gradually after spool rotation angle reaching 15°.

Figure 16. Simulation results and theoretical values of flow.

Influence of Metering Port and Arched Groove Structure on Flow and Torque

In Figure 17, the low pressure zone appears only on one side rather than two sides of the arched groove. That is because jet angle $\alpha$ is affected by the bottom of arched grooves so that the jet is not complete. In Figure 18, angle formed by the theoretical throttling area and the bottom of arched groove is small so that the actual throttling area reduces when the metering ports are fully open. So the flow reduces as well. Therefore metering port and arched groove structure can be further improved based on Ref. [11], [17] and [18].

SIMULATION AND ANALYSIS OF THE IMPROVED SCHEMES

The New Scheme 1
In this scheme the direction of metering ports is changed, which is parallel to the horizontal direction. Thus the actual throttling area and the jet angle are increased at the same time. The specific sizes of simulation model are shown in Figure 19. Different models with spool rotation angle $\theta$ at the interval of 5° are conducted in the simulation. The simulation results are shown in Table 1.

![Figure 17](image17.png)

**Figure 17.** Contours of pressure when spool turns to the angle 30°.

![Figure 18](image18.png)

**Figure 18.** Partial enlargement of contours of velocity vector when spool turns to the angle 30°.

![Figure 19](image19.png)

**Figure 19.** Sizes of simulation model of new scheme 1.

**Table 1.** Simulation results of new scheme 1.
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

<table>
<thead>
<tr>
<th>$\theta$ (°)</th>
<th>Torque (N/m)</th>
<th>Flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>-0.1295</td>
<td>0.2587</td>
</tr>
<tr>
<td>10</td>
<td>-0.2348</td>
<td>0.5148</td>
</tr>
<tr>
<td>15</td>
<td>-0.2498</td>
<td>0.7487</td>
</tr>
<tr>
<td>20</td>
<td>-0.2175</td>
<td>0.9669</td>
</tr>
<tr>
<td>25</td>
<td>-0.1339</td>
<td>1.1672</td>
</tr>
<tr>
<td>30</td>
<td>0.0510</td>
<td>1.3023</td>
</tr>
</tbody>
</table>

The New Scheme 2

The bottom shapes of arched grooves are changed to guarantee the complete jet and enough throttling area in the new scheme 2. The specific sizes of simulation model are shown in Figure 20. Similarly, the different models with spool rotation angle $\theta$ at the interval of 5° are conducted in the simulation. The simulation results are shown in Table 2.

![Figure 20. Sizes of simulation model of new scheme 2.](image)

Table 2. Simulation results of new scheme 2.

<table>
<thead>
<tr>
<th>$\theta$ (°)</th>
<th>Torque (N/m)</th>
<th>Flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>-0.1643</td>
<td>0.2693</td>
</tr>
<tr>
<td>10</td>
<td>-0.1947</td>
<td>0.5012</td>
</tr>
<tr>
<td>15</td>
<td>-0.1899</td>
<td>0.7277</td>
</tr>
<tr>
<td>20</td>
<td>-0.2567</td>
<td>1.0150</td>
</tr>
<tr>
<td>25</td>
<td>-0.1881</td>
<td>1.3665</td>
</tr>
<tr>
<td>30</td>
<td>0.0867</td>
<td>1.6309</td>
</tr>
</tbody>
</table>

The New Scheme 3

In the new scheme 3, the combined effect on the flow torque and the flow is studied by combination of the new scheme 1 and the new scheme 2. The specific sizes of simulation model are shown in Figure 21. The simulation results are shown in Table 3.

![Figure 21. Sizes of simulation model of new scheme 3.](image)
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow

Table 3. Simulation results of new scheme 3.

<table>
<thead>
<tr>
<th>$\theta$ (°)</th>
<th>Torque (N/m)</th>
<th>Flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>-0.1735</td>
<td>0.2695</td>
</tr>
<tr>
<td>10</td>
<td>-0.2543</td>
<td>0.5126</td>
</tr>
<tr>
<td>15</td>
<td>-0.2753</td>
<td>0.7672</td>
</tr>
<tr>
<td>20</td>
<td>-0.2889</td>
<td>1.0492</td>
</tr>
<tr>
<td>25</td>
<td>-0.2825</td>
<td>1.4028</td>
</tr>
<tr>
<td>30</td>
<td>-0.0427</td>
<td>1.6791</td>
</tr>
</tbody>
</table>

The New Scheme 4

Increasing the size of radial orifice connecting the opposite arched grooves can apparently increase the actual throttling area. The specific sizes of simulation model are shown in Figure 22 and the simulation results are shown in Table 4.

![Figure 22. Sizes of simulation model of new scheme 4.](image)

Table 4. Simulation results of new scheme 4.

<table>
<thead>
<tr>
<th>$\theta$ (°)</th>
<th>Torque (N/m)</th>
<th>Flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>-0.0189</td>
<td>0.2434</td>
</tr>
<tr>
<td>10</td>
<td>-0.0256</td>
<td>0.4728</td>
</tr>
<tr>
<td>15</td>
<td>-0.0288</td>
<td>0.6857</td>
</tr>
<tr>
<td>20</td>
<td>-0.0405</td>
<td>0.8805</td>
</tr>
<tr>
<td>25</td>
<td>0.0180</td>
<td>0.1087</td>
</tr>
<tr>
<td>30</td>
<td>0.1089</td>
<td>1.2840</td>
</tr>
</tbody>
</table>

Analysis and Comparison of Simulation Results

![Figure 23. Comparison of all simulation flow torques.](image)
In Figure 23, steady-state flow torque values of all schemes decline at first and then increase gradually with the increase of the spool rotation angle. It can be concluded that a bigger size of radial orifice connecting the opposite arched grooves has better effective influence on the steady-state flow torque.

In Figure 24, all schemes can increase the flow to different extent. The new scheme 1 and the new scheme 4 can meet the demand closely although their steady-state flow torque is in extreme position.

CONCLUSIONS
This research has presented the physical model of a novel rotary electro-hydraulic valve, where a spool driven by a motor is rotatably fitted in a valve sleeve. The flow area gradient is controlled by another motor, thus changing the flow gain (rated flow) readily. A mathematical model is derived and a detailed simulation analysis of the steady-state flow torque and the flow characteristic of different schemes are described. It can be concluded that different structures of spool have different impacts on the valve properties. Based on the simulation results, it is easily concluded that a bigger size of radial orifice connecting the opposite arched grooves has better performance of the valve. This paper will provide the reference to develop the electro-hydraulic rotary valve.

ACKNOWLEDGMENTS
This research is supported by the National Natural Science Foundation of China through the grant number 51275068 and the support by the Fundamental Research Funds for the Central Universities Grant No. DUT15LK21.

REFERENCES
Design of a novel electro-hydraulic rotary valve with continuous adjustable rated flow


