Design & Simulation of a Hydraulic Back Pressure Valve with a Large Flow Range

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Abstract. A hydraulic back pressure valve with a large flow range up to not less than 100L/min was designed. In order to meet the requirements of low cracking pressure and large applicable flow range, the valve was designed in a pilot operated style with a flat valve port of main valve. Area difference between the front and rear of main spool was adopted to keep the valve port self-sealing and adjustable damping plugs were used to easily regulate the dynamic performance of the valve. The mathematic models of the valve were established and simulations were carried out. Then, the influences of some major structural parameters such as volume before main throttle, volume of main spring chamber, dimensions of slender holes of damping plugs and the main throttle were obtained. The simulation results show that the valve has good flow adaptability, and the pressure shock can be cut down remarkably by reasonably setting the parameters.

1. Introduction
In order to prevent seawater from penetrating into marine hydraulic systems through outboard actuators, a back pressure valve is usually connected in series in system circuits. The pressure in return tubes is set a little higher than outsider seawater pressure by the back pressure valve, and then seawater is kept outboard even when large leakage happens because of failures of outboard actuators’ seals. The back pressure valve is a branch of relief valve with a normally low cracking pressure. This paper presents a back pressure valve which operates well with a cracking pressure range of 0.5MPa to 1.5MPa, and a flow range up to not less than 100L/min. Numerical decompositions based on the physical structure of the valve were executed, mathematic models were established, and simulation work was carried out.

2. Structure design
Several efforts were made to satisfy the demands of low cracking pressure and large applicable flow range of the valve [1~5].

a. A pilot valve was applied. The pilot valve is widely used in the situations that the cracking pressure of a direct relief valve is too large to adjust manually. In this paper, in order to meet the demand of large flow, the port flow area of valve was designed to be relatively large to reduce system return flow velocity. And the manual regulation force is related to the product of the port pressure and the flow area. Generally, flowrate through pilot valve is fairly small. Therefore, the pilot valve was definitely practicable to release the manual regulation intensity of the valve.

b. The port of main valve adopted a form of flat valve. On behalf of expanding the range of applicable flowrate, it is necessary to reduce the effect of flowrate on the steady state of the spool. And the effect is embodied in the steady jet force acting on the spool. Commonly, the flow is extremely
positive correlation to steady jet force, and the latter plays a negative role in regulation accuracy of back pressure valve. According to the jet force calculation equation \( f_z = c_d \pi d x \sin 2 \alpha \cdot \Delta p \), the flat valve had a half cone angle \( \alpha \) whose value could reach theoretically 90°. Consequently, the jet force became negligible, and then the regulation accuracy could be guaranteed.

c. The sealing load of main valve was considered. Usually when the inlet flow is not large enough to move the main spool, the sealing of valve port is generally ensured by the pre-compression force of main spring. Paradoxically, the low cracking pressure requires that the pre-compression force of main valve should not be too large. The valve involved in this paper created an area difference between the front and rear of the main spool. The same inlet pressure acted on both ends of the spool simultaneously, which always made the force in the spring chamber larger than the inlet force. That is, this structure had a certain self-sealing effect.

d. The damping plug was adjustable. Damping plugs are used to compensate for the underdamping of hydraulic systems. Adjustable damping plugs did help a lot during actual manufacturing and testing. Based on the designs mentioned above, the detail sketch diagram of the back pressure valve could be obtained (as seen in Figure 1).

3. Mathematic models

Figure 2 shows the principle diagrammatic sketch of the valve. Table 1 shows the physical significances of variables referred in the equations. Some assumptions were made to simplify the mathematic models. Then the mathematic models could be established [5-8].

- Drain port pressure was negligibly small and set zero.
- Cavitations didn’t occur at the valve throttles.
- Gravity was ignored.

(1) Motion differential equation of pilot spool:

\[
p_p A_p = m_p \ddot{x}_p + f_{ap} \dot{x}_p + k_p (x_p + x_{po}) + c_p \pi d_p x_p \sin 2 \beta \cdot \Delta p
\]  

(2) Flowrate equation through pilot throttle:

\[
q_p = c_p \pi d_p x_p \sin \beta \sqrt{\frac{2}{\rho} \Delta p}
\]  

(3) Flowrate continuity equation before pilot throttle:

\[
q_b + q_s = \frac{V_m}{E} \Delta \dot{p} + q_p + A_p \dot{x}_p
\]  

(4) Flowrate equation through pilot damping plug:
\[ q_s = \frac{\pi d_s^4}{128\mu c}(p_s - p_m) \]  
\hspace{2cm} (4)

(5) Flowrate equation through main damping plug:
\[ q_s = \frac{\pi d_s^4}{128\mu b}(p_m - p_s) \]  
\hspace{2cm} (5)

(6) Flowrate continuity equation of the main spring chamber:
\[ A_m \dot{x}_m = q_s + \frac{V_v}{E} \dot{p}_m \]  
\hspace{2cm} (6)

(7) Motion differential equation of main spool:
\[ p_m A_m - p_s A_s = m_m \dot{x}_m + f_m \dot{x}_m + k_m (x_m + x_m^0) + c_m \pi d_m \sin 2\alpha \cdot p_m \]  
\hspace{2cm} (7)

(8) Flowrate equation through main throttle:
\[ q_m = c_m \pi d_m x_m \sin \frac{2}{\rho} p_m \]  
\hspace{2cm} (8)

(9) Flowrate continuity equation before main throttle:
\[ q_m = q_b + V_v \dot{p}_m + A_m \dot{x}_m \]  
\hspace{2cm} (9)

Table 1. Variables mentioned in the equations above.

| Symbol | Description |
|--------|-------------|
| \( A_m \) | Effective area of main throttle |
| \( A_p \) | Effective area of pilot throttle |
| \( A_s \) | Effective area of main spring chamber |
| \( c_m \) | Flow coefficient of main throttle |
| \( c_p \) | Flow coefficient of pilot throttle |
| \( d_b \) | Diameter of main damping plug |
| \( d_c \) | Diameter of pilot damping plug |
| \( d_m \) | Diameter of main throttle |
| \( d_p \) | Diameter of pilot throttle |
| \( E \) | Bulk modulus of oil |
| \( f_m \) | Viscous coefficient of main spool |
| \( f_p \) | Viscous coefficient of pilot spool |
| \( k_m \) | Stiffness of main spring |
| \( k_p \) | Stiffness of pilot spring |
| \( l_b \) | Length of main damping plug |
| \( l_c \) | Length of pilot damping plug |
| \( m_m \) | Effective mass of main spool |
| \( m_p \) | Effective mass of pilot spool |
| \( p_m \) | Pressure before main throttle |
| \( p_p \) | Pressure before pilot throttle |
| \( p_s \) | Pressure in main spring chamber |
| \( q_b \) | Flowrate through main damping plug |
| \( q_m \) | Flowrate through main throttle |
| \( q_p \) | Flowrate through pilot throttle |
| \( q_s \) | Flowrate through pilot damping plug |
| \( \dot{x}_m \) | Displacement of main spool |
| \( \dot{x}_p \) | Displacement of pilot spool |
| \( x_m^0 \) | Pre-compression of main spring |
| \( x_p^0 \) | Pre-compression of pilot spring |
| \( \alpha \) | Half cone angle of main spool |
| \( \beta \) | Half cone angle of pilot spool |
| \( \mu \) | Dynamic viscosity of oil |
| \( \rho \) | Density of oil |

Eliminate the derivation items in the equations above and then the static state \( p_m \) was obtained.

\[ p_m = \frac{k_m (x_m + x_m^0)}{A_m - c_m \pi d_m \sin 2\alpha} + \frac{A_m \cdot k_p (x_p + x_p^0)}{(A_m - c_m \pi d_m x_m \sin 2\alpha)(A_p - c_p \pi d_p x_p \sin 2\beta)} \]  
\hspace{2cm} (10)

According to the structure design results, \( \alpha \) was considered 90°. That is, \( A_m \gg c_m \pi d_m x_m \sin 2\alpha \). Then equation (10) could be rewritten as below.

\[ p_m = \frac{k_m (x_m + x_m^0)}{A_m} + \frac{A_m \cdot k_p (x_p + x_p^0)}{A_p - c_p \pi d_p x_p \sin 2\beta} \]  
\hspace{2cm} (11)

Suppose \( x_m^0 \gg x_m, x_p^0 \gg x_p \), and \( A_p \gg c_p \pi d_p x_p \sin 2\beta \), then the cracking pressure of the valve was able to be estimated as below:
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\[ p_m = \frac{k_m \cdot x_{m0}}{A_m} + \frac{A_m}{A_p} \frac{k_p \cdot x_{p0}}{A_p} \]  

(12)

4. Simulation analysis

AMESim was adopted for the simulation analysis. Figure 3 shows the simulation diagrammatic.

4.1. Flow adaptability analysis

The performances of pressure before main throttle under a large range of flowrate through the main throttle were analysed. The simulated excitation flowrate \(q_{in}\) varied every 0.5 seconds from 10L/min to 20L/min, 30L/min, 50L/min, 70L/min, 90L/min, 100L/min gradually. The results are shown in Figure 4. Table 2 shows the specific data gained from the analysis.

| \( q_{in}/L\cdot min^{-1} \) | Stage1 | Stage2 | Stage3 | Stage4 | Stage5 | Stage6 | Stage7 | Max | Min | Average |
|-----------------------------|--------|--------|--------|--------|--------|--------|--------|-----|-----|---------|
| 10                          | 0.4965 | 0.5001 | 0.4965 | 0.5025 | 0.5027 | 0.5038 | 0.5040 | 0.5040 | 0.4965 | 0.5009 |
| 20                          | 1.0015 | 1.0020 | 0.9994 | 1.0002 | 1.0027 | 1.0037 | 1.0039 | 1.0039 | 0.9994 | 1.0019 |
| 30                          | 1.5003 | 1.5004 | 1.5006 | 1.5009 | 1.5014 | 1.5018 | 1.5019 | 1.5019 | 1.5003 | 1.5010 |

\( p_m \) curves converge very fast. The settling time of each excitation is less than 0.1 seconds and the values of steady \( p_m \) show good stability. Suppose the fluctuation of \( p_m \) could be computed as below:

\[ k = \frac{Max(p_m) - Min(p_m)}{Average(p_m)} \times 100\% \]  

(13)

Then the fluctuations of steady \( p_m \) could be described quantitatively by 1.505%, 0.455% and 0.111%, which indicates the valve has good flow adaptability. Meanwhile, pressure shock that can’t be ignored happens at every excitation. The phenomenon of pressure shock is more prominent in the case of step large flowrate input (as seen in Figure 5 and Figure 6) than gradual flowrate input. It can be clearly seen that the larger the step flow, the more significant the pressure shock. It is kind of unaccepted if the pressure shock is as high as 4.8MPa when the cracking pressure is set 0.5MPa.
4.2. Parameters influence analysis

The purpose of analysing the influences of structural parameters on characteristics of the back pressure valve was to reduce the instantaneous pressure shock. Several main parameters that may have greater influences on the dynamic characteristics of the valve were initially selected.

4.2.1. Influences by inlet volume before main throttle $V_{in}$

The volume before main throttle contained the volume of return tubes upstream of the valve. The longer return tube, the larger volume $V_{in}$. Figure 7 shows the step dynamic responses of the valve. As seen, the steady $p_m$ are almost the same under different $V_{in}$ conditions, but the instantaneous pressure shock is obviously lowered down by larger $V_{in}$ without any conspicuous extension of settling time. The peak of pressure shock is cut down from 4.801MPa to 2.52MPa.

4.2.2. Influences by volume of main spring chamber $V_s$

The main spring chamber where the main spring is deployed locates between the main spool and the pilot damping plug. Figure 8 shows the step dynamic responses of the valve. It is also found that there are no visible changes of the steady $p_m$ under several circumstances. And the settling time behaves the similar characteristic. The peak of pressure shock is cut down to 3.737MPa. Note that $V_s$ is directly related to the main spring structure and the whole dimension of the valve. Therefore, it should be reasonably designed.
4.2.3. Influences by diameters of slender holes of damping plugs \( d_0 \)
Assume the main and pilot damping plugs have the same diameter of slender holes. That is \( d_b = d_c = d_0 \). In general, the smaller the damping hole, the larger the damping and the greater the pressure shock, the easier the system should converge. The results shown in Figure 9 and figure 10 are good testimonies to this view. Figure 10 shows partial details of the simulation results, and it is clearly seen that oscillation occurs when \( d_0 \) reaches 2.0mm, which means it is not convergent. The peak of pressure shock is cut down to 3.573MPa when \( d_0 = 1.2 \text{mm} \).

4.2.4. Influences by diameter of main throttle \( d_m \)
While analysing the influences of the diameter of the main throttle, in order to ensure that the cracking pressure did not change greatly, the diameter ratio of the front and rear chambers of the main spool \( A_m / A_r \) was kept constant. Figure 10 shows changes of \( p_m \) when \( d_m \) changes. As seen, \( d_m \) is slightly negative correlation to the pressure shock. The peak of pressure shock is only cut down to 4.448MPa by sacrificing part of the flow capacity of the back pressure valve.

4.3. Optimization results
Final simulations are shown in Figure 12 by applying the analysis results gained from chapter 4.2.1 to chapter 4.2.4. The peak of pressure shock is noticeably cut down from 4.801MPa to 1.693MPa with a faintly extension of settling time when input step flow \( q_{in} \) is set 100L/min.

5. Conclusions
Generally, some understandings are obtained for designing of the back pressure valve.
The back pressure has good flow adaptability and can maintain good pressure stability when inlet flowrate changes from 10L/min to not less than 100L/min.

The volumes of chambers play effective roles in decreasing the pressure shock. But the dimensions of the valve should be considered at the same time. It is tentative to place an accumulator with enough buffer volume in front of the valve.

The diameters of slender holes of damping plugs play positive roles in bringing down the pressure shock. Oversize or undersize would cause divergence or high pressure shock.

Although the optimization methods are effective, it is undeniable that there is still a large pressure shock when the input step flow is kind of large.

Further research and experimental work are recommended to be carried out.

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