Structural Optimization of a Water Hydraulic Pilot-Operated Pressure-Reducing Valve

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Abstract. The characteristics of a pilot-operated pressure-reducing valve for water hydraulics are analysed. The valve operates well with a primary pressure of up to 15MPa, a secondary pressure range of 10MPa to 14MPa, and a flow range of 20L/min to 40L/min. Optimizations are applied to reduce the cavitation noise of the valve and to improve the pressure regulation accuracy of secondary pressure. The first optimization is executed by reducing pressure drop and velocity through the throttle of main spool and the second optimization is carried out by lowering down the steady jet force on both main spool and the pilot spool. The dynamic characteristics of the secondary pressure are obtained by simulations. The results show the valve behaviours better with the improvement structure. The average variation of the secondary pressure is brought down from 0.308MPa to 0.175MPa under different primary pressure conditions and from 0.288MPa to 0.136MPa under different inlet flowrate conditions. The velocity through throttle of the main spool is 31.18% cut down on average.

1. Introduction

Constant pressure output pressure-reducing valve is a kind of valve which decreases the system pressure for load use and is capable to keep the reduced pressure stable when system input pressure or flowrate fluctuates in a certain range. This paper presents a pilot-operated pressure-reducing valve for water hydraulics, which operates well with a primary pressure of up to 15MPa, a secondary pressure range of 10MPa to 14MPa, and a flow range of 20L/min to 40L/min [1~3]. Figure 1 illustrates the original structure details of the valve. Figure 2 shows the dynamic characteristics of secondary pressure of the valve under different circumstances (p_{outlet} is set at about 10MPa). The secondary pressure variations of the valve are shown in Table 1.

| q_{inlet} | p_{outlet} variations (p_{inlet}=15MPa → 11MPa) | p_{outlet} | p_{outlet} variations (q_{inlet}=40L/min → 20L/min) |
|----------|---------------------------------|-----------|---------------------------------|
| ~40 L/min | 0.446MPa | 15MPa | 0.456MPa |
| ~30 L/min | 0.265MPa | 13MPa | 0.233MPa |
| ~20 L/min | 0.211MPa | 11MPa | 0.176MPa |
| Average  | 0.308MPa | Average | 0.288MPa |
Where $p_{outlet}$ is the pressure at outlet; $p_{inlet}$ is the pressure at inlet; $q_{inlet}$ is the flowrate through inlet. As known, water hydraulic components have more severe cavitation noise issues because of water’s lower viscosity, lower compressibility and higher density compared to oil hydraulic components. This paper focuses on weakening these problems by reducing pressure drop and fluid velocity through the throttles of the pressure-reducing valve. And of course, the pressure regulation accuracy is equally significant during the optimizations.

2. Structural optimization & analysis

2.1 Cavitation noise optimization

The structure of two-serial-throttles is widely applied in hydraulic components to reduce pressure drop and fluid velocity through each throttle and the effectiveness is verified [4~5]. It is considerable that where the two-serial-throttles should be applied to between the main valve and the pilot valve. It is decided the velocity is prior because the pressure loss is determined and always changed by outside load. The velocity through throttle can be estimated by the formula as follows.

$$v = \frac{q}{A_{\text{spool}}} = \frac{q}{\pi dx \sin \alpha}$$

(1)

Where $q$ is the flowrate through the throttle; $d$ is the diameter of spool; $x$ is the displacement of spool; $\alpha$ is the half cone angle of spool. Then the velocities are computed in Table 2.
Table 2. Velocities of main valve and pilot valve under different circumstances.

| q_{inlet} | Main valve | Pilot valve |
|-----------|------------|-------------|
|           | P_{inlet}  | p_{outlet}  | P_{inlet}  | p_{outlet}  | P_{inlet}  | p_{outlet}  | P_{inlet}  | p_{outlet}  | P_{inlet}  | p_{outlet}  |
|           | 15MPa      | 15MPa       | 13MPa      | 13MPa      | 11MPa      | 11MPa      |
| q_{inlet} |            |             |            |            |            |            |
| 40L/min   | 82.94 m/s  | 5.40 m/s    | 5.40 m/s   | 20.02 m/s  | 25.79 m/s  | 29.94 m/s  |
| 30L/min   | 81.09 m/s  | 20.02 m/s   | 20.02 m/s  | 22.59 m/s  | 24.56 m/s  |
| 20L/min   | 82.10 m/s  | 25.79 m/s   | 25.79 m/s  | 24.56 m/s  | 29.94 m/s  |

It is obviously seen velocity through the throttle of main valve is larger than that of pilot valve, especially when pressure drop between $p_{inlet}$ and $p_{outlet}$ is larger. Then the main valve is chosen to apply the two-serial-throttles. Figure 3 illustrates the modified simulation structure, and Figure 4 shows the change of $p_{outlet}$ under different circumstances ($p_{outlet}$ is set at about 10MPa).

![Figure 3. Simulation diagrammatic of the valve](image)

![Figure 4. Change of $p_{outlet}$](image)

Then the secondary pressure variations of the valve could be also obtained, which are shown in Table 3.

Table 3. $p_{outlet}$ variations under different circumstances.

| q_{inlet} | $p_{outlet}$ variations (P_{inlet}=15MPa→11MPa) | P_{inlet} | $p_{outlet}$ variations (q_{inlet}=40L/min→20L/min) |
|-----------|-----------------------------------------------|----------|-------------------------------------------------|
| ~40 L/min | 0.367MPa                                      | 15MPa    | 0.247MPa                                       |
| ~30L/min  | 0.282MPa                                      | 13MPa    | 0.259MPa                                       |
| ~20L/min  | 0.307MPa                                      | 11MPa    | 0.187MPa                                       |
There are two noteworthy things. The first is $p_{\text{middle}}$ (pressure at the middle of the two-serial-throttles, shown in Figure 3) is almost half of sum of $p_{\text{inlet}}$ and $p_{\text{outlet}}$, which means the pressure drop through each throttle is almost half cut down. The second noteworthy thing is that the $p_{\text{outlet}}$ variation is enlarged in most cases except ultimate pressure drop and flow rate situations. The results indicate that two-serial-throttles functions perfectly in large pressure drop applications.

2.2 Accuracy regulation optimization

The aim of accuracy regulation is to control the variations of secondary pressure under different circumstances. The secondary pressure of pilot-operated pressure-reducing valve can be commonly evaluated by the formula as follow [6].

$$p_{\text{outlet}} = \frac{k_m(x_{\text{inlet}} + x_{\text{max}} - x_m)}{A_m} + \frac{k_p(x_p + x_p)}{A_p - c_p \pi d_p x_p \sin 2\alpha_p} - \frac{q \cos \alpha_p}{A_m} \sqrt{2\rho (p_{\text{outlet}} - p_{\text{inlet}})}$$

(2)

Where subscript $m$ or $p$ refers to parameters of main valve or pilot valve; $k$ is the spring stiffness; $x_0$ is the pre-compression of spring; $x$ is the displacement of valve spool when it reaches the stable status; $x_{\text{max}}$ is the maximum allowable displacement of valve spool; $A$ is the sectional area of valve spool; $c_p$ is the flow coefficient; $d$ is the diameter of valve spool; $q$ is the flow rate through the throttle; $\alpha$ is the half cone angle of spool.

The first item on the right side of equation (2) represents the influence the main spring has on the valve, the second item the pilot spring, and the third item the steady jet force on main spool. Then steps could be adopted to enhance the stabilities of the valve.

**Step 1.** Noticed that the third item functions negatively against the valve springs, and it is unpredictable when the valve is in-service. It can be concluded that lowering down the steady jet force helps the secondary pressure to stabilize when pressure or flow rate excitations take place. Some research work on steady jet force was carried out and methods were proved effective on decreasing the steady force [6–8].

A specific shape of main spool is adopted to compensate the jet force by generating a negative force [6]. The negative force can counteract part of the jet force produced by flow through rectangle shoulder of the throttles. The detail shape should be gained by experiments, and then the compensation is pretty fine.

A large angle cone structure is attached to the pilot spool. The half cone angle $\beta$ is designed $\sim 80^\circ$. In general conditions, the jet force can be calculated by the equation $f_{\text{jet}} = \rho q v \cos \alpha$, and $\beta$ functions as replacing the spool angle $\alpha$. Therefore, the larger $\beta$ is, the smaller $f_{\text{jet}}$ is. Research shows the steady jet force can be greatly cut down by this method [8].

The following Figure 5 illustrates the detail optimization structure.

**Step 2.** Compared to equation (2), in the two-serial-throttles situation, the formula should be improved.

$$p_{\text{outlet}} = \frac{k_m(x_{\text{inlet}} + x_{\text{max}} - x_m)}{A_m} + \frac{k_p(x_p + x_p)}{A_p - c_p \pi d_p x_p \sin 2\alpha_p} - \frac{q \cos \alpha_p}{A_m} \sqrt{2\rho (p_{\text{outlet}} - p_{\text{inlet}})}$$

(3)

Assumed $p_{\text{middle}}=(p_{\text{inlet}}+p_{\text{outlet}})/2$, then equation (3) can be rewritten as follows.

$$p_{\text{outlet}} = \frac{k_m(x_{\text{inlet}} + x_{\text{max}} - x_m)}{A_m} + \frac{k_p(x_p + x_p)}{A_p - c_p \pi d_p x_p \sin 2\alpha_p} - \frac{\sqrt{2} q \cos \alpha_p}{A_m} \sqrt{2\rho (p_{\text{outlet}} - p_{\text{inlet}})}$$

(4)
It is found that the third item is enlarged by $2^{\frac{1}{2}}$ times and it provides a strong evidence to support the results in Table 1 and Table 3. It explains well why secondary pressure variation is enlarged by simply adopting the two-serial-throttles when the other parameters, like the flowrate through the throttles, are the same.

As seen in equation (4), $x$, $q$ and $\sqrt{p_{in}-p_{out}}$ (pressure drop through throttle) are main variable factors on secondary pressure accuracy regulation. In order to keep $p_{outlet}$ stable, it is necessary to bring down the proportions of these factors. There are also two valid ways to accomplish the target by enlarging the pre-compression of springs $x_0$ or scaling up the sectional area of valve spool $A$. In general structural design, $x_m$ and $x_p$ are already greatly larger than $x_m$ and $x_p$ respectively. Therefore, in this case, the latter way is adopted by increasing diameter of main spool from 18mm to 20mm and diameter of pilot spool from 3mm to 4mm.

2.3 Simulation analysis

The final simulation diagrammatic of the valve is nearly the same as the diagrammatic shown in Figure 3. Several improvements are executed. The jet force coefficients are simply set at a fairly low level based on efforts made in chapter 2.1. Spring stiffness $k_m$ and $k_p$ are proportionally increased with the increasing sectional area $A_m$ and $A_p$. Figure 5 illustrates the final structure of the pressure-reducing valve and Figure 6 shows the dynamic characteristics of secondary pressure of the modified valve under different circumstances ($p_{outlet}$ is set at about 10MPa).

![Diagrammatic of the valve after optimization](image)

Figure 5. Diagrammatic of the valve after optimization

![Change of $p_{outlet}$](image)

Figure 6. Change of $p_{outlet}$
Then the secondary pressure variations of the modified structure could be obtained, which are shown in Table 4.

| \( q_{\text{inlet}} \) | \( p_{\text{outlet}} \) variations \( (p_{\text{inlet}}=15\text{MPa} \rightarrow 11\text{MPa}) \) | \( p_{\text{inlet}} \) | \( p_{\text{outlet}} \) variations \( (q_{\text{inlet}}=40\text{L/min} \rightarrow 20\text{L/min}) \) |
|----------------|--------------------------------------------------------------------------------|--------|--------------------------------------------------------------------------------|
| ~40 L/min     | 0.173MPa                                                                         | 15MPa  | 0.145MPa                                                                         |
| ~30L/min      | 0.207MPa                                                                         | 13MPa  | 0.145MPa                                                                         |
| ~20L/min      | 0.145MPa                                                                         | 11MPa  | 0.117MPa                                                                         |
| Average       | 0.175MPa                                                                         | Average| 0.136MPa                                                                         |

Comparing Table 4 with Table 1, it is found the average variation of the secondary pressure is brought down from 0.308MPa to 0.175MPa under different primary pressure conditions and from 0.288MPa to 0.136MPa under different inlet flowrate conditions. It indicates that the accuracy regulation optimization is quite effective. Meanwhile, it is also found that the dynamic qualities, such as pressure overshoots, response times, adjustment times by stimulating and so on, are better than acceptable from Figure 6.

The velocities through main throttles are computed by equation (1) and shown in Table 5. Table 6 shows the reduced proportions of calculated velocities compared Table 5 with Table 2.

| \( q_{\text{inlet}} \) | \( p_{\text{inlet}}=15\text{MPa} \) | \( q_{\text{inlet}}=30\text{L/min} \) | \( q_{\text{inlet}}=20\text{L/min} \) |
|----------------|----------------------------------|----------------------------------|----------------------------------|
| \( p_{\text{inlet}} \) | 56.61 m/s                        | 56.64 m/s                        | 56.21 m/s                        |
| 15MPa           | 43.28 m/s                        | 42.99 m/s                        | 42.57 m/s                        |
| 13MPa           | 22.90 m/s                        | 22.18 m/s                        | 21.56 m/s                        |

The results indicate the velocity through throttle of the main spool is 31.18% cut down on average. It proves that the fluid velocity through the throttle is notably reduced by applying the two-serial-throttles structure. Meanwhile, it is also found that the velocities are still at a high level. That is, the cavitation noise is controlled but still exists.

### 3. Conclusions

Generally some understandings are obtained for designing of the pressure-reducing valve.

- The pilot-operated pressure-reducing valve has a better performance after structural optimization.
- The two-serial-throttles structure functions positively on decreasing the pressure drop and fluid velocity through spool throttles. It helps a lot in controlling valve’s cavitation and noise.
- Steady jet force plays a negative role in accuracy regulation of pressure-reducing valve. Lowering down the jet force can raise the secondary pressure regulation accuracy.
- The diameters of main spool and pilot spool play positive roles in accuracy regulation of pressure-reducing valve. But the dimensions of the valve should be reasonably considered at the same time.

Further experimental work will be carried out to verify these conclusions mentioned above.
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