

Characteristic analysis of a water hydraulic pilot-operated pressure-reducing valve

Xuyao Mao¹, Junhua Hu¹, Chao Wu¹, Yiou Liu¹ and Yinshui Liu²

¹ Wuhan Second Ship Design and Research Institute, Wuhan, China

² Huazhong University of Science and Technology, Wuhan, China

E-mail: dreamforty@126.com; bobyhu@126.com; wuchaowhut@163.com;
liuyou.hust@163.com; liuwater@tom.com

Abstract. Comprehensive characteristics of a seawater hydraulic pilot-operated pressure-reducing valve with constant pressure output were analyzed. A rated pressure of 15MPa and a rated flowrate of 40L/min were offered in the numerical work. Static and dynamic analyses show good behaviors: The settling time is less than 0.2s, the output pressure variation is about 0.3MPa at the maximum when input pressure or flowrate is flucturing, and the steady external leakage is below 0.025L/min. The pilot spring regulates the output pressure and the main spring has an ability to adjust the output pressure variation faintly. The narrow hole diameter of the adjustable damping plugs is negatively related to the respond time. And appropriately raising the spring chamber volume can evidently reduce outlet pressure impact of the valve when input mutations happen.

1. Introduction

Pressure-reducing valve are commonly used for pressure control of oil or water hydraulic systems and studies have been made of their static and dynamic characteristics [1-4]. Constant pressure pressure-reducing valve is one kind of these valves. The valve decreases the system pressure for load use and the reduced pressure is able to keep stable when the system pressure or flowrate fluctuates in a certain range. This paper presents some researches on the static and dynamic characteristics of a pilot-operated constant pressure pressure-reducing valve by utilizing water as fluid media. Figure 1 illustrates the structural details of the valve.

Water is highly corrosive, and also a strong electrolyte. Both anticorrosion and anti-couple corrosion were taken into consideration on component material selections. Major component materials are shown in Table.1. Stainless steel was widely used combined with NBR and engineering plastics.

Vibration and hydraulic shock are admittedly issues for hydraulic component and system development. Hydraulic systems are mostly under damping. The problems are amplified when the fluid becomes water because of its lower viscosity, lower compressibility and higher density. Therefore two damping plugs were settled to increase the system damping, one of which was inside the main spool and the other in front of pilot throttle.

Numerical decompositions based on the physical structure of the valve were executed, mathematic model was established, and then simulation work was carried out. The results were pretty suggestive on optimal design and improvement of the pressure-reducing valve.



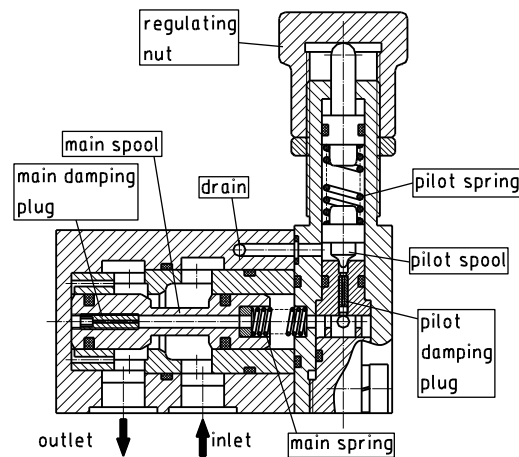


Figure 1. Cross-section of the pressure-reducing valve

Table 1. Main component material list

Title	Material	Title	Material
main valve body	1Cr18Ni9Ti	main damping plug	QA19-4
main spool	17-4PH	main spool sealing	wear-resisting plastic+NBR
main spring	Ni36CrTiAl	pilot valve body	1Cr18Ni9Ti
pilot damping plug	1Cr18Ni9Ti	pilot spool	17-4PH
pilot spring	Ni36CrTiAl	other sealing	NBR

2. Mathematic models

The following assumptions were made to simplify the mathematic model [5-8]. Figure 2 shows the principle diagrammatic sketch of the valve. Most of the variables and corresponding physical significances were also displayed.

- The jet force of main spool was partly considered by multiplying a jet force coefficient (the value was set 0.5).
- Flow through narrow holes of the two damping plugs was regarded as laminar.
- The viscous friction and windage force of main spool were ignored.
- Drain port pressure was negligibly small and set zero.
- Cavitations didn't occur at the valve throttles.
- Gravity was ignored.

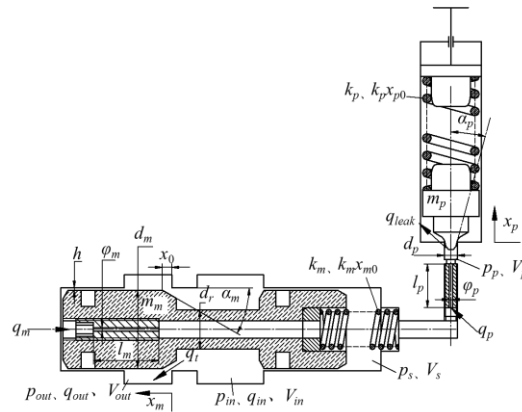


Figure 2. Diagrammatic of the pressure-reducing valve

1. Motion differential equation of pilot spool

$$m_p \ddot{x}_p + k_p (x_p + x_{p0}) = p_p \cdot A_p - f_{jetp} \cdot c_d \pi d_p \sin 2\alpha_p \cdot x_p \cdot p_p \quad (1)$$

2. Flowrate equation through pilot throttle

$$q_{leak} = c_q \pi d_p \sin \alpha_p \cdot x_p \sqrt{\frac{2p_p}{\rho}} \quad (2)$$

3. Flowrate continuity equation before pilot throttle

$$q_p = q_{leak} + A_p \dot{x}_p + \frac{V_p}{E} \dot{p}_p \quad (3)$$

4. Flowrate equation through pilot damping plug

$$q_p = \frac{\pi \phi_p^4}{128 \mu l_p} (p_s - p_p) \quad (4)$$

5. Flowrate continuity equation of the main spring chamber

$$q_m = q_p - A_m \dot{x}_m + \frac{V_s}{E} \dot{p}_s \quad (5)$$

6. Flowrate equation through main damping plug

$$q_m = \frac{\pi \phi_m^4}{128 \mu l_m} (p_{out} - p_s) \quad (6)$$

7. Flowrate continuity equation of outlet

$$q_t = q_{out} + q_m + A_m \dot{x}_m + \frac{V_{out}}{E} \dot{p}_{out} \quad (7)$$

8. Flowrate equation of the main throttle

$$q_{out} = c_q \pi d_m \sin \alpha_m \cdot \sqrt{x_m^2 + h^2} \cdot \sqrt{\frac{2(p_{in} - p_{out})}{\rho}} \quad (8)$$

9. Flowrate continuity equation of inlet

$$q_{in} = q_t + \frac{V_{in}}{E} \dot{p}_{in} \quad (9)$$

10. Motion differential equation of main spool

$$m_m \ddot{x}_m - \text{sgn}(\dot{x}_m) f_{fric} + k_m (x_{m0} + x_0 - x_m) = p_{out} A_m - p_s A_m + f_{jetm} \cdot c_d \pi d_m \sin 2\alpha_m \cdot x_m (p_{in} - p_{out}) \quad (10)$$

Where f_{jetp} and f_{jetm} were the jet force coefficients of pilot and main throttles, f_{fric} was the friction between main spool and the valve pocket, c_d was the flow velocity coefficient, c_q was the flow rate coefficient, and E was water bulk modulus.

$$\text{And } A_p = \frac{\pi}{4} d_p^2, A_m = \frac{\pi}{4} d_m^2.$$

Eliminate the derivation items in the equations above and then the static state equations are obtained. Suppose $x_m \ll x_0 + x_{m0}$, $x_p \ll x_{p0}$, $A_p \ll f_{jetp} c_d \pi d_p \sin 2\alpha_p$, then the output pressure is able to be estimated as:

$$p_{out} \approx \frac{k_m (x_{m0} + x_0)}{A_m} + \frac{k_p x_{p0}}{A_p} \quad (11)$$

3. Analysis

Simulation analysis was divided into two parts. One was the basic static and dynamic characteristics, and the other was influences of some typical variables. AMESim was adopted for the simulations (as seen in Figure 3).

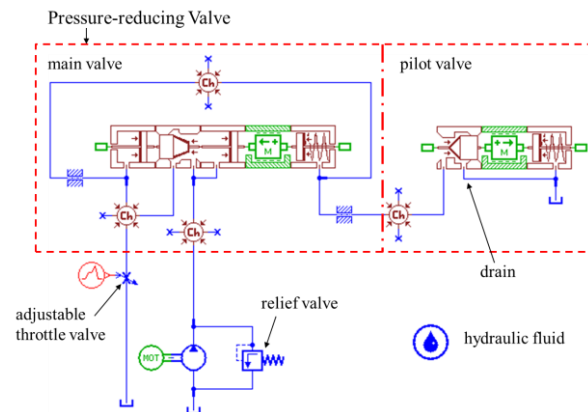


Figure 3. AMESim diagrammatic of simulation

3.1. Primary Characteristic Analysis

Pressure stability, flowrate stability, settling time and external leakage were selected as representations of primary static and dynamic characteristics of the valve. A step flowrate was given as the original input, and the pressure and the flowrate on the valve were regulated by a relief valve and an adjustable throttle valve. Figure 4~Figure 7 were different performances of p_{out} , q_t , q_{leak} and x_m when simulation conditions changed. Openings mentioned in these figures were related to throttle valve relative openings. The outlet pressure p_{out} was set at about 10MPa.

p_{out} curves converge very fast. The first step excitation settling time is less than 0.2s when instantaneous p_{out} fluctuation is below 3% of steady p_{out} , and posterior settling time is even less than 0.025s. The initial poppet lift, that is, the initial opening of main spool, is 2.2mm. But the main spool locates much nearer to the throttle when posterior excitations happen (The displacements of main spool against the throttle are shown in Figure 7). That explains it takes longer time for the main spool to reach the steady position at the first excitation.

The flowrate q_t curves are almost overlapping under different inlet pressure p_{in} , but q_t changes as the relative opening of throttle valve changes. That is, relative opening is almost one-to-one correspondence to q_t . Then the steady p_{out} variation behaviors are obtained in Table 2. It is found that p_{out} variation is about 0.3MPa at the maximum at a large range of p_{in} and q_t . The pressure-reducing valve has good flowrate stability and good pressure stability. Note that values of column 4 are much smaller than those of column 2, which means the valve has better flowrate stability.

The theoretical pressure regulating range could be 0.79MPa ~ 14.7MPa by simulation. On account of structural limitations and actual applications, p_{out} is recommended to be set among 10MPa~14MPa.

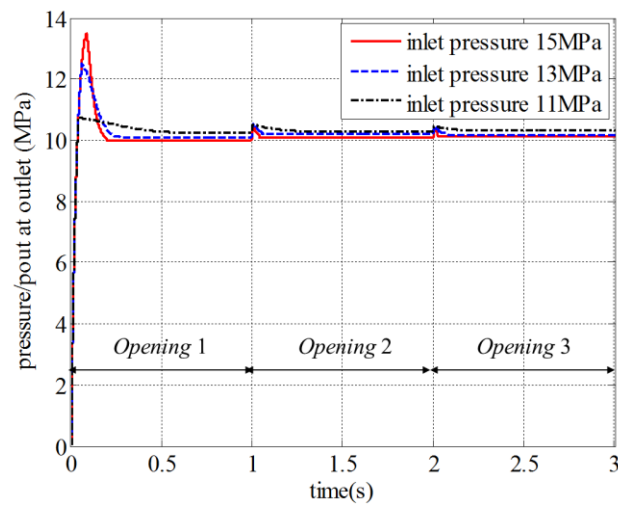


Figure 4. Change of pressure p_{out} at outlet

Table 2. Table of stability

q_t	p_{out} variation at different p_{in} (15MPa→11 MPa)	p_{in}	p_{out} variation at different q_t (40 L/min →26.5L/min)
~40L/min	0.270MPa	15MPa	0.116MPa
~31.4L/min	0.238MPa	13MPa	0.088MPa
~26.5L/min	0.226MPa	11MPa	0.072MPa

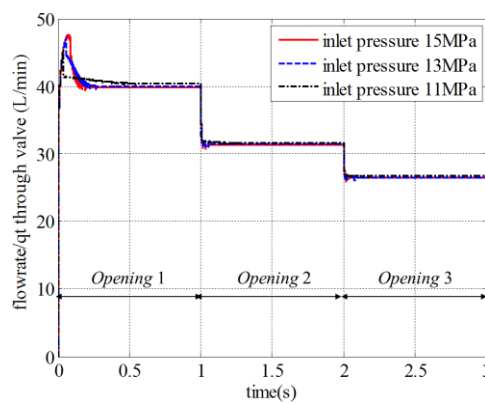


Figure 5. Change of flowrate q_t through valve

Figure 6 shows external leakage q_{leak} through drain port under different circumstances. The trends of all curves are similar: Pulse leakages occur accompanied with mutations of input. The peak leakages happen at the first step excitations. Nearer p_{out} to p_{in} , larger q_{leak} ; smaller q_t , larger q_{leak} . The steady external leakage is below 0.025L/min.

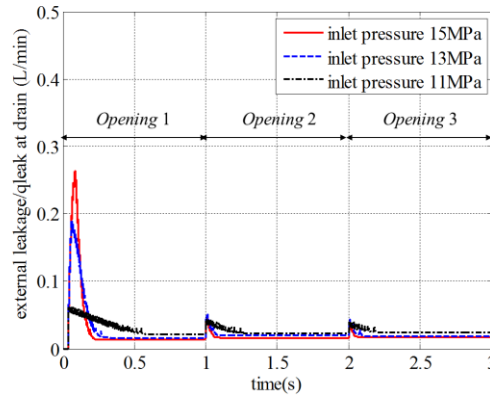


Figure 6. Change of external leakage q_{leak} at drain

Figure 7 indicates that the opening of the main throttle is adaptive to make sure a constant p_{out} as long as p_{in} and q_t is waving in a specific range. It helps to prove the good flowrate and pressure stability of the pressure-reducing valve.

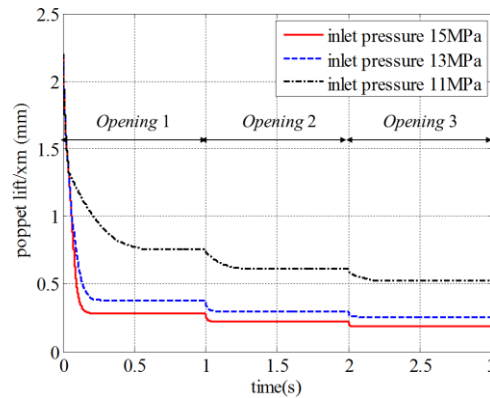


Figure 7. Change of poppet lift x_m of main spool

3.2. Variables Influence Analysis

3.2.1. Influences by pilot spring precompression force. As known, the pilot spring is designed to adjust p_{out} tangibly. In Equation (11), usually $A_p \ll A_m$ because $d_p \ll d_m$. In this case, A_m is 36 times larger than A_p . That is, it is more effective to regulate p_{out} by adjusting pilot spring parameters than main spring parameters. The precompression force and the stiffness of pilot spring determine the output to a great extent.

Figure 8 and Figure 9 respectively show the pressure stability and the flowrate stability of the valve with different pilot spring precompression force f_{p0} . It is observed that p_{out} follows the change of precompression force quite well. The pressure stability and the flowrate stability of the valve are undifferentiated. Note that the settling time is a little longer when p_{out} is closer to p_{in} . It could be predicted that the stiffness of pilot spring has seldom effect on the steady behaviors as long as f_{p0} is constant according to equation (11).

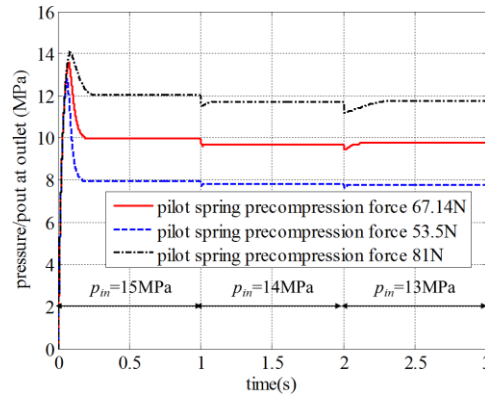


Figure 8. Changes of pressure p_{out} with constant $q_t=40\text{L/min}$

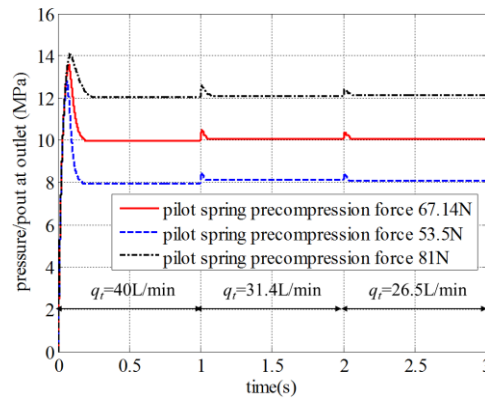


Figure 9. Changes of pressure p_{out} with constant $p_{in}=15\text{MPa}$

3.2.2. Influences by main spring precompression force. If the pilot spring functions as coarse adjustment, the main spring's role could be regarded as fine adjustment. Figure 10 and Figure 11 respectively show the pressure stability and the flowrate stability of the valve with different main spring precompression force f_{m0} . As seen, a high f_{m0} raises steady p_{out} slightly, which could be predicted from equation (11); the pressure variations are enlarged when f_{m0} is oversize or undersize; besides, the settling time is noticeably lengthened when f_{m0} is undersize, especially when the differential pressure between p_{in} and p_{out} is small. The stiffness of main spring k_m plays a similar role in pressure adjustment according to equation (11).

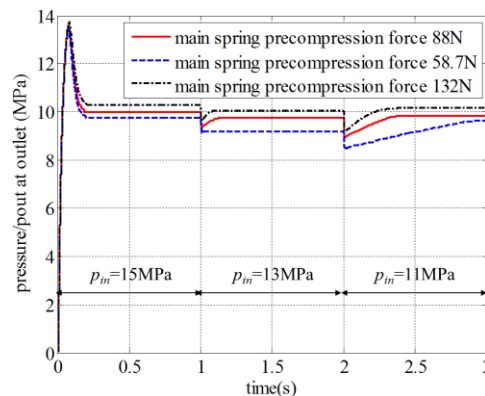


Figure 10. Changes of pressure p_{out} with constant $q_t=40\text{L/min}$

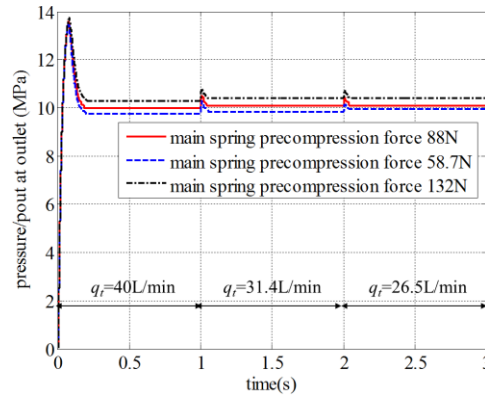


Figure 11. Changes of pressure p_{out} with constant $p_{in}=15\text{MPa}$

3.2.3. Influences by narrow hole diameter of damping plugs. The narrow hole diameters of two damping plugs were set at a same value. As seen in Figure 12~Figure 13, if the diameter is undersize, the instantaneous pressure impact would be high and the settling time would be apparently long; if the diameter is oversize, the output pressure is sine shaped waving when it reaches the stable state and the stability becomes a little worse with a larger pressure variation. Namely there exists a locally optimal solution of the narrow hole diameter of the damping plugs. According to equation (4) and equation (6), it could be predicted that the narrow hole length (l_m , l_p) functions inversely but much more ineffectively as hole diameter does.

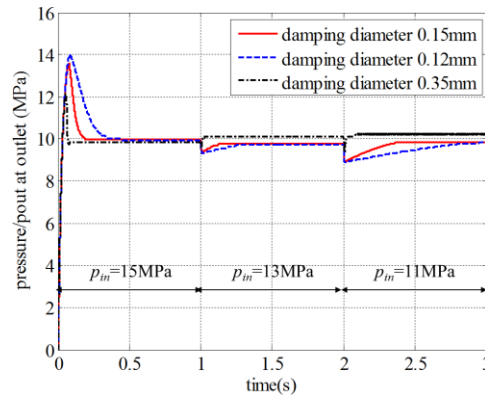


Figure 12. Changes of pressure p_{out} with constant $q_t=40\text{L/min}$

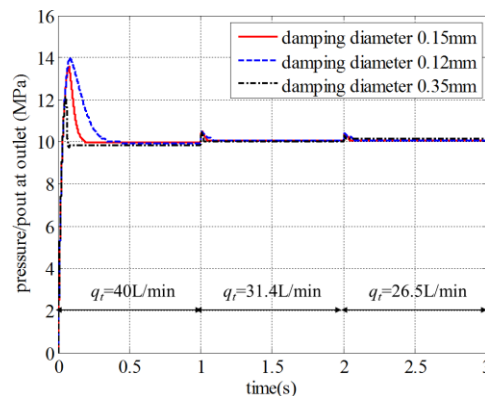


Figure 13. Changes of pressure p_{out} with constant $p_{in}=15\text{MPa}$

3.2.4. Influences by spring chamber volume. The spring chamber where the main spring was deployed located between two damping plugs. Figure 14 shows the pressure stability of the valve with different

spring chamber volumes V_s and Figure 15 the flowrate stability. As seen, the steady p_{out} is almost the same under several circumstances. Figure 14 indicates that instantaneous pressure impact when the input leaps is obviously lowered down by larger V_s without any extension of settling time. 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. Figure 15 also indicates that p_{out} disperses a little when the volume changes. It illustrates V_s affects more the flowrate stability than the pressure stability.

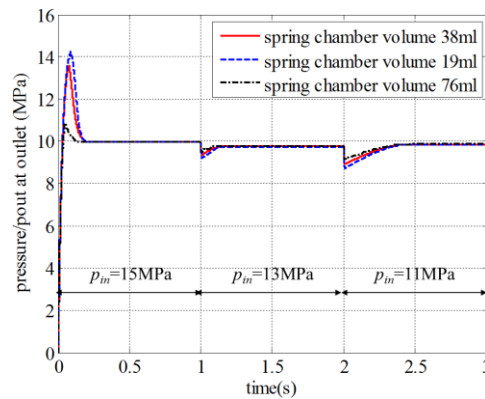


Figure 14. Changes of pressure p_{out} with constant $q_t=40\text{L/min}$

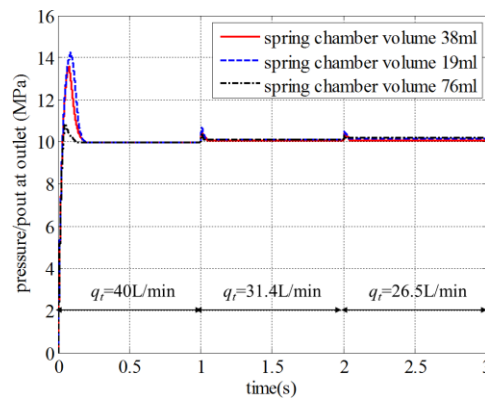


Figure 15. Changes of pressure p_{out} with constant $p_{in}=15\text{MPa}$

4. Conclusions

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

- The valve has good pressure stability (output pressure variation under 3%) and a good flowrate stability (output pressure variation under 1.2%).
- It is recommended to enlarge the differential pressure between inlet pressure and set output pressure when the valve is used.
- There exist local optimal solutions to the parameters of damping plugs and springs for better characteristics of the valve. Oversize or undersize will deteriorate the behaviors of the valve.
- The instantaneous pressure impact could be brought down conspicuously by increasing the spring chamber volume, but the dimensions of the valve should be reasonably considered at the same time.

Some experimental work was carried out and the conclusions above were well verified [9].

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