Load Boundary Analysis of the Load-sensing Hydraulic System with Multi-way Valve and Pressure-compensated Valve

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Abstract. According to the analysis on the working principle of the load-sensing hydraulic system with multi-way valve and constant-difference relief pressure-compensated valve in the load-sensing circuit, it is concluded that there are two working conditions during the operation of the system, load-sensing condition and constant pressure supply condition. Without considering the effect of balance valves, the load boundaries of the forward and backward motion of two conditions were derived, as well as the whole load boundary. The selection principle of balance valves was studied and analyzed to expand the load boundaries. The above conclusions can be applied to this type of load-sensing hydraulic system with any valve-controlled cylinder structure and also provide the theoretical guidance for the design and apply of the system.

1. Introduction

The load-sensing hydraulic system can adjust its own oil supply pressure and flow according to changes of load pressure and flow, so as to match the output power of the system with the power required to drive the load, which can reduce ineffective pressure and flow loss, and improve efficiency compared with traditional hydraulic systems [1, 2]. At present, the load boundary of the traditional constant pressure oil supply hydraulic system is relatively confirmed, but there are few studies on the load boundary of the load-sensing hydraulic system [3]. This study [4, 5] took the shuttle valve load-sensing hydraulic system as the research object and analyzed the load boundary of the system under various working conditions. The conclusions of this research can be the effective guidance of the system design to prevent the occurrence of overpressure and cavitation problems in the cylinder. The shuttle valve load-sensing hydraulic system has the problem of failure of load-sensing function because the shuttle valve is used as the selection element of feedback pressure, resulting in false high oil supply pressure and excrecent pressure loss. However, the multi-way valve load-sensing hydraulic system adopts multi-way valve as its selective component, so the oil supply pressure can always follow the change of the load pressure, which solves this problem and improves the efficiency [6]. At present, there are few researches on the load boundary of the multi-way valve load-sensing hydraulic system and the selection principle of the balance valve.

Therefore, this paper intends to take the load-sensing hydraulic system with multi-way valve and pressure-compensated valve as the research object, and analyze the load boundary and the selection principle of balance valves under various working conditions during the operation of the system. The
research results will provide theoretical guidance for the design of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve and the selection of balance valves to improve the design and application of this type of load-sensing hydraulic system.

2. Load-sensing Hydraulic System with Multi-way Valve and Pressure-compensated Valve

2.1 Working principle of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve

The schematic diagram of a typical load-sensing hydraulic system with multi-way valve and pressure-compensated valve is shown in Figure 1. Compared with the traditional constant pressure oil supply hydraulic system, the hydraulic system adds a load-sensing circuit composed of a proportional multi-way valve 8 and a constant-difference relief pressure-compensated valve 7. When the system oil supply pressure does not exceed the set pressure $P_{max}$ of the relief valve 5, its size is adjusted by the pressure-compensated valve. The function of the multi-way valve is to select the working chamber pressure of the hydraulic cylinder 10 as the feedback pressure and lead it to the spring side oil chamber of the pressure-compensated valve. Inside the pressure-compensated valve, the feedback pressure from the multi-way valve and the spring force of the pressure-compensated valve will be dynamically balanced with the system oil supply pressure. And the system oil supply pressure is determined by the feedback pressure and the spring force. Since the spring force of the pressure-compensated valve remains constant after adjustment, the oil supply pressure of the system will follow the feedback pressure (essentially the working chamber pressure) with a constant pressure difference $\Delta P$, so as to achieve self-adaptation to the load and reduce the pressure loss of the system. In addition, the variable frequency oil supply is used by the system. Therefore, the oil supply flow can be adjusted according to the working flow required by the system to realize the self-adaptation to the working flow and reduce the flow loss.

![Figure 1. Schematic diagram of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve](attachment:image.png)

1-quantitative pump; 2-variable frequency motor; 3-filter; 4-check valve; 5-relief valve; 6-solenoid valve; 7-constant-difference relief pressure-compensated valve; 8-proportional multi-way valve; 9-balance valve; 10-asymmetric hydraulic cylinder; 11-displacement sensor
2.2 Working condition of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve

According to the working principle of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve, the oil supply pressure of the system is determined by the pressure-compensated valve and the relief valve. Specifically, when the system moves forward, the working chamber of the hydraulic cylinder is the non-rod chamber, and the oil return chamber is the rod chamber. At this time, the feedback pressure from the multi-way valve is the pressure of the non-rod chamber. If $P_1 + \Delta P \leq P_{\text{max}}$, the oil supply pressure is adjusted by the pressure-compensated valve and it is $P_1 + \Delta P$. It’s only higher than the working chamber. When the system is at the load-sensing working condition, the oil supply pressure is changed along with the feedback pressure. If $P_1 + \Delta P > P_{\text{max}}$, the oil supply pressure will be determined by the set pressure of the relief valve and it is $P_{\text{max}}$. When the system is in working condition of constant pressure oil supply, the load-sensing function does not work. When the system moves in the reverse direction, there are the two working conditions mentioned above. In summary, the system oil supply pressure can be expressed as $P_S = \min\{P_1 + \Delta P, P_{\text{max}}\}$. When the system is working forward, the non-rod chamber pressure $P_1$ is operated as feedback pressure ($i = 1$). When the system is working backward, the rod chamber pressure $P_2$ is operated as feedback pressure ($i = 2$).

3. Load Boundary

The asymmetric hydraulic cylinders are mostly used for load-sensing hydraulic system with multi-way valve and pressure-compensated valves. The structure diagram of the valve-controlled asymmetric cylinder is shown in Figure 2. The direction indicated by the arrow in the figure is the forward direction of each parameter.

![Figure 2. Schematic diagram of valve-controlled asymmetric cylinder structure](image)

Usually, the oil return pressure of the system is small and we can suppose $P_0 = 0$. When the system is working forward ($x \geq 0$), the valve port flow equation is

$$Q_1 = C_d W_1 x \sqrt{2(P_2 - P_1)/\rho}, \quad Q_2 = C_d W_2 x \sqrt{2P_2/\rho}$$  \hspace{1cm} (1)

When the system is working backward ($x < 0$), the valve port flow equation is

$$Q_1 = C_d W_1 x \sqrt{2P_1/\rho}, \quad Q_2 = C_d W_2 x \sqrt{2(P_2 - P_1)/\rho}$$  \hspace{1cm} (2)

In equation (1) and (2), $Q_1$ and $Q_2$ respectively is the flow of non-rod chamber and rod chamber in hydraulic cylinder; $W_1$, $W_2$, $W_3$ and $W_4$ are the area gradient of valve port ($W_1 = W_2 = W_3 = W_4$); $P_1$ and $P_2$ respectively is the pressure of non-rod chamber and rod chamber; $\rho$ is the oil density; $P_{\text{max}}$ is the system oil supply pressure; $P_{\text{max}}$ is the set pressure of relief valve; $x$ is the valve opening; $C_d$ is the flow coefficient of valve port.
To ensure that the deduced results can be applied to load-sensing hydraulic systems with any valve-controlled cylinder structure, and to facilitate subsequent deductions, we make

\[ n = A_i / A_x, \quad 0 < n \leq 1 \quad \text{and} \quad m = W_i / W_x = W_i / W_x, \quad n \leq m \leq 1 \]

The load pressure and flow of the system are defined as

\[ P_1 = P_1 - nP_2, \quad Q_x = Q_x / n \quad (3) \]

When the system is working forward \((x \geq 0)\), from simultaneous equation (1) and (3), the equation is

\[ P_1 = \frac{m^2 P_i + n^3 P_x}{m^2 + n^3}, \quad P_2 = \frac{n^2 \left( P_x - P_1 \right)}{m^2 + n^3} \quad (4) \]

When the system is working backward \((x < 0)\), from simultaneous equation (2) and (3), the equation is

\[ P_2 = \frac{m^2 P_1 + n^3 P_x}{m^2 + n^3}, \quad P_2 = \frac{n^2 P_x - n^3 P_1}{m^2 + n^3} \quad (5) \]

### 3.1 Forward load boundary analysis

Without considering the function of balance valves, the system is working forward \((x \geq 0)\). According to the working principle of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve, there will be load-sensing and constant pressure oil supply condition in the system working process. These two working conditions will be discussed separately below.

#### 3.1.1 Load-sensing condition

When the system is working forward, the feedback pressure is the non-rod chamber pressure \(P_1\). When \(P_1 + \Delta P \leq P_{max}\), the oil supply \(P_2\) is adjusted by pressure-compensated valve. When the system is in load-sensing working condition \(P_2 = P_1 + \Delta P\). By substituting it into equation (4), the equation got is

\[ P_1 = \frac{m^2 P_i + n^3 \Delta P}{m^2 + n^3}, \quad P_2 = \frac{n^2 \left( P_{max} - P_1 \right)}{m^2 + n^3} \quad (6) \]

Under this working condition, the pressure in two chambers shall follow:

\[ 0 < P_1 \leq P_{max} - \Delta P, \quad 0 < P_2 < P_{max} \quad (7) \]

From simultaneous equation (6) and (7), the load pressure \(P_L\) shall be in below range:

\[ P_L \in \left( -\frac{n^3}{m^2} \Delta P, P_{max} - \frac{m^2 + n^3}{m^2} \Delta P \right) \quad (8) \]

#### 3.1.2 Constant pressure supply condition

If \(P_1 + \Delta P > P_{max}\), the oil supply pressure \(P_1\) is adjusted by relief valve. When the system is in condition of constant pressure oil supply, \(P_2 = P_{max}\). By substituting it into equation (4), the equation got is

\[ P_1 = \frac{m^2 P_i + n^3 P_{max}}{m^2 + n^3}, \quad P_2 = \frac{n^2 \left( P_{max} - P_1 \right)}{m^2 + n^3} \quad (9) \]

Under this working condition, the pressure of two chambers shall be:

\[ P_{max} - \Delta P < P_1 < P_{max}, \quad 0 < P_2 < P_{max} \quad (10) \]

From simultaneous equation (9) and (10), the load pressure \(P_L\) shall be in below range:

\[ P_L \in \left( P_{max} - \frac{m^2 + n^3}{m^2} \Delta P, P_{max} \right) \quad (11) \]

Take the union set of the load boundaries in the sum. When the system is running in the forward direction, the load boundary is

\[ P_L \in \left( -\frac{n^3}{m^2} \Delta P, P_{max} \right) \quad (12) \]
When the load pressure changes within the above range, the pressure in the non-rod chamber and rod chamber are

\[ P_1 \in (0, P_{\text{max}}), \quad P_2 \in \left[ 0, \frac{n^2}{m^2} \Delta P \right] \]  

(13)

### 3.2 Backward load boundary analysis

Without considering the function of balance valve, the system is working backward \((x_c < 0)\), there will be load-sensing and constant pressure oil supply condition in the system. These two working conditions will be discussed separately below.

#### 3.2.1 Load-sensing condition

When the system is working backward, the feedback pressure is the rod chamber pressure \(P_2\).

If \(P_2 + \Delta P \leq P_{\text{max}}\), the oil supply pressure \(P_S\) is adjusted by pressure-compensated valve. When the system is in load-sensing working condition, \(P_2 = P_2 + \Delta P\). By substituting it into equation (5), the equation got is

\[ P_1 = \frac{m^2}{n^2} \Delta P, \quad P_2 = \frac{m^2 \Delta P - n^2 P_L}{n^2} \]  

(14)

Under this working condition, the pressure in two chambers shall follow:

\[ 0 < P_1 < P_{\text{max}}, 0 < P_2 \leq P_{\text{max}} - \Delta P \]  

(15)

From simultaneous equation (14) and (15), the load pressure \(P_L\) shall be in below range:

\[ P_L \in \left[ -nP_{\text{max}} + \frac{m^2 + n^2}{n^2} \Delta P, \frac{m^2}{n^2} \Delta P \right], P_{\text{max}} > \frac{m^2}{n^2} \Delta P \quad \text{and} \quad P_L \in \mathbb{R}, P_{\text{max}} \leq \frac{m^2}{n^2} \Delta P \]  

(16)

#### 3.2.2 Constant pressure supply condition

If \(P_2 + \Delta P > P_{\text{max}}\), the oil supply pressure \(P_S\) is adjusted by relief valve. When the system is in condition of constant pressure oil supply, \(P_S = P_{\text{max}}\). By substituting it into equation (5), the equation got is

\[ P_1 = \frac{m^2 (P_2 + nP_{\text{max}})}{m^2 + n^2}, \quad P_2 = \frac{m^2 P_{\text{max}} - n^2 P_L}{m^2 + n^2} \]  

(17)

Under this working condition, the pressure in two chambers shall follow:

\[ 0 < P_1 < P_{\text{max}}, P_{\text{max}} - \Delta P < P_2 < P_{\text{max}} \]  

(18)

From simultaneous equation (17) and (18), the load pressure \(P_L\) shall be in below range:

\[ P_L \in \left[ -nP_{\text{max}}, -nP_{\text{max}} + \frac{m^2 + n^2}{n^2} \Delta P \right], P_{\text{max}} > \frac{m^2}{n^2} \Delta P \quad \text{and} \quad P_L \in \left[ -nP_{\text{max}}, \frac{m^2 (1 - n) + n^3}{m^2} P_{\text{max}} \right], P_{\text{max}} \leq \frac{m^2}{n^2} \Delta P \]  

(19)

Take the union set of the load boundaries in the sum. When the system is running in the backward direction, the load boundary is

\[ P_L \in \left[ -nP_{\text{max}}, \frac{m^2}{n^2} \Delta P \right], P_{\text{max}} > \frac{m^2}{n^2} \Delta P \quad \text{and} \quad P_L \in \left[ -nP_{\text{max}}, \frac{m^2 (1 - n) + n^3}{m^2} P_{\text{max}} \right], P_{\text{max}} \leq \frac{m^2}{n^2} \Delta P \]  

(20)

When the load pressure changes within the above range, the pressure in the non-rod chamber and rod chamber are

\[ P_1 \in \left[ 0, \frac{m^2}{n^2} \Delta P \right], \quad P_2 \in (0, P_{\text{max}}) \]  

(21)

### 3.3 Whole load boundary analysis

The resulting forward and backward load boundary ensure that the pressure of the hydraulic cylinder is in range \((0, P_{\text{max}})\) when the system is moving forward and backward. The overpressure or cavity erosion
will not exist in system. In order to ensure the safe operation and normal operation of the whole system, 
the whole load boundary of the system should be the intersection of the forward and backward load 
boundary. Take the union set of the load boundaries in the sum and the range is 
\[
\left\{ \begin{array}{l}
P_L \in \left[ -\frac{n^3}{m^2} \Delta P, P_{\text{max}} \right] \\
P_L \in \left[ -\frac{n^3}{m^2} \Delta P, P_{\text{max}} \right] 
\end{array} \right. 
\]
\[
\cap \left\{ \begin{array}{l}
-\frac{n P_{\text{max}} m^2}{n^3} \Delta P < m^2 \Delta P \\
-\frac{n P_{\text{max}} m^2}{n^3} \Delta P < m^2 \Delta P 
\end{array} \right. 
\]
\[
\cup \left\{ \begin{array}{l}
m \leq m^2 \Delta P \\
m \leq m^2 \Delta P 
\end{array} \right. 
\]
\[
(22) 
\]

It can be simplified as below:
\[
P_L \in \left( -\frac{n^3}{m^2} \Delta P, \frac{m^2}{n^2} \Delta P \right), P_{\text{max}} > \frac{m^2}{n^2} \Delta P \quad \text{and} \quad P_L \in \left( -\frac{n^3}{m^2} \Delta P, \frac{m^2}{n^2} \Delta P \right), P_{\text{max}} \leq \frac{m^2}{n^2} \Delta P 
\]
\[
(23) 
\]

The above equation is the load boundary of the load-sensing hydraulic system with multi-way valve 
and pressure-compensated valve without considering the action of balance valves, and the load pressure 
can ensure that there will not be the problem of overpressure and cavity erosion in two chambers when 
the hydraulic cylinder is moving forward and backward. To improve the efficiency of the load-sensing 
hydraulic system, the set pressure difference \( \Delta P \) of pressure compensator is usually small, 
from 0.5~2MPa. Because \( m \) and \( n \) is similar, for normal hydraulic system, the set pressure of relief 
valve can meet \( P_{\text{max}} > \frac{m^2}{n^2} \Delta P \). The whole load boundary of the system is 
\[
\left( -\frac{n^3}{m^2} \Delta P, \frac{m^2}{n^2} \Delta P \right) 
\]

### 3.4 Case Analysis

The load-sensing hydraulic system with multi-way valve and pressure-compensated valve adopts a 
symmetrical valve-controlled asymmetrical cylinder structure, the parameters of the system are \( m = 1 \), 
\( n = 0.8 \), \( \Delta P = 1.0\text{MPa} \) and \( P_{\text{max}} = 10\text{MPa} \). By substituting these parameters into equation (8), (11), (12), 
(16), (19), and (20), the equation got is the load boundary of the system shown in Table 1.

From Table 1., the system adopts a valve-controlled asymmetric cylinder structure, so there is 
asymmetrical movement of the system in forward and backward directions, and the load boundaries of 
forward and backward are significantly different. When moving forward, the load boundary of the 
system is (-0.51MPa, 10MPa), the forward load is bigger than backward load; when moving backward, 
the load boundary of the system is (-8MPa, 1.56MPa), the backward load boundary is bigger than the 
forward load boundary. When the system is moving forward and backward, it can withstand a large 
forward load, but the ability to withstand a backward load (excess load) is weak. Since the whole load 
boundary of the system is the intersection of the forward and backward load boundaries, its range is 
smaller. The whole load boundary of the above system is only (-0.51MPa, 1.56MPa). The load pressure 
changes within this range to ensure that the pressure of the two chambers of the hydraulic cylinder 
within the range \((0, P_{\text{max}})\) regardless of the operating conditions. But the load boundary is obviously too 
small, far from reaching the design carrying capacity of system.

| Load boundary            | Working forward | Working backward |
|--------------------------|-----------------|------------------|
| Load-sensing condition   | (-0.51MPa, 8.49MPa) | [-5.64MPa, 1.56MPa) |
| Constant pressure oil supply condition | (8.49MPa, 10MPa) | (-8MPa, -5.64MPa) |
| Unidirectional load boundary | (-0.51MPa, 10MPa) | (-8MPa, 1.56MPa) |
| Whole load boundary      | (-0.51MPa, 1.56MPa) |
4. Selection principles of Balance Valves
The above conclusions about the load boundary are all obtained without considering the effect of balance valves. The analysis shows that without the addition of balance valves, the load boundary of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve is extremely small, which can hardly satisfy the normal system work. When the load is too large, the two chambers of the hydraulic cylinder are prone to problems of overpressure or cavitation. In order to improve the carrying capacity of the system and expand the load boundary of the system, it is necessary to add balance valves at the inlet and outlet ports of the hydraulic cylinder [7, 8]. At present, the reasonable selection of the balance valve for the load-sensing hydraulic system with multi-way valve and pressure-compensated valve is not confirmed, which is necessarily to be studied.

The balance valve model is used from literature [7]. If the influence of the back pressure of the balance valve return oil is ignored, the relationship between the pressures can be simplified as

\[ k_1P_1 + P_2 = P_{\text{vl}} \geq 0 \quad \text{and} \quad k_2P_2 + P_1 = P_{\text{vl}} < 0 \] (24)

In the equation, \( k_1 \) and \( k_2 \) respectively are the conversion coefficients of the balance valve control port at the oil port of the non-rod chamber and the rod chamber, which is essentially the area ratio of the balance valve control port and the throttle port; \( P_{\text{vl}} \) and \( P_{\text{vl}} \) are the equivalent setting pressures of the balance valve springs at the oil ports of the non-rod chamber and rod chamber. From simultaneous equation (3) and (24), the equation got is

\[ P_L = \frac{P_{\text{vl}} - (1 + nk_1)P_2}{k_2}, x_1 \geq 0 \quad \text{and} \quad P_L = \frac{(n + k_1)P_1 - nP_{\text{vl}}}{k_1}, x_1 < 0 \] (25)

Because of \( 0 < P_L < P_{\text{max}} \) and \( 0 < P_L < P_{\text{max}} \), from equation (3), the equation got is

\[ -nP_{\text{max}} < P_L < P_{\text{max}} \] (26)

The load pressure range is also the load boundary of a fully matched asymmetric valve control asymmetric cylinder system. From simultaneous equation (25) and (26), the equation got is

\[
\begin{align*}
&\frac{nP_{\text{vl}} - nP_{\text{max}}}{1 + nk_2} < P_1 < \frac{nP_{\text{vl}} + P_{\text{max}}}{1 + nk_2}, \quad P_{\text{vl}} < \frac{nP_{\text{vl}} + P_{\text{max}}}{1 + nk_2}, x_1 \geq 0 \\
&\frac{P_1 - P_{\text{max}}}{n + k_1} < \frac{nP_{\text{vl}} - nkP_{\text{max}}}{n + k_1}, \quad P_1 < \frac{nP_{\text{vl}} + kP_{\text{max}}}{n + k_1}, x_1 < 0
\end{align*}
\] (27)

Because of \( 0 < P_1 < P_{\text{max}} \) and \( 0 < P_2 < P_{\text{max}} \), the equation got is

\[
\begin{align*}
&\frac{nP_{\text{vl}} - nP_{\text{max}}}{1 + nk_2} \geq 0, \quad \frac{nP_{\text{vl}} + P_{\text{max}}}{1 + nk_2} \leq P_{\text{max}}, \quad x_1 \geq 0 \quad \text{and} \quad \frac{P_1 - P_{\text{max}}}{n + k_1} \geq 0, \quad \frac{nP_{\text{vl}} + kP_{\text{max}}}{n + k_1} \leq P_{\text{max}}, x_1 < 0
\end{align*}
\] (28)

Finally, the equation is

\[ P_{\text{max}} \leq P_{\text{vl}} \leq k_2P_{\text{max}} \leq k_2P_{\text{max}} \leq P_{\text{vl}} \leq P_{\text{vl}} \geq k_2P_{\text{max}} \leq P_{\text{vl}} \leq P_{\text{vl}} < 0 \quad \text{and} \quad P_{\text{max}} \leq P_{\text{vl}} \leq k_2P_{\text{max}} \leq k_2P_{\text{max}} \leq P_{\text{vl}} \leq P_{\text{vl}} \leq k_2P_{\text{max}}, k_2P_{\text{max}} \leq P_{\text{vl}} < 0 \] (29)

To get the equation (29), it is easy to know that the conversion coefficient of the two balance valves should meet \( k_1 = k_2 = 1 \), and \( P_{\text{vl}} = P_{\text{vl}} = P_{\text{max}} \). And the load boundary of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve can be expanded to \( (-nP_{\text{max}}, P_{\text{max}}) \). The whole load boundary in system will change from (-0.51MPa, 1.56MPa) to (-8MPa, 10MPa). The essence of increasing the balance valve to expand the load boundary is to increase the oil return back pressure of the system, so that the system has the ability to withstand the over load.

5. Conclusion
This paper analyzes and studies the load boundary of the load-sensing hydraulic system with multi-way valve and pressure-compensated valve and the selection principle of the balance valve. The conclusions are as follows:
Without considering the balance valves, the load range corresponding to the load-sensing and constant pressure oil supply conditions of the system is derived when the system is moving forward and backward. Meanwhile, the forward and backward load boundary and the whole load boundary are also derived;

- In the case of no balance valve, the ability of system to withstand the excess load is extremely small, resulting in a narrow range of the whole load boundary of the system, so a balance valve must be added to improve the carrying capacity of system;

- The selection principle of balance valve is derived. According to this principle, the selection of balance valve can expand the system load boundary to \( (-nP_{\text{max}}, P_{\text{max}}) \);

- In the derivation process, any valve-controlled cylinder structure is considered, and the conclusions obtained are suitable for the load-sensing hydraulic system with multi-way valve and pressure-compensated valve of any valve-controlled cylinder structure.

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