Research on Control Strategy of Vehicle Shock Absorber Testing Bench

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Abstract. The testing platform of shock absorber powered by hydraulic system is researched and developed, which can meet the need of performance testing for the high-speed and high-precision shock absorber. The characteristics of valve coefficient asymmetry caused by different loads when the electro-hydraulic servo system moves in two directions are analyzed by establishing mathematical model of the hydraulic testing platform power mechanism. The nonlinear compensation strategy of servo valve flow is introduced to make the linear relationship between the load flow and spool displacement, and then realizing the linear control for this system. To avoid hydraulic shock in the process of hydraulic dynamical outfit on reversing, the impact suppression parameters are designed in the nonlinear flow compensation control strategy. A rapidly control prototype based on xPC of the testing platform is devised, and the performance test was conducted on the developed shock absorber. Test results prove that the developed shock absorber testing platform electro-hydraulic servo control system meets the needs of performance testing, which has vital engineering research significance.

Introduction

Shock absorber is an important damping elements of the vehicle suspension systems, the main function of which can effectively absorb vibration from road and then transform the vehicle kinetic energy into heat emitted to air[1]. There are many kinds of shock absorbers, which are in form of electromagnetic variable fluid, electromagnetic and hydraulic mechanical. Nowadays, hydraulic shock absorber have widely used in the market[2]. Generally, it has the form of internal and external tube structure. The hydraulic fluid drawn by piston alternating motion between the inner and outer cylinder so that the liquid from different directions with the influence of different motion control valves to obtain different damping force[3]. Owing to lacking of the regulating and controlling element with the movement of liquid in two directions, damping force obtained by the traditional shock absorber is fixed[4]. With the development of vehicle, people’s car riding comfort demanding high performance, so damping adjustable shock absorber arise at the historical moment[5]. At present, it’s been widely used in luxury cars and sports car. Domestic automobile, shock absorber manufacturers and scientific research institutes have also been poured a lot of effort into the research work in the field of damping adjustable shock absorber[6]. The research work of shock absorber can never get rid of testing segment, and evaluating a shock absorber performance should be also verified by experiments. It is an important link in the developing process of shock absorber, and then the developed shock absorber will be tested and demonstrated by setting up an experimental platform in the laboratory[7]. Hammering was a earlier testing method for the performance of the shock absorber with low cost and quickly judgment ability. However, the complicated test can’t be realized and the relatively simple of evaluation index, which can’t satisfy the requirement of estimating for its comprehensive performance index. Subsequently, the motor-driven shock absorber testing platform was emerging, and had achieved the experimental speed adjustment[8]. But it held the more complex transmission mechanism and relatively smaller output. Because of the larger output and the higher bandwidth characteristics, the hydraulic vibration exciter is developed to meet the needs of various testing for shock absorbers[9]. This mechanism is comparatively simple, and even provides possibility for the whole of vehicle test, so it becomes more increasingly and widely application. In
this paper, a testing platform powered by hydraulic had developed depending on the needs of experiment for the developed adaptive damping shock absorber.

Mathematical Model

The dynamic mechanism of vehicle shock absorber testing platform has been designed in the form of valve control cylinder mode, the testing system is composed of displacement sensor, force sensor, real time controller and other components. A mathematical model is developed for the power mechanism in order to design the controller. In this paper, the valve control cylinder schematic diagram shown in Fig.1. The servo valve and the hydraulic cylinder are symmetrical, which makes the dynamics of system uniformity in two directions, and which is beneficial to study the control algorithms for this system.

![Principle diagram of power mechanism.](image)

The mathematical model of the dynamic mechanism mainly includes three aspects: the flow equation of slide valve, the continuity equations of the hydraulics and the force equilibrium equations of the hydraulic cylinder piston. The first is the flow equation of slide valve. It needs to give the following assumptions before the flow equation is established[10]:

- The four throttle orifice of servo valve is matching and symmetry, which is the ideal-zero-opened and four-through slide-valve. Neglecting the influence of the fluid compressibility in the valve; The orifice flow is treated as turbulent[11].

According to the throttle formula, the flow into the left chamber of the hydraulic cylinder can be written as

\[ q_1 = \begin{cases} 
  C_d w x \frac{2}{\rho} (p_s - p_1) & x \geq 0 \\
  C_d w x \frac{2}{\rho} (p_s - p_1) & x < 0 
\end{cases} \]

where, \( q_1 \) is the flow rate of cylinder’s left chamber, m\(^3\)/s; \( C_d \) is the discharge coefficient of the orifice, non-dimensional; \( w \) is the area gradient; \( x \) is spool displacement of servo valve, m; \( \rho \) is oil density, kg/m\(^3\); \( p_s \) is the oil source pressure, Pa; \( p_1 \) is the pressure of left chamber, Pa.

The oil flows into the right chamber of hydraulic cylinder is formed as

\[ q_2 = \begin{cases} 
  C_d w x \frac{2}{\rho} (p_2 - p_1) & x \geq 0 \\
  C_d w x \frac{2}{\rho} (p_2 - p_1) & x < 0 
\end{cases} \]

where, \( q_2 \) is the flow rate of cylinder’s right chamber, m\(^3\)/s; \( p_2 \) is the right chamber pressure, Pa.

Secondly, according to the continuity equations of the hydraulics, the key assumptions are as followed:

1. Ignoring friction losses, influence of fluid quality and dynamics in the pipelines.
2. Pressure is same in everywhere in the chambers of hydraulic cylinder; oil temperature and elastic modulus are treated as constants.
(3) Leakage of the hydraulic cylinder is regarded as laminar flow.

Therefore, flow continuity equations used in the left chamber of the hydraulic cylinder can be written as

\[ q_1 = A \dot{v}_1 + \frac{v_1}{\beta} \dot{p}_1 + c_{ic} (p_1 - p_2) + c_{ec} \dot{p}_1. \]  

(3)

where, \( A \) is the effective area of the piston, \( m \); \( \dot{v}_1 \) is the piston displacement, \( \text{Pa} \); \( v_1 \) is the volume of the left chamber, \( \text{m}^3 \); \( c_{ic} \) is internal leakage coefficient, \((\text{m}^3/\text{s})/\text{Pa})\); \( c_{ec} \) is internal leakage coefficient, \((\text{m}^3/\text{s})/\text{Pa})\). Respectively, the flow continuity equations of the right chamber for the hydraulic cylinder can be determined as

\[ q_2 = A \dot{v}_2 - \frac{v_2}{\beta} \dot{p}_2 + c_{ic} (p_1 - p_2) - c_{ec} \dot{p}_2. \]  

(4)

where, \( \dot{v}_2 \) is the volume of the right chamber, \( \text{m}^3 \). In above formulas, the volume in left and right chambers respectively are:

\[
\begin{align*}
\dot{v}_1 &= v_0 + A \dot{l} \\
\dot{v}_2 &= v_0 - A \dot{l}
\end{align*}
\]

where, \( v_0 \) is the original volume of each chamber when the hydraulic cylinder is located at the middle position, \( \text{m}^3 \).

Finally, the force equilibrium equations of the hydraulic cylinder piston are established by

\[ f_l = m \ddot{l}. \]  

(6)

where, \( m \) is mass of inertia load and piston, \( \text{kg} \); \( f_l \) is driving force generated by the hydraulic cylinder, \( \text{N} \).

Meanwhile, the driving force generated by the hydraulic cylinder can be expressed as

\[ f_l = A (p_1 - p_2). \]  

(7)

Defining the load flow \( q_L = 0.5(q_1 + q_2) \), and the load pressure \( p_l = p_1 - p_2 \); considering \( p_s = p_1 + p_2 \) and \( p_r = 0 \); then adding equations (1) and (2) can obtain

\[ q_L = C_{wc} \sqrt{\frac{p_2 - \text{sign}(x_s) p_L}{\rho}}. \]  

(8)

Assuming that the hydraulic cylinders are minimum amplitude motion in the middle position, then adding the Eqs. of (3) and (4).

\[ q_L = A \dot{l} + \frac{V}{4\beta} \dot{p}_L + c_{ec} p_L. \]  

(9)

where, \( C_{tc} = C_{ic} + 0.5C_{ec} \) is total leakage coefficient of the hydraulic cylinder; \( V_t \) is total volume of two chambers, \( \text{m}^3 \).

According to equations (6)–(9), the nonlinear dynamic model of the electro-hydraulic servo system is expressed as the block diagram shown in Fig. 2. The nonlinear parts mainly taking the nonlinear effect of spool flow equation into consideration.

Figure 2. Block diagram of the nonlinear motor.
In the electro-hydraulic servo system, the control signals are voltage signals which is calculated by computer. However, the driving signal of the electro-hydraulic servo valve is current signal; Signal transformation between the voltage and current can be realized by the amplifier. Because the bandwidth of amplifier is generally very high, so the relationship between the voltage and the current can be equivalent to proportional function. Generally, the dynamic relationship between the spool displacement of servo valve and the input current can be described by using oscillation system[12]. When the inherent frequency of power mechanism is less than 50Hz, the dynamic relationships between the spool displacement and the current can be expressed in inertia element.[13] When the servo valve bandwidth is far more than the nature frequency of power mechanism, the proportional element can be used to describe the dynamic characteristics of the servo valve. When the servo valve bandwidth selected is far more than the nature frequency of power mechanism in the shock absorber test platform, the relationship between the servo valve spool and the computer command voltage can be described as

\[
\frac{X_v}{u} = k_{sv}.
\]

where, \(u\) is the instruction voltage, \(V\); \(k_{sv}\) is amplification factor of instruction voltage to spool displacement, \(m/V\).

**Controller Design**

Generally, in engineering applications, electro-hydraulic servo system is considered as a linear system. However, there are many nonlinear factors such as servo valve dead zone, hysteresis phenomenon and coulomb friction of hydraulic cylinder in the hydraulic servo system. This paper only takes the nonlinear influence of flow equation for slide valve into consideration. When load changes, the system operating zero position is also changing. The nonlinear flow equation will lead to different valve flow coefficients both in forward and reverse movement, in further, an asymmetrical movement generates in result.

In order to get a linear proportional relationship between the load flow and spool displacement, a nonlinear compensation function \(h(p_L)\) is introduced in the nonlinear model. After this compensation, a linear equation of the flow can be acquired and illustrated in Fig. 3.

Thus, the spool displacement of servo valve becomes:

\[
x_v' = h(p_L) x_v.
\]

The flow nonlinear compensation function \(h(p_L)\) is defined as

\[
h(p_L) = \frac{p_L}{\sqrt{p_r - \text{sign}(x_v) p_L}}.
\]

This nonlinear compensation function is related to the load pressure which can be measured by a pressure sensor. After this compensation, a linear flow of serve valve is gained and expressed as

\[
q_c = C_w x_v \sqrt{\frac{p_r - \text{sign}(x_v) p_L}{\rho}} = C_w x_v \sqrt{\frac{p}{\rho}}.
\]
When cylinder movement direction changes, the hydraulic impact occurs, and the transient load pressure will turn into relatively high. In this case the numerical denominator root in formulas (12) will become very small, leading to a very large increment of spool displacement. In order to prevent this kind of situation, it needs to set a limiting factor $k_{lim}$ for the load pressure. Because the load pressure is the oil source pressure of $2/3$ at the maximum power point. Therefore, the limiting factor $k_{lim}$ can be written as

$$k_{lim} = \begin{cases} 
1 & p_L \leq \frac{2}{3} p_s \\
\frac{2p_r}{3p_L} & p_L > \frac{2}{3} p_s
\end{cases} \quad (14)$$

Replacing Eqs. (14) into (12), the flow nonlinear compensation function will become

$$h^*(p_L) = \frac{p_r}{p_s - k_{lim} \cdot \text{sign}(x_r) p_L}. \quad (15)$$

**Experiment**

The experimental bench is shown in Fig. 4-a). Based on the experiment platform, two experimental aspects will be conducted: one is to validate the improvement of this system performance by using nonlinear flow compensation control strategy; another one is to test and verify the performance of the designed adaptive damping shock absorber.

Respectively, setting system’s frequency at 6.67 Hz, 8 Hz and 10 Hz, the comparison results have acquired and described as figure 4-b) to 4-d). In these figures, the solid line is the excitation signal for the system; and the dotted line is the system response signal. Reproduction accuracy of sine wave is controlled within 1% in this case.

![Figure 4. Hydraulic testing platform.](image)

The characteristics of shock absorber are performed based on this hydraulic experimental platform. This paper mainly aims to the test of its indicator diagram. The shock absorber is mounted on the platform, the electro-hydraulic servo actuator is connected to the piston rod through the transition part, whose original position is about in the middle of the shock absorber; the piston rod movement is driven by an actuator, which arises damping force after the hydraulic oil pass through a variety of valve. The damping force can gain directly via a force sensor, which fixed on the beam on the top of the shock absorber, the displacement of the piston rod is sine curve, which based on the frequency and the amplitude relationships, and the derivative of displacement curve is the piston speed, according to the maximum peak speed of 0.1 m / s and 0.52 m / s carry out the test on the shock absorber.

Recording the piston rod displacement, the corresponding damping force and drawing in the same coordinate system, we can get indicator diagram of the shock absorber shown in Fig. 5.
Conclusion

The developed experimental platform of the hydraulic shock absorber achieves 10Hz sinusoidal signal with higher bandwidth, higher accuracy and the error can be limited to 1%. The nonlinear compensation strategy proposed in this paper improves the system bandwidth and control precision. The developed testing platform meets the requirement of laboratory experiments in the process of absorber development. According to the hydraulic test platform, the indicator diagram of the shock absorber can be obtained accurately. It can be seen from the indicator diagram, the developed experimental platform of the hydraulic shock absorber has four damping force. The faster the speed is, the larger the damping force will be; The fuller the graphic is, the better the performance of shock absorber will be.

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References


