

Linear Control Performance Improvement of High Speed On-Off Valve Controlled by PWM

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ABSTRACT

High speed on-off valve is applied widely in vehicle control systems. When high speed on-off valve is controlled by Pulse Width Modulation (PWM) of high frequency, the valve core can float at a certain position which is adjusted by changing the duty ratio within a certain effective range. Then the high speed on-off valve can control the flow and pressure linearly like proportional valve. Thus it is essential to extend the effective range of duty ratio to improve the linear control performance of high speed on-off valve.

In this paper, the high speed on-off valve of the automotive Electronic Stability Program (ESP) is the focus, and its flow force is analyzed in detail to get the effects of hydraulic parameters on the valve performance. The mathematic model of the high speed on-off valve is derived. Then the valve structural parameters are optimized according to the Genetic Algorithm(GA), offering the theoretical references for extending the effective duty ratio of PWM. Besides, the simulation model of high speed on-off valve is established in MATLAB/Simulink, and then embedded in the Hydraulic Control Unit (HCU) simulation model established in AMESim, resulting in the co-simulation model of hydraulic actuator. Finally the simulation model is verified by the hardware-in-loop tests. Based on theoretical analysis, simulations and tests, it is critical to increase the range of flow force for extending the effective range of duty ratio. The pressure difference, the valve seat angle and the throttle diameter of the high speed on-off valve have great effects on the flow force. At last the effects of structural parameters of the high speed on-off valve on extending the range of the linear control are proposed, serving as references for improving the linear control performance of high speed on-off valve.

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INTRODUCTION

Electronic stability program (ESP) is an advanced active safety system, which consists of an Electronic Control Unit (ECU), Hydraulic Control Unit(HCU) and sensors. When the automobile is in an emergency, the ECU can work out indispensable yaw torque to make the automobile stay on the right track. Then the HCU adjusts the brake cylinder pressure to get the yaw torque. That is to say, HCU is the actuator of ESP. It is very important to control the brake cylinder pressure precisely by HCU to achieve the expected ESP control effect.

The sketch of HCU is shown in Figure1. Inlet valve, outlet valve, pre-charge valve, change-over valve are all high speed on-off valves[1]. It is obvious that the high speed on-off valves are vital components for HCU. In recent years, it has been confirmed that the valve core of high speed on-off valve controlled by high frequency Pulse Width Modulation (PWM) can float at a certain position. By changing the duty ratio of PWM, the position is adjusted. It can control the flow and pressure proportionally. But the duty ratio of PWM is available in a certain range. The valve is easy to open and close completely when duty ratio is beyond the range. Consequently it would be helpful to mathematically define the duty ratio the first time

it is used, and it is necessary to study the parameters of the high speed on-off valve to extend the effective duty ratio range and improve the linear control performance of the high speed on-off valve.

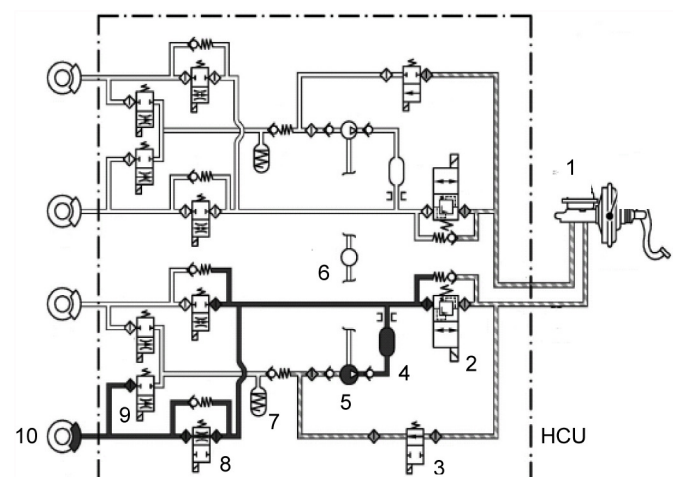


Figure 1. The sketch of ESP hydraulic control unit 1-master cylinder 2-change-over valve 3-pre-charge valve 4-damper 5-piston pump 6-motor 7-accumulator 8-inlet valve 9-outlet valve 10-brake cylinder

Many researchers studied the HCU of ESP by testing and modeling. Sornioti and Velardocchia (2004) conducted many experiments about ESP hydraulic component on the Hardware-In-the-Loop (HIL) brake systems test bench, such as step response and frequency response of ESP by using PWM[2]. Pressure gradients and pressure oscillations are obtained in the tests. Herbert Schuette and Peter Waeltermann(2005) evaluated the dynamic characteristics of ESP and Electro-Hydraulic Brake (EHB) hydraulic components based on HIL methodology[3]. Ding Nenggen(2004) got the experienced model of the speed of brake pressure building up via tests[4]. Zhang Biao(2007) tested the rise and fall of pressure to acquire the method of brake pressure precise adjustment[5]. Both of them indicate that the brake cylinder pressure affects pressure control performance of valve. Ijin Yang, Woogab Lee, et al(2003) conducted a model-based component design analysis by comparison of test data[6][7].

In hydraulic fields, high speed on-off valve and its applications are widely studied. Sung-Hwan Park (2009) designed the servo valve-type water hydraulic valve [8]. He used proportional poppet as the main valve to avoid flow and pressure fluctuation caused by high speed on-off valves under the control of PWM. Heon-Sul Jeong(2001) analyzed theoretically and experimentally the pressure control characteristics of a three-way high speed on-off valve driven by a PWM signal[9]. He concluded that the mean pressure and the ripple depend on the duty ratio and the carrier frequency of PWM. Zhu Kangwu(2012) developed a two-stage high speed on-off valve which is used in a hydraulic propeller system to solve the conflict between large flow rate and high frequency response performance[10]. John Mahrenholz (2010) described a modeling method for high speed on-off valves which focused on the nonlinearities of the electromagnetic, fluidic and mechanical domains [11].

In this paper, the flow force of high speed on-off valve is analyzed in detail. The flow force affects deeply the control performance of high speed on-off valve. The effect of structure parameter on the flow force and linear control performance of valve is shown. In brief, the paper would be organized as follows. In [section 2](#), the mathematical model of high speed on-off valve is presented, which includes the flow force analysis. In [section 3](#), the parameter is optimized by Genetic Algorithm(GA). In [section 4](#), the co-simulation model of HCU established by AMESim and MATLAB is presented. Then, in [section 5](#), the simulation results and HIL test results are shown and discussed. The crucial parameters on the improvement of linear control performance of high speed on-off valve are acquired. Finally, some conclusions are presented in [section 6](#).

HIGH SPEED ON-OFF VALVE MODEL

The change-over valve is a high speed on-off valve, its configuration and free body diagram are shown in [Figure 2](#). At the beginning it opens by the effect of spring force. The fluid flowing through the valve brings up the flow force. When the solenoid of valve is electrified, the electromagnetic force makes the valve open rate smaller. The motion of the valve is mainly determined by the electromagnetic force, the flow force and the spring force. The motion equation of the valve is as follows,

$$m_f \ddot{x}_f = F_X + F_p - F_s - F_B \quad (1)$$

Where, m_f is the mass of the movement part of valve, x_f is the displacement of the movement part of valve, F_X is the electromagnetic force, F_p is the flow force, F_s is the spring force, F_B is the damping force.

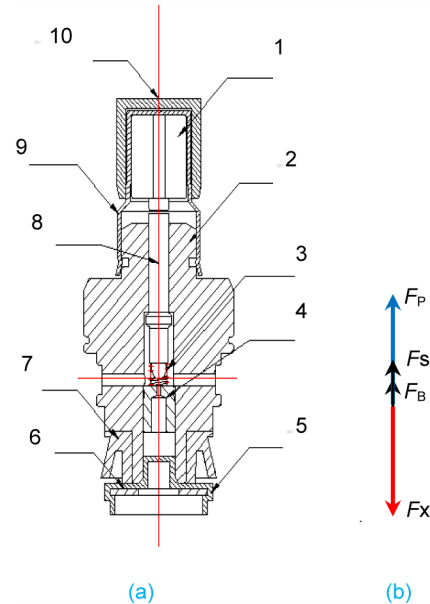


Figure2. The change-over valve1- moving iron 2- seat 3-spring 4-throttle 5-seal 6-washer 7-seal 8-tod 9-plunger tube 10-cap

Electromagnetic force

The circuit equation is as follows,

$$U = i \cdot R_w + N_w \frac{d\phi_m}{dt} \quad (2)$$

Where, U is voltage, i is current, R_w is coil resistance, N_w is coil turns, ϕ_m is magnetic flux.

The magnet circuit equation is given by

$$iN_w = \phi_\delta (R_\delta + R_n + R_m) \quad (3)$$

Where, ϕ_δ is the air gap magnetic flux, R_δ is work air gap magnetic resistance, R_n is non-working air gap magnetic resistance, R_m is magnetic circuit magnetic resistance.

Leakage coefficient is given by

$$\sigma = \frac{\phi_m}{\phi_\delta} \quad (4)$$

The electromagnetic force is given by

$$F_X = \frac{\phi_\delta^2}{2\mu_0 \cdot S} \quad (5)$$

Where, μ_0 is vacuum permeability, S is magnetic poles area in the air gap.

Based on the [equation \(2\)](#), [\(3\)](#), [\(4\)](#), [\(5\)](#), the electromagnetic force can be determined.

Flow Force

Flow force which consists of steady flow force and transient flow force plays an important role in the movement of valve. The fluid between the inlet and the orifice of the valve is concerned, which is named as control volume, as shown in [Figure3](#). Based on the fluid momentum theorem, the steady flow force is derived as follows,

$$F_{stx} = \rho Q v_0 - \rho Q v_j \cos \alpha + P_0 A_1 + \int P_j 2\pi r dr - P_2 A_j \cos \alpha \quad (6)$$

Where, v_0 is flow speed at the inlet, v_j is flow speed at the orifice, P_0 is pressure at the inlet, P_j is pressure at the orifice, P_2 is pressure at the outlet, A_0 is surface area of inlet, A_1 is surface area of throttle, A_j is surface area of orifice.

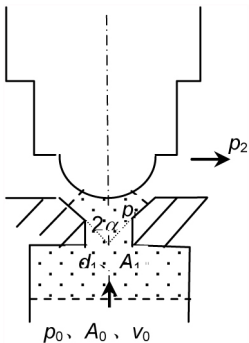


Figure3. The layout of control volume of change-over valve

According to the Bernoulli equation, the fluid pressure at the orifice is acquired as follows,

$$P_j = P_2 + c_d^2 \Delta P \left(\frac{A_j^2}{A_2^2} + \zeta \frac{A_j^2}{A^2} - 1 \right)$$

Where, c_d is discharge coefficient, ζ is coefficient of local resistance.

Because of $\frac{A_j^2}{A_2^2} + \zeta \frac{A_j^2}{A^2} \ll 1$, the previous equation is simplified as follows,

$$P_j = P_2 - c_d^2 \Delta P \quad (7)$$

In [equation \(6\)](#), the flow rate can be substituted by the following formula,

$$Q = c_d A_j \sqrt{\frac{2}{\rho} \Delta P} \quad (8)$$

Thus

$$\rho Q v_0 = \frac{2 c_d^2 A_j^2}{A_0} \Delta P \quad (9)$$

$$\rho Q v_j \cos \alpha = 2 c_d A_j \Delta P \cos \alpha \quad (10)$$

Usually the pressure of the output is supposed to be zero in theoretical analysis, the flow force on the top of the valve core is therefore ignored. But it is not applicable in reality because there is always load pressure on the outlet of valve. In this paper, there is the pressure P_3 on the top of the valve core. When the valve reaches steady state, P_3 is equal to P_2 , then the steady flow force is as follows

$$F_{st} = F_{stx} - P_2 A_1 \quad (11)$$

Based on the [equation \(6\)](#), [\(7\)](#), [\(8\)](#), [\(9\)](#), [\(10\)](#), [\(11\)](#), the steady flow force is given by

$$F_{st} = 2 \frac{c_d^2}{A_0} \Delta P A_j^2 - (P_2 \cos \alpha + 2 c_d \Delta P \cos \alpha) A_j + (A_1 - c_d^2 A_{d12}) \Delta P \quad (12)$$

Where, A_{d12} is area of the poppet seat in horizontal direction.

The area of the orifice is stated

$$A_j = \frac{c_d \pi d_f x_v}{R} \sqrt{R^2 - \left(\frac{d_f}{2}\right)^2} \quad (13)$$

Where, d_f is the mean diameter of throttle, x_v is the opening of valve core, R is the radius of ball valve.

According to the [equation \(12\)](#), F_{st} is a second order equation about the area of valve orifice. The constant is about the steady flow force while valve is close completely. The first order coefficient affects the gradient of steady flow force while the valve open rate changes. Because the maximum displacement of valve core is just 0.22mm and the stiffness of spring is 700N/m, the spring force is very small. Compared with the flow force and electromagnetic force, the spring force can be negligible. If the range of flow force would be wider, the bigger range of electromagnetic force could be balanced. As a result, the effective duty ratio range is extended and the high speed can be