Wheel cylinder pressure estimation of EHB based brake-by-wire system

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Abstract
This paper introduces a method of on-line estimation of the wheel cylinder pressure, which can improve the regulation accuracy of the wheel cylinder pressure and reduce the actuation time of the solenoid valves, thus, bring the advantages of improving the quality of pressure control, the longevity of the solenoid valves and the comfortableness of vehicle. Firstly, a mathematical model is established to estimate the wheel cylinder pressure using lumped-parameter bond graph technique. The discrete system difference equation can be derived from the proposed model to estimate the real wheel cylinder pressure using the data from pressure sensors in real time. Secondly, a proper digital filter is applied to post process the estimated wheel cylinder pressure to make it suitable for pressure controller, for certain fluctuation exists at the brake transmission line with high frequency, which is undesirable for the controller. The pressure lag introduced by the filter is compensated by data processing. Finally, a device is introduced to improve the transient characteristics of the braking system, which would improve the quality of pressure control. Test has been made on the test bench, and the accuracy of the on-line estimation of wheel cylinder pressure is proved to be satisfactory by test result.

Keywords: Electro Hydraulic Brake, Wheel Cylinder Pressure, Bond Graph, Fluid Transients, Pressure Filter

1. Introduction

The (Electro-Hydraulic Brake) EHB system has many advantages. Wheel cylinder pressure can be closed-loop controlled independently without vacuum booster, which provides a better braking performance and is extremely suitable for electric vehicles. Anti-lock Brake System (ABS)/Electronic Stability Program (ESP)/Electrical Park Brake (EPB) functions can be integrated in the EHB system without any additional components.

As one of the key technology of the EHB system, the pressure regulation method is always a bottleneck with the increasing demand for the fast and accurate response. Various EHB systems have been developed by automotive manufactures and researchers. Regardless of the difference in actuators and control strategy, the basic method of pressure regulation is feedback control. Pressure sensors, whose output is considered as the wheel cylinder pressure in pressure feedback control, are mounted inside the Hydraulic Control Unit (HCU) in a variety of EHB systems, which minimizes size and cost of the system (Jonner, et al., 1996, Nakamura, et al., 2002). Nevertheless, the pressure obtained by pressure sensor is not in accord with the real wheel cylinder pressure, because the HCU connects wheel cylinder by brake tubes and brake hoses which inevitably lead to the pressure loss and pressure lag. Therefore, it is necessary to investigate a method to estimate wheel cylinder pressure based on the output of pressure sensors.

Researches have been carried out to investigate pressure regulation method with better performance in accuracy and response time. Most of them focused on the pressure control strategy, which calculates driven signal for solenoid valves based on the target pressure and the real pressure acquired by the pressure sensor. PID control (Soga, et al., 2002, Petruccelli, et al., 2003), feedforward control (Park, et al., 2009) and sliding mode (Aoki, et al., 2007) control are applied in the wheel cylinder pressure regulation controller. Nevertheless, few attentions have been paid on the calibration of the output of pressure sensor and the estimation of wheel cylinder pressure. There are some investigations...
made into the estimation of wheel cylinder pressure for Electronic Stability Program (ESP) system without pressure sensors installed in individual wheel cylinder (Zanten, et al., 1996, Liang, et al., 2008), however the estimation method cannot apply to the EHB system since the accuracy is hardly acceptable.

The key to wheel cylinder pressure estimation is the investigation of fluid transient characters in brake transmission lines. Frequency-domain representation is adopted in various researches and which handles arbitrary boundary conditions directly, but for on-line estimations, time-domain solutions are necessary (Yang, et al., 2012). Several relatively accurate mathematical models that describe fluid transient characters in transmission lines have been proposed in the time-domain. Characteristic curve method can obtain the fluid dynamics in pipings based on the analysis of the transient characteristics and frequency characteristics. It is simple and accurate, but the computation of boundary conditions is complex (Ogino, et al., 1991). "Dissipative Model" appears to be the most comprehensive for laminar flow conditions. It is quite complex because the transfer function consists of both hyperbolic function and Bessel function (Brown, 1984). Wongputorn proposes approximations for Dissipative Model to obtain rational polynomial transfer function using a linear least-square criterion, but the transfer function order is too high for on-line estimation (Wongputorn, 2003a, 2003b, 2005). Another method is to establish modal models by bond graph, utilizing rational transfer functions in the Laplace domain solution or separation of variables techniques, which are highly enough but too complex to be implemented (Yang, et al., 2012). In summary, the above mentioned methods are all distributed-parameter models with good accuracy but too complex for on-line estimation of brake wheel cylinder pressure.

For on-line estimation, besides of accuracy, it is important to be uncomplicated enough and have acceptable real-time performance considering the implementation in the Electronic Control Unit (ECU). Therefore, in this paper, lumped-parameter models (Goodson and Leonard, 1972) are adopted instead of distributed-parameter models. Bond graph is used in modeling, for it can provide a direct link between a system under researching and its mathematical equations (Gad, 2013).

This paper introduces a method of on-line estimation of brake wheel cylinder pressure, which can improve the regulation accuracy of wheel cylinder pressure. Because the estimation is more close to the real cylinder pressure than the output of the pressure sensors, a more accurate feedback pressure can be provided for the controller. Moreover, the method can reduce actuation time of solenoid valves, which improves the life time of the solenoid valves and the comfortableness of vehicle. The mathematical models, post processing of the estimated pressure and a device to improve the characteristics of the brake system are discussed in this paper.

2. EHB system configuration

Figure 1 shows the typical scheme of an EHB system. Besides the components of a conventional braking system, EHB system is comprised of a brake pedal stroke sensor, a pedal simulator, six pressure sensors (one for the master cylinder, one for accumulator, and the others for wheel cylinders) and a Hydraulic Control Unit (HCU). The HCU is comprised of twelve solenoid valves, a pump motor and a high pressure accumulator. All pressure sensors are also mounted in the HCU.

Upon detection of the driver's brake intention characterized by the brake pedal stroke sensor (s1 in Fig.1) and the master cylinder pressure sensor (s2 in Fig.1), the desired braking force is calculated. The actuation of inlet valves (v1 in Fig.1) and outlet valves (v2 in Fig.1) is regulated according to the target pressure derived from the desired braking force and the pressure feedback from the pressure sensor.

A wheel cylinder pressure sensor is added at the further end of the brake hose and near the caliper to acquire the real wheel cylinder pressure. The sensor is especially used for verifying the proposed estimation model, but not used by the feedback controller in order to be consistent with the real system. The ECU outputs driving commands to the solenoid valves and pump motor based on the target pressure and the pressure feedback.
3. Wheel cylinder pressure estimation
3.1 System modeling

Figure 2 shows the brake transmission line system structure that is used in the transient characteristic analysis of the EHB system. The original pressure sensors (s5 in Fig. 2) are embedded in the HCU near the outlet port, and the outlet port of HCU connects wheel cylinder by brake tubes and brake hoses which inevitably lead to the pressure loss and pressure lag. An additional pressure sensor is mounted near the wheel cylinder as s6 in Fig. 2 to gain the real wheel cylinder pressure in the following experiments, making it possible to research and verify the method of estimating the real wheel cylinder pressure utilizing the output of the original pressure sensor such as s5 in Fig. 2. The input of the brake line model is the export of the original pressure sensors, and the output of the system modeling is the real wheel cylinder pressure which can better express the real pressure controlled by the pressure controller.

Bond graph is adopted in the system modeling. It is an excellent tool for capturing a wide variety of systems ranging from the mechanical, electrical and hydraulic systems, because it can provide a direct link between the system under investigation and its mathematical equations. The fundamental idea of a bond graph is that power is transmitted between connected components by a combination of effort and flow.

The bond graph of the brake line system including brake tube, brake hose and brake wheel cylinder is shown in Fig. 3. The brake tube is modeled in the bond graph by a resistance (R-type element, R1). The brake hose can be
simplified as a capacitor (C-type element, \( C_1 \)), a resistance (R-type element, \( R_2 \)) and an inductance (L-type element, \( L_1 \)), due to the flexibility of the hose wall and the relatively large volume of the brake hose. The brake wheel cylinder is simplified as a big capacitor (C-type element, \( C_2 \)) for its big volume and a spring. The capacitor, resistance and inductance are all simplified as linear components in this model.

\[ \begin{align*}
R_1 & \quad \text{Q}_1 \quad \text{P}_1 \\
R_2 & \quad \text{Q}_2 \quad \text{P}_2
\end{align*} \]

Fig. 3 The bond graph of the brake transmission line. Pressure rate is the input of the bond graph. The two variables are pressure and flow, whose product is power. The pressure and flow variables are all shown in this figure. The brake tube is simplified as a resistance, where the brake hose is modeled as a resistance, an inductance and a capacitor for the flexible hose wall. The wheel cylinder is simplified as a capacitor.

Constitutive equations can be derived from the bond graph model as Eq. (1)-(4), where \( P_1 \) is the pressure of the output of pressure sensor which mounted in the HCU, and \( P_3 \) is the pressure of the output of additional pressure sensor near the wheel cylinder, which can represent the real cylinder pressure.

\[
\begin{align*}
\text{P}_1 - \text{P}_2 &= R_1 \dot{\text{Q}}_1 \\
\text{Q}_1 - \text{Q}_2 &= C_1 \dot{\text{P}}_2 \\
\text{P}_3 - \text{P}_2 &= R_2 \dot{\text{Q}}_2 + L_1 \dot{\text{P}}_2 \\
\text{Q}_2 &= C_2 \dot{\text{P}}_3
\end{align*}
\]  

(1) \hspace{1cm} (2) \hspace{1cm} (3) \hspace{1cm} (4)

The continuous system transfer function \( G(s) \) can be derived by using Laplace transformation to Eq. (1)-(4). The transfer function \( G(s) \) can be expressed by Eq. (5), which represents the frequency relationship between the output of pressure sensor and the real cylinder pressure.

\[
G(s) = \frac{P_3(s)}{P_1(s)} = \frac{1}{R_1 C_1 I_1 \left( s^2 + \frac{R_1 C_1 I_1}{R_2 C_2 I_1} s + 1 \right)}
\]  

(5)

The parameters in the transfer function are all assumed to be linear component. The fluid is parabolic flow, assuming the length of the pipe is longer than flow length, which is reasonable in this situation. Fluid resistance can be expressed by Eq. (6)-(7) for the viscous drag is significant. \( L_1 \) and \( D_1 \) are the length and the internal diameter of the brake tube, and \( L_2 \) and \( D_2 \) are the length and the internal diameter of the brake hose.

\[
\begin{align*}
R_1 &= \frac{128 \mu L_1}{\pi D_1^4} \\
R_2 &= \frac{128 \mu L_2}{\pi D_2^4}
\end{align*}
\]  

(6) \hspace{1cm} (7)

Fluid inductance of the brake hose is calculated by Eq. (8) in consideration of the fluid mass effect. \( A_2 \) is the
internal area of the brake hose, which can be derived from the internal diameter of the brake hose.

\[ I_1 = \frac{l_1 \rho}{A_2} = \frac{4l_1 \rho}{\pi d_i^2} \]  

(8)

Considering the compression characteristic of the brake fluid and the flexibility of the brake hose, fluid capacitor for the brake hose is expressed by Eq. (9). The equivalent capacitor of the brake hose is denoted by \( \beta_{eq} \) and it can be derived from the bulk modulus of fluid \( \beta_1 \) and the bulk modulus of the flexible wall of the brake hose \( \beta_2 \). For the wheel cylinder, the fluid capacitor is expressed by Eq. (10) for the compression characteristic of the brake fluid and the stiffness of the spring in the wheel cylinder, where \( V_c \) denotes the volume of the wheel cylinder, \( A_c \) denotes the area of the piston, and \( K \) denotes the stiffness of the piston return spring.

\[ C_1 = \frac{(A_2 l_2)}{\beta_{eq}} = \frac{(A_2 l_2)}{\frac{\beta_1 \beta_2}{\beta_1 + \beta_2}} \]  

(9)

\[ C_2 = \frac{V_c}{\beta_1} + \frac{A_c^2}{K} \]  

(10)

The transfer function between the output of pressure sensor and the real cylinder pressure can be derived from Eq. (5)-(10), as shown in Eq. (11), where the parameters \( p_1, p_2, p_3 \) can be derived from Eq. (5)-(10).

\[ G(s) = \frac{P(s)}{P_i(s)} = \frac{1}{p_3 s^3 + p_2 s^2 + p_1 s + 1} \]  

(11)

3.2 Pressure estimation

Wheel cylinder pressure cannot be directly estimated in real-time by transfer function as Eq. (11). Equation (11) is continuous and described in the frequency domain, however an equation with discrete form and described in the time-domain is essential for the on-line estimation of wheel cylinder pressure. Bilinear transformation is applied to discretize the transfer function, making the S-plane mapped to Z-plane as shown in Eq. (12). In this paper, the sampling interval of the wheel cylinder pressure is 0.001s. Substitute Eq. (12) into Eq. (11), the discrete transfer function of the brake line system is described in Eq. (13), where the parameters \( a_0, a_1, a_2, a_3, b_0, b_1, b_2 \) can be derived from Eq. (11)-(12).

\[ s = 2 \cdot F_s \cdot \frac{z - 1}{z + 1} \]  

(12)

\[ G(z) = \frac{a_0 + a_1 z^{-1} + a_2 z^{-2} + a_3 z^{-3}}{1 + b_0 z^{-1} + b_1 z^{-2} + b_2 z^{-3}} \]  

(13)

Further, rewrite Eq. (13) into the following form as Eq. (14). Finally, the discrete system difference equation (Eq. (15)) can be derived from Eq. (14). The discrete system difference equation can realize estimation of the real wheel cylinder pressure using the data acquired from pressure sensors.

\[ y(z)(1 + b_0 z^{-1} + b_2 z^{-2} + b_2 z^{-3}) = x(z)(a_0 + a_1 z^{-1} + a_2 z^{-2} + a_3 z^{-3}) \]  

(14)

\[ y(n) = a_0 x(n) + a_1 x(n-1) + a_2 x(n-2) + a_3 x(n-3) - b_0 y(n-1) - b_1 y(n-2) - b_2 y(n-3) \]  

(15)

The method of estimation of the real wheel cylinder pressure based on EHB system were examined and verified through bench test. The test bench is shown in Fig.4. The test bench is a complete brake system including EHB as the key component of this brake system. Other components are brake pedal, brake tubes, brake hoses and disc brakes. An additional pressure sensor is mounted near between the wheel cylinder and the brake hose. The structure of the test bench is the same as which presented in Fig.1 and Fig.2, including the hydraulic and sensor installation. Pressure control program and the algorithm of real cylinder pressure estimation are embedded in the Electronic Control Unit (ECU) of the EHB system.
Fig. 4 Test bench used for investigating and verifying the method for wheel cylinder pressure. The test bench is comprised of an EHB system, master cylinder, brake discs, brake calipers, brake pedal, brake tubes and hoses.

Figure 5 illustrates experiment of wheel cylinder pressure increase operation. Environment temperature during the test is around 20 degree centigrade, and the DOT 4 brake fluid is used in this brake system. The investigated method above is applied to estimate the real wheel cylinder pressure using the output of the pressure sensor in real time. The result indicates that the estimated pressure agrees with the real cylinder pressure well and the error is acceptable. Therefore, the model established above is accurate enough for the on-line estimation of wheel cylinder pressure.

Fig. 5 The test results of the method for estimation of wheel cylinder pressure. Pressure increasing operation is implemented as shown in the figure. The data of wheel cylinder sensor (blue) is obtained by the pressure sensor mounted in the HCU, while the real wheel cylinder pressure (black) is gained from the pressure sensor mounted near the wheel cylinder. On-line estimation of the wheel cylinder pressure (red) agrees with the real wheel cylinder pressure sensor well.
4. Wheel cylinder pressure post processing

Due to the motion of solenoid valves and pump to regulate wheel cylinder pressure and the characteristic of brake fluid, the pressure at the outlet of HCU has certain fluctuation. Therefore, the real wheel cylinder pressure would also have this fluctuation. By analyzing data of brake wheel cylinder pressure, it shows that the main fluctuation occurs in certain frequency and its harmonics. The result of estimated wheel cylinder pressure should be filtered before being used in feedback control to mitigate the fluctuation, for the fluctuation is of high frequency and is undesirable for the controller. In order to mitigate the fluctuation and its harmonics, a 16-point average filter is selected considering the real-time character of algorithm. The equation of the filter is shown in Eq. (16).

\[
y(n)_f = \frac{y(n-15) + y(n-14) + \ldots + y(n-1) + y(n)}{16}
\]

(16)

\[
y(n)_f = y(n)_f + (y(n-1)_f - y(n-2)_f) \times \frac{8}{10}
\]

(17)

The 16-point average filter will cause a delay of approximately 8ms, as the sampling interval of the wheel cylinder pressure is 0.001s. A method for the compensation of the delay is proposed. Firstly, the mode in which the system is activated is determined, e.g. pressure increase, decrease or hold, then assume that the gradient of pressure variation is relatively constant when control mode is invariant, finally compensate the filtered pressure with the pressure of last two control cycles. The typical equation is shown as Eq. (17), where the control period is 10ms and the filter is a 16-point average filter. Figure 6 illustrates experiment of the same wheel cylinder pressure increase operation with that of Fig. 5. The wheel cylinder pressure sensor output, the actual brake wheel cylinder pressure, estimated brake wheel cylinder pressure and filtered pressure are illustrated in this figure. The figure shows that the digital filter above and its compensation is quite good because the filtered curve is quite smooth, which offers the pressure controller the possibility to control pressure more accurately and smoothly. Meanwhile, the filtered pressure with fewer fluctuation leads to the reduction of the actuation time of solenoid valves, which can improve the longevity of the solenoid valves and the comfortableness of the vehicle. The effectiveness of the post process of the estimated pressure is verified.

![Graph showing wheel cylinder pressure post processing](image_url)

Fig. 6 Post processing of the data of the estimation of real wheel cylinder pressure sensor. The operation condition is the same as that of Figure 5. The filtered wheel cylinder pressure (green) is quite smooth compared with the estimated pressure (red). The mitigation of the fluctuation brings large benefit for the pressure control, which will reduce the actuation time of the solenoid valves, improve the longevity of the solenoid valves and the comfortableness of vehicle.
5. A device to reduce fluctuation

Figure 7 shows different experimental results of two test bench. One system has larger fluctuation and overshoot than the other as for the real cylinder pressure. Therefore, the characteristics of brake system have an obvious influence on the pressure control. Obviously, the second one with smaller actual cylinder pressure fluctuation can achieve a better performance in the brake operation. It is not only simpler for the pressure controller to acquire and make use of the real cylinder pressure, but also more comfortable for the vehicle.

![Graph 1](image1)

Fig. 7 The output of pressure sensor (red) and the real wheel cylinder pressure (blue) of two test bench. The real wheel cylinder pressure is gained by the pressure sensor mounted near the wheel cylinder. Operation condition is increase the wheel cylinder pressure by step. The upper test bench has a larger fluctuation and overshoot compared with the lower one, making it difficult to control the pressure accurately and smoothly.

To mitigate the fluctuation and the overshoot of the brake transmission line in the structure design point of view, a buffer device (c3 in Fig.8) is supposed to be mounted between the brake hose and the wheel cylinder, whose characteristics can be simplified into a big capacitor as shown in Fig. 8.
Fig. 8 The buffer device (c3) is added near the wheel cylinder, to get a better brake regulation performance with less fluctuation and overshoot. A solenoid valve (v3) is applied to cut off the connection between the wheel cylinder and the buffer device, if a high pressure increase rate is needed, such as emergency braking.

Figure 9 illustrates the updated bond graph with the buffer device, which is represented by C3. The step response characteristics of the brake transmission line with different buffer device capacitor C3 is shown in Fig. 10. The curves in Fig. 10 show that larger C3 leads to less fluctuation and overshoot, but with a longer respond time. Therefore, a solenoid valve (v3 in Fig. 8) is placed between the buffer device and the wheel cylinder. The valve is open in conventional braking to provide a smoother pressure respond, but in the emergency braking the valve is closed to obtain a faster pressure response.

Fig. 9 The bond graph to improve the performance of the original brake transmission lines. A buffer device with a big fluidic capacitor is added between the brake hose and the wheel cylinder.
Fig. 10 The step response characteristics of the brake transmission line with different buffer device capacitor $C_3$. A larger $C_3$ can achieve less fluctuation and overshoot but a longer respond time.

6. Conclusion

This paper introduces a method for estimating brake wheel cylinder pressure. System model including brake tube, brake hose and wheel cylinder is established by using bond graph. Differential equations are derived from the bond graph. By Laplace transformation and discretization, the transfer function and the difference equation are obtained. The estimation of real cylinder pressure is realized by utilizing the difference equation. Test results show that the estimated cylinder pressure is consistent with the real one, which demonstrate the method of estimation is effective.

A digital filter is applied to the result of estimation before being used in feedback control in order to mitigate the fluctuation in the brake wheel cylinder. By analyzing data of brake wheel cylinder pressure regulation, a multi-point average filter is proposed to mitigate the fluctuation of the real pressure. The delay introduced by the digital filter is compensated by utilizing the fact that the gradient of pressure regulation remains almost the same in a single activation mode. The effectiveness of the digital filter and its compensation is verified by bench test.

A buffer device is introduced to improve the transient characteristics of the braking system, aiming at the decrease of fluctuation and overshoot. The buffer has a characteristic of large hydraulic capacitor and is mounted near the wheel cylinder. The decrease of fluctuation and overshoot during the pressure regulation operation is verified by simulations.

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