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High-Precision Modulation of A Safety-Critical Cyber-Physical System: Control Synthesis and Experimental Validation

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Abstract— As a typical safety-critical cyber-physical system, the brake-by-wire system, which provides improved control and enhanced performance, shows promise as an automotive braking scheme. In addition, the BBW system can be incorporated into all types of vehicles, including conventional vehicle with an internal combustion engine, electrified vehicles, and intelligent vehicles. However, the high cost and complexity of existing BBW systems limit their usage. In this paper, we consider a novel BBW system that we have developed, which has the advantages of having a simple structure and low cost. To enhance control accuracy of hydraulic pressure modulation and reduce noise, two pressure modulation methods, namely, the hydraulic pump based pressure modulation (HPBPM) and the close-loop pressure-difference-limiting (CLPDL) modulation, were proposed. So as to demonstrate the performance of the proposed approach, hardware-in-the-loop (HiL) experiments were conducted. The experimental results indicated that the proposed BBW system, which incorporates the CLPDL control method, appears to be a promising system in terms of overall performance, including control performance, noise reduction, and system simplification.

Index Terms— Brake-by-wire system, hydraulic modulation, on/off valve, PWM control, experimental validation.

I. INTRODUCTION

CYBER physical systems (CPS), which are distributed, networked systems that fuse computation and physical processes, exhibits a multidisciplinary nature and have recently become a research focus [1]. As a typical application of the safety-critical CPS, the brake-by-wire system, which mechanically decouples the brake pedal with downstream brake circuits, features the flexibility of the braking circuit arrangement and high-precision pressure modulation. Because of the inherent pedal-decoupling, the BBW system is usually chosen to be the platform for implementing brake blending control of different forms of electrified vehicles, [2,3]. Moreover, the possibility of fast and accurate modulation of mechatronic actuators is the foundation of many vehicle dynamics control systems and intelligent vehicles [4-7].

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As the BBW system promises benefits including efficiency and precision, it has become a popular area, attracting increasingly interest from both industry and academia. Currently there are two types of BBW system that have been developed: the electrohydraulic brake (EHB) and the electromechanical brake (EMB). In EHB system, the brake pressure is established by a hydraulic pump which is driven by an electric motor. In addition, high pressure accumulators and proportional valves are always equipped to achieve accurate hydraulic pressure control and noise reduction [8-10]. While the EMB solution removes all the hydraulic components, instead, it adopts an electric motor as actuator to provide braking torque, largely reducing the system complexity [11-13]. Although existing BBW systems can achieve continuous modulation of the braking torque, the high cost of proportional valves and the complexity of high pressure accumulators and actuators of EMB limit their usage. This limitation motivated us to develop a new type of BBW system that simply adds conventional actuators and sensors to available hydraulic braking layouts, realizing an overall control performance equivalent to that of an existing EHB.

In conventional hydraulic braking systems, the low-cost switching valves are widely utilized. By applying pulse-width modulation (PWM) control, it can achieve an acceptable control accuracy. However, the inherent discrete behavior of on/off valves causes not only the degradation of the pressure modulation accuracy but also an increase in the high frequency noise that passengers find most intolerable. Thus, several comprehensive research studies have been conducted into the areas of parameter design and control methods for on/off valves to address the above problems. In [14], valve parameter optimizations were conducted to enhance the dynamic performance of hydraulic valves. The relationship between dynamic performance and valve geometric parameters were analyzed. In [15], in order to compensate the control precision degradation caused by the discrete nature of actuator dynamics, a sliding mode control based method was proposed. Using the similar idea, a modified PWM based algorithm was proposed to enhance the responsiveness of on-off valves [16]. A learning-based method was implemented to guarantee the effectiveness of the proposed approach in the face of various external disturbances. In [17], to mimic the performance of the proportional valve, a PWM-based modulation approach with a high frequency ranging from 2 to 5 kHz was developed for on/off valves. In [18], an observer was designed for position estimation of valve core; together with the sliding mode control

technique which is similar to [15], the pressure modulation accuracy was improved. Nevertheless, because of the discrete nonlinearity of on-off valves, the existing control methods mentioned above can hardly achieve a desirable linear performance as a proportional valve behaves. Moreover, the noise reduction has been rarely considered in the existing studies.

A novel quasi-BBW system and a linear control method for on/off valves were proposed in [2, 19-22]. However, the proposed system can only decouple the mechanical connection between the braking pedal and the hydraulic brake circuits in some specific cases. Moreover, the proposed linear control method is an open loop method, which is sensitive to external disturbances. Motivated by the system and the control method in [2, 19-22], in this paper, a novel BBW system is designed simply by adding conventional actuators and sensors to the usual hydraulic layouts. To improve the pressure modulation accuracy of the hydraulic control system and reduce noise, two pressure modulation methods, namely, the hydraulic pump based pressure modulation (HPBPM) and the close-loop pressure-difference-limiting (CLPDL) modulation, are proposed and compared. The HPBPM is developed to increase the hydraulic braking modulation precision by coordinating the hydraulic pump and corresponding solenoid valves, while CLPDL leverages on the valve current control technology. The main novelty and contribution of this work are as follows: 1) the novel BBW system is developed simply by adding conventional actuators and sensors to the conventional hydraulic component layout that is simpler than the system in [23]; 2) two pressure modulation methods of the newly developed BBW system, namely, HPBPM and CLPDL, are proposed, and the control performance and noise were compared via HiL validation; and 3) the CLPDL control method is applied to a brake blending control strategy of EVs, validating the effectiveness and feasibility of the CLPDL method in actual implementation.

The remaining parts of this work are organized as follows: in Section II, the system and working principle of the novel BBW is described. In Section III, the HPBPM for the BBW system is proposed. In Section IV, the linear correlation between the pressure difference across the valve and applied coil current is analyzed, and the CLPDL method of the BBW system is developed. In Section V, the performances with respect to tracking control and noise level of the system under HPBPM and CLPDL methods are compared; the effectiveness of the CLPDL approach is demonstrated by a normal regenerative braking scenario. In Section VI, the study's conclusions are provided.

II. DESCRIPTION OF THE NEWLY PROPOSED BBW SYSTEM

A novel BBW system, which can be implemented on all types of vehicles, including traditional internal combustion engine (ICE) vehicles and different forms of electrified and intelligent ones, is schematically depicted in Fig. 1. Unlike the standard BBW system—for example, Toyota's electronically controlled brake (ECB), [23]—the high pressure accumulator and linear valves are not needed in the proposed system; all of

the valves in the proposed system are normal on/off valves. Only two relief valves (*RV1*, *RV2*) are added, unlike the standard electronic stability program (ESP) systems [24], the differences between proposed BBW system and conventional BBW system have been highlighted in Fig. 1. The pressures of the master cylinder and each wheel cylinder are measured by the corresponding pressure sensors.

Fig. 1. Diagram of the structure of the proposed BBW system.

MC: master cylinder; MCP: master cylinder pressure sensor; PSS: braking pedal stroke simulator; ISV: isolation valve; RV: relief valve; IV: inlet valve; OV: outlet valve; WCP: wheel cylinder pressure sensor; FR: front right wheel; FL: front left wheel; RL: rear left wheel; RR: rear right wheel; NO: normally open valve; NC: normally closed valve.

To achieve good tracking performance of the target pressure, the wheel cylinder pressure should be modulated precisely. The state machine for wheel cylinder pressure modulation is illustrated in Fig. 2. The wheel cylinder pressure is threshold controlled and has three modes, including pressure increase, pressure hold and pressure decrease. These modes can shift from one to another by judging the logic thresholds, which are the various differences between the actual and target wheel cylinder pressures. The operation principle of the developed BBW system is explained in details below.

Fig. 2. State machine of the wheel cylinder pressure modulation (error: difference between the actual and the target wheel cylinder pressures; brk_cmd: driver braking intention that can be interpreted from the master cylinder pressure).

During the normal brake mode, the four normally open isolation valves are activated and closed. Then, the master cylinder pressure, which represents driver's deceleration demand, is physically isolated from the brake pressures of the

downstream wheel cylinders. The braking fluid in the master cylinder is directly passed into the pedal stroke simulator, ensuring good brake pedaling feeling. As all of the wheel cylinders have the same braking circuit layout, we take the example of the front right wheel to explain the pressure modulation process.

During the pressure increase process, two hydraulic pumps are driven by the hydraulic pump-motor, which is controlled by PWM signals. Brake fluid is pumped directly from the tank instead of the master cylinder, leaving the brake pedaling feel unaffected. The opening of the normally closed relief valve *RVI* is controlled to keep the upper stream pressure of the inlet valve *IVI* constant or at the upper limit, depending on the relief valve (*RVI*) control method. By adjusting the opening of the inlet valve *IVI*, the wheel cylinder pressure can be modulated. The precise pressure increase is achieved by the coordinated control of the hydraulic pump, relief valve, and inlet valve. The detailed control algorithms for relief and inlet valves will be investigated in Sections 3 and 4.

During the pressure decrease process, the hydraulic pumps stop and the relief valve *RVI* is fully open. At the same time, the *IVI* is fully closed, and the outlet valve *OVI* is controlled with PWM signals, which modulates the decreasing ratio of the brake pressure in wheel cylinder. The hydraulic fluid flows from the wheel cylinder to the tank directly.

During the pressure hold process, both inlet valve *IVI* and outlet valve *OVI* are fully closed. The pump operates at a low speed, and the relief valve *RVI* is opened to a certain level. This property keeps the pressure of the entire braking circuit above a certain level, which ensures responsiveness for the next pressure increase process.

As the pressure of the four wheel cylinders can be controlled independently, the proposed BBW system can also be used for other control functions, including anti-lock braking, traction control, and stability control program. In addition, this novel design can be installed on vehicles with any type of braking circuit arrangement. Note that when the hydraulic pump fails, as in a conventional vehicle, a backup circuit ensures the pressure build up by drivers; this is highlighted in Fig. 1.

III. CONTROL ALGORITHM DESIGN FOR HYDRAULIC PUMP BASED PRESSURE MODULATION

As mentioned in the introductory section, the following two performances must be guaranteed: first, the tracking performance of the target wheel cylinder pressure; second, low noise during BBW system operation. Thus, the control algorithms for the on/off valves and hydraulic pump are worth investigating. Two different control schemes have been implemented; these schemes are presented in separate sections (3 and 4).

The overall controller configuration for pressure modulation of the two different scheme is illustrated in Fig. 3. In the first scheme, we aimed at achieving the control goals by employing the conventional PWM control technique for inlet valves, and a newly proposed coil current control technique is implemented in the second scheme. The first scheme is introduced as follows.

Fig. 3. Overall controller configuration of pressure modulation.

The noise generated by the BBW system mainly originates from two parts: the on/off valve and the hydraulic pump. Compared to the valve generated noise, which is mostly centralized in the high frequency range, the hydraulic pump contributes more to low frequency noise. However, the high frequency noise is most perceivable and intolerable for passengers, especially when the opening of an on/off valve is being modulated frequently. Thus, an intuitive method to reduce the annoying high frequency noise is to keep the PWM command of the inlet valve constant. The tracking performance of the brake pressure in a wheel cylinder is guaranteed by modulation of the rotational speed of the pump-motor. To illustrate the main idea of the HPBPM method, a simple but valid spring and position model [17-22] is used. The dynamics of the wheel cylinder pressure p_w and the displacement of the spring x_w of the cylinder can be given by:

$$dp_w = \frac{k_w}{\pi r_w^2} \cdot \frac{\sigma_{PWM}}{100} dx_w \quad (1)$$

$$dx_w = \frac{Q}{\pi r_w^2} dt \quad (2)$$

where k_w represents the stiffness coefficient of the spring, r_w is the of the piston in the brake cylinder, σ_{PWM} (which varies within 0-100) represents the value of the PWM signal for inlet valve, and Q is the flow rate of the fluid passing through the valve. According to the working principle of the hydraulic pump, the flow rate Q is expressed as,

$$Q = 2\pi r_{pump}^2 \eta_{pump} e_{pump} \omega_{hyd_motor} \quad (3)$$

where r_{pump} , η_{pump} , e_{pump} and ω_{hyd_motor} are the radius of the piston, efficiency, eccentricity, and rotational speed of the pump-motor, respectively. By substituting equations (2) and (3) in (1), the wheel cylinder pressure p_w can be rewritten as

$$\dot{p}_w = \frac{2k_w r_{pump}^2 e_{pump} \eta_{pump}}{\pi r_w^4} \cdot \frac{\sigma_{PWM}}{100} \cdot \omega_{hyd_motor} \quad (4)$$

It can be seen from equation (4) that when the PWM command value of the inlet valve is constant, and the dynamics of the pressure is directly decided by the rotational velocity of the pump-motor. Based on the property of PWM control, a suitable PWM command value for the inlet valve should be chosen based on the following criteria. Regarding the first criterion, the valve generated noise becomes most perceivable and intolerable when the PWM command value is approximately 50, at which the impact between valve coil and seat is most dramatic.

Thus, less noise will be generated when the inlet valve PWM command chosen is closer to 0 or 100. Regarding the second criterion, because of the discontinuous property of the

hydraulic pump, pressure fluctuations will be generated. These fluctuations can be reduced by opening the inlet valve up to a small level. In other words, the PWM command is supposed to be close to 100 for a normally open inlet valve.

Fig. 4. (a) Relationship between the wheel cylinder pressure and the value of the PWM signal of the pump-motor (with the PWM value of the inlet valve at 80); (b) relationship between the wheel cylinder pressure and different PWM command values of the pump-motor (with the PWM value of the inlet valve at 80).

Figs. 4 (a) shows the relationship between the variation rate of the brake pressure in wheel cylinder and the value of the PWM signal of the pump-motor. In both experiments, the inlet valve PWM value is set to 80, at which the noise is acceptable, and the pressure fluctuation generated by the hydraulic pump is adequately suppressed. A linear relationship between the variation rate of the wheel cylinder pressure and the value of the PWM signal for the pump-motor is observed, as shown in Fig. 4 (b). This result validates the correctness of equation (4).

To improve the tracking performance and robustness to external disturbances, a closed-loop control algorithm for the hydraulic pump-motor is developed, as illustrated in Fig. 5. The feed-forward lookup map is derived by experiments, as shown in Fig. 4. Because the wheel cylinder pressure can be obtained by the pressure sensor, feedback control is utilized to offset the wheel cylinder pressure tracking error. Note that the relief valve, which is controlled inactively, plays a role only of safety outlet valves in the HPBPM control method. When the upper stream pressure of the inlet valve exceeds 15 MPa, the relief valve will be pushed open to release the excess pressure. At other times, the upper stream pressure of the inlet valve is equal to the outlet pressure of the hydraulic pump.

IV. CONTROL ALGORITHM DESIGN FOR CLOSE-LOOP PRESSURE-DIFFERENCE-LIMITING MODULATION

The coil current based valve control method, which is different from the control method discussed in the last section, is explored in this section. A close-loop modulation method is developed based on the above discussed linear property for on/off valves.

A. Linear modulation of on/off valve

The relative position of a normally open valve in closed-state is illustrated in Fig. 6. Taking the lowest point which valve core can reach as the origin, the local coordinate system of the valves can be set up [22].

Fig. 6. Diagram of the coordinate system of the inlet valves.

Suppose the valve core comes in contact with the valve seat after being energized, the axial forces applied on the valve can be represented as

$$-F_e + F_s + F_h + F_N \sin \alpha = 0 \quad (5)$$

where F_e , F_s , F_h , F_N and α are the electromagnetic force, spring, hydraulic force, supportive force, and valve seat cone angle, respectively [22].

The valve can be defined to be in a critical balanced position when the following two conditions are satisfied: first, the valve core is in contact with the valve seat; second, the contact force between valve core and valve seat is zero, i.e., $F_N = 0$. In this state, the valve core is just about to leave the valve seat and the force balance equation in (5) is simplified as

$$-F_e + F_s + F_h = 0 \quad (6)$$

In equation (5), the only term that can be actively controlled is the electromagnetic force F_e . Thus, to realize the critical balanced state, the electromagnetic force should be regulated in a way that makes $F_N = 0$. The electromagnetic force can be expressed as

$$F_e = \frac{(NI)^2}{2R_g l} \quad (7)$$

where I and N are the applied current and turns number of the coil, respectively. l and R_g are the length and magnetic reluctance of air gap [22]. Then, we linearize equation (7) at a fixed operation point and reform it as:

$$\begin{aligned} F_e &= \left. \frac{\partial F_e}{\partial I} \right|_{I=I(t)} \cdot I(t) + \left. \frac{\partial F_e}{\partial x_v} \right|_{x_v=x_v(t)} \cdot x_v(t) \\ &= K_i I(t) + K_{x_v} x_v(t) \end{aligned} \quad (8)$$

Fig. 5. Structural diagram of the PWM control algorithm for wheel cylinder pressure.

where x_v is the displacement of the valve core, K_i and K_{x_v} are the first-order current-force and displacement-force coefficient.

Note that, the displacement of valve core is $x_v = 0$ when the critical balanced position is reached. Then, equation (8) can be simplified as

$$F_e = K_i I(t) \quad (9)$$

Based on the definition of the OX coordinate in Fig. 6, the spring force of a normally open valve is expressed as

$$F_s = k_s (x_0 + x_m - x_v) \quad (10)$$

where x_0 , x_m and k_s present the pre-tension displacement, maximum displacement and the stiffness of the return spring, respectively. Besides, the spring force can be simplified in this state as

$$F_s = k_s (x_0 + x_m) \quad (11)$$

When the valve core reaches the fully closed position, the F_h is decided by the pressure drop Δp between valve inlet and outlet and A_s , which is the surface area exposed to the fluid in axial direction. F_h , A_s , and Δp are calculated as follows [22]:

$$F_h = \Delta p \cdot A_s \quad (12)$$

$$A_s = \pi R_v^2 (\cos \alpha)^2 \quad (13)$$

$$\Delta p = p_{in} - p_{out} \quad (14)$$

where R_v is the spherical radius in the valve core.

Substituting equations (9), (11), and (12) into equation (6), we can derive a linear correlation between coil current and the pressure difference at valve's critical balanced position [22]:

$$\Delta p = \frac{K_i}{\pi R_v^2 (\cos \alpha)^2} I - \frac{k_s (x_0 + x_m)}{\pi R_v^2 (\cos \alpha)^2} \quad (15)$$

Substituting equation (14) in equation (15), we can obtain the linear modulation property of valve outlet pressure by adjusting the coil current,

$$p_{out} = p_{in} - \frac{K_i}{\pi R_v^2 (\cos \alpha)^2} I + \frac{k_s (x_0 + x_m)}{\pi R_v^2 (\cos \alpha)^2} \quad (16)$$

If the input pressure p_{in} can be measured or estimated, we will be able to address the output pressure tracking problem by calculating the coil current command value based on equation (16) as has been done in [22]. However, the following two problems must be addressed to achieve better tracking performance of the output pressure. First, as mentioned in the introductory section, the control algorithm, which directly

implements equation (16) in an open-loop manner as done in [22], is not robust enough to disturbances. Second, unlike the case in [22], the value of the input pressure p_{in} can be acquired accurately by using the pressure sensor. However, the pressure of the valve input port cannot be measured in the proposed BBW system. Thus, a close-loop tracking control algorithm is required to be designed for the proposed system, ensuring wheel cylinder pressure control accuracy and low noise generation.

B. Close-loop pressure-difference-limiting control

The structure of the CLPDL control for hydraulic pressure modulation is shown in Fig. 7. To improve the robustness and accuracy of the control algorithm, a feed-forward and feed-back scheme is employed. The current map which reflects the linear characteristics as equation (15) shows is derived from the off-line experiments. A simple PI controller is utilized to eliminate external disturbances, such as temperature change and input pressure fluctuation.

p_{w_act} can be obtained by the hydraulic pressure sensor installed in the wheel cylinder. However, there is no sensor installed for measuring the inlet pressure p_{input} . Motivated by the methods adopted in conventional BBW systems, the ideal p_{input} is to be controlled to a stable value. To achieve a stable upstream pressure for the inlet valve, a coordinate control strategy for the pump-motor and relief valve is developed. Unlike the method in Section 3, the relief valve is controlled actively. Note that the relief valve is normally closed; thus, the relation shown in equation (16) needs to be rewritten as follows:

$$p_{out_relief} = p_{in_relief} - \frac{k_{s_relief} (x_0 + x_m)}{\pi R_{v_relief}^2 (\cos \alpha_{relief})^2} + \frac{K_{i_relief}}{\pi R_{v_relief}^2 (\cos \alpha_{relief})^2} I \quad (17)$$

where p_{in_relief} and p_{out_relief} are the input and output pressures of the relief valve, respectively. As indicated in Fig. 1, the output port of the relief valve is directly connected to the tank, where the pressure is zero; thus, $p_{out_relief} = 0$. The upstream pressure for the inlet valve is equal to p_{in_relief} ; therefore, equation (17) can be simplified as follows:

$$p_{in_relief} = \frac{k_{s_relief} (x_0 + x_m)}{\pi R_{v_relief}^2 (\cos \alpha_{relief})^2} - \frac{K_{i_relief}}{\pi R_{v_relief}^2 (\cos \alpha_{relief})^2} I \quad (18)$$

Based on the analysis above, by setting constant command values for the relief valve and pump-motor, the stable upstream pressure for the inlet valve can be derived.

Fig. 7. Structural diagram of the close-loop CLPDL control algorithm for wheel cylinder pressure modulation.

According to the working principle of the pressure-difference-limiting control, the upstream pressure for the input valve should be greater than or equal to the current driver braking intensity. In the developed BBW system, accurate modulation of the coil current is achieved by the current control chip, whose best working region is from 100 mA to 600 mA. Thus, the optimal value of the pressure difference is set within the range from 0 MPa to 4 MPa. A switched coordinated control strategy for relief valve and pump-motor has been designed, as shown in Fig. 8. The pressure increase mode has two sub-phases, denoted as low and high braking intensity mode, which can realize the different stable upper stream pressure for inlet valves. This property ensures that the current control chip works within its optimal region and that there is enough up stream pressure for the inlet valves. Hydraulic pump only works in the pressure hold and increase modes. Because no noise is generated when the valve is controlled by the current method, the noise is significantly reduced with CLPDL control.

Fig. 8. Schematic diagram of the switch coordinated control strategy for the relief valve and the pump-motor. Error: the difference between actual and target wheel cylinder pressures; brk_cmd: driver braking intention, which could be interpreted from the pressure in master cylinder; P_m: the pressure of master cylinder; pump PWM cmd: the PWM command value of pump-motor; relief valve current cmd: the current command value for relief valves.

Note that the current command value for relief valves RV_cmd1 , RV_cmd2 , and RV_cmd3 , are different for the brake cylinders in front and rear wheels. However, for the same wheel cylinder, the relative relationship among the three values is the same, which given by is $0 < RV_cmd3 < RV_cmd1 < RV_cmd2$.

V. HARDWARE-IN-THE-LOOP TEST RESULTS

In this section, to validate and compare the control performance of the proposed approaches, the HiL experiments were carried out with the new BBW system designed in Section 2. The dSPACE Autobox serves as a real-time platform, in which the models of vehicle motion, electric motor and battery are running. Meanwhile, the real braking system including hydraulic circuits, master and wheel cylinders, a vacuum booster and brake pedal, is equipped in the test bench. Several key parameters of the models used in tests are listed in Table 1.

TABLE 1.
KEY PARAMETERS OF THE VEHICLE [25].

Parameter	Value	Unit
Total mass (m)	1360	kg
Wheel base (L)	2.50	m
Coefficient of air resistance (C_D)	0.32	-
Nominal radius of tire (r)	0.295	m
Gear ratio	7.881	-

The Freescale BA13 is used as the current control chip to generate a stably controlled coil current. The proposed hydraulic control algorithms are implemented in a real-time controller. The controller communicates with the dSPACE simulation platform through CAN (controller area network) bus. Fig. 9 shows the overall experimental setup of the test bench.

First, the tracking performances of a ramp input under HPBPM and CLPDL control are compared. Next, a normal regenerative braking scenario is used to test the feasibility of the CLPDL control algorithm through actual implementation.

Fig. 9. HiL test bench control algorithm tests.

A. Comparison of HPBPM and CLPDL control

So as to validate the performance of the designed modulation algorithms, the HiL testing scenario is set as: a normal deceleration is taken the operation condition; the objective value of the master cylinder pressure is set as a ramp input stabilizing at 3 MPa to mimic the normal braking intensity, and the adhesion coefficient is set at a high value of 0.8, indicating a dry surface.

The HiL test results with HPBPM are shown in Fig. 10(a). At 0.3 s, the target master cylinder pressure starts to increase and the hydraulic pump driven by the electric motor builds up the pressure rapidly during the time period from 0.3 s to 1.3 s. However, as the hydraulic pump requires 200-300 ms to establish negative pressure, there exists a short delay before pressure actually begins to build up at the beginning of braking. This delay degrades the pressure tracking performance. It was observed that the value of the PWM command decreases as the difference between actual and target pressures decrease. There are two steps in the PWM signal for the pump-motor: one is from 1.3 s to 1.5 s, and the other is from 2.4 s to the end. The reason for the two steps is that the system is in the pressure hold mode and the pump is operating at a constant speed to maintain the pressure level of the whole braking circuit, thereby ensuring the responsiveness for the next pressure increase mode.

Fig. 10. (a) HiL test results of the hydraulic pump based pressure modulation for the pressure tracking scenario; (b) HiL test results of the CLPDL for the pressure tracking scenario.

The same HiL test scenario is implemented to test the CLPDL control algorithm. The experimental data are presented in Fig. 10(b). As the target wheel pressure increases, the coil current of the inlet valve jumps from 0 mA to 300 mA, allowing

the brake fluid to flow into the wheel cylinder. The coil current value of the relief valve remains constant, thereby cooperating with the hydraulic pump to achieve a stable upstream pressure of the inlet valve. With the difference between the target and actual wheel cylinder pressures decreasing, the system transitions into pressure hold mode, and the value of the coil current is modulated to be smaller, thereby fully closing the inlet valve.

The superior tracking performance of the CLPDL control algorithm is evident from the comparison with the HPBPM control algorithm. In addition, the CLPDL control algorithm, in which there is no noise generated from the valves, provides more advantages over HPBPM control, in which PWM control for the pump-motor and inlet valve is used. The tracking performances of the target wheel cylinder pressure with HPBPM, CLPDL and conventional PWM control, which are adopted in [2, 19, 20], are compared quantitatively in Table 2. The average absolute value and RMS of pressure tracking error with CLPDL control are 0.0875 MPa and 0.1049 MPa, which are 61.69% and 68.13%, respectively, better than those with HPBPM and 47.45% and 55.89%, respectively, better than those with conventional PWM control. This finding indicates the enhancement of pressure modulation accuracy with the CLPDL control method.

TABLE 2.
COMPARISON OF PRESSURE TRACKING PERFORMANCE UNDER CONVENTIONAL PWM, HPBPM AND CLPDL CONTROL METHODS.

	Conventional PWM	HPBPM	CLPDL
Average absolute value of the pressure tracking error (MPa)	0.1665	0.2284	0.0875
RMS (MPa)	0.2378	0.3291	0.1049

According to the above discussed HiL experimental results, with HPBPM, although the system noise is reduced compared to that with conventional PWM control for the inlet valve, the tracking performance of the target pressure is not guaranteed. The average absolute value and RMS of pressure tracking error with HPBPM control are 37.18% and 38.39%, respectively, which are worse than those with conventional PWM control. Thus, from both perspectives of control accuracy and noise reduction, the newly proposed CLPDL control appears to be an effective method to address the wheel cylinder pressure modulation problem.

B. Brake blending algorithm based on CLPDL modulation

The brake blending performance between regenerative braking and hydraulic braking has a significant effect upon the regeneration efficiency and brake safety. Thus, the requirement of hydraulic pressure modulation accuracy is crucial. In addition, unlike emergency conditions, such as anti-lock braking, brake comfort and noise level, are also evaluated during regenerative braking, making the hydraulic pressure modulation even more difficult. To verify the performance of the CLPDL algorithm in real world applications, a typical brake blending test of EVs is conducted on the HiL test bench. The set-up of the testing scenarios is the same as the one adopted in the tracking performance test. The brake maneuver is initialized

with the vehicle speed of 72 km/h. The target vehicle is a front-wheel driven one, which equips with a centralized electric motor. The same brake blending control strategy as introduced in [21, 22, 25] is employed. The only modification is the replacement of the conventional PWM valve control with the CLPDL control algorithm. The details of the brake blending algorithm are presented in [21, 22, 25].

The HiL test results for the front axle are shown in Fig. 11 (a). At approximately 0.8 s, the target master cylinder pressure begins to increase. The electric motor applies the braking torque gradually on the front axle. According to the brake blending strategy, the hydraulic braking, which is under the control of the CLPDL algorithm, decreases when the regenerative braking torque increases. After 1.5 s, the target value of master cylinder pressure remains stable at approximately 3 MPa, and accordingly the hydraulic brake pressure is modulated dynamically under CLPDL control.

During the pressure increase phase, the outlet valve of the brake cylinder is not energized. Starting from 6.5 s, the regenerative braking torque gradually decreases and finally down to zero because of the characteristics of the electric motor. Thus, the hydraulic brake pressure needs to be increased accordingly to fulfill the brake demand. Under the control of CLPDL, the hydraulic braking pressure changes smoothly and the driver's braking intention is well satisfied.

(a)

(b)

Fig. 11. (a) HiL test results of the CLPDL-based brake blending control strategy for the front axle; (b) HiL test results of the CLPDL-based brake blending control strategy for the rear axle.

The HiL tests results for brake cylinders in rear axle are shown in Fig. 11 (b). Since there is no regenerative braking applied on the rear axle, the control task is same as those illustrated in Section 5.A. As shown in Fig. 11 (b), the target pressure of the rear wheel cylinder is well-tracked. The overall experimental results show that the target brake pressure is followed very well through valve modulation with the proposed approach, and the blended braking brakes at the front wheels are coordinated well, thereby validating the effectiveness of the CLPDL control algorithm of the hydraulic brake pressure.

VI. CONCLUSIONS

High-precision control of mechatronics is an important foundation of the development of safe, smart and sustainable CPS [26-31]. In this paper, a typical safety-critical CPS, i.e. the BBW system, was introduced. Compared to the existing BBW system, the newly developed system enjoys the advantage of a simple structure and low cost because only conventional valves and sensors are added to the usual hydraulic layouts. Two pressure modulation methods, namely, the HPBPM and CLPDL modulation, were proposed to improve the modulation precision of hydraulic brake pressure and reduce valve's operation noise as well. Experiments were conducted in HiL test rig to demonstrate the performance of the proposed control methods. Experimental results showed that, in spite of the reduction in noise in the HPBPM control method, when compared to the conventional PWM control, control accuracy is not guaranteed. The CLPDL method achieves both good performance of tracking target pressure and reduced noise. To further validate the feasibility of the newly proposed BBW system and the CLPDL control method, a typical regenerative braking scenario was utilized. The CLPDL control method was implemented within a regenerative braking strategy. The HiL test results illustrated that both the front and rear wheel cylinders fulfilled the driver braking intention and that the regenerative braking and hydraulic braking cooperate well with each other. For the future work, real vehicle tests will be conducted, and qualitative comparisons between proposed BBW system and existed BBW systems will be explored.

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