Electro-hydraulic Composite Emergency Braking System Based on Frequency Reconstruction Control

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Abstract. For four-wheel independently driven (FWID) electric vehicle (EV), in order to guarantee braking reliability, hydraulic braking system and motor anti-drag braking system are usually used at the same time. In this paper, a novel electro-hydraulic composite braking control strategy is given to solve the distribution problem of braking torque between hydraulic braking torque and motor anti-drag braking torque. The high-frequency part of the desired braking torque is distributed to the motor brake through a high-pass filter so as to fully utilize the fast response characteristics of the motor. Meanwhile, the low-frequency part of the desired braking torque is distributed to the hydraulic brake through a low-pass filter so as to reduce the load of the motor. The cost functions are designed to determine the cut-off frequency of low-pass and high-pass filters. Through the joint simulation of Matlab/Simulink and TruckSim, the braking effect of the electro-hydraulic composite braking is better than that of pure hydraulic braking with no motor to modulate the braking torque.

1. Introduction

In the automotive industry, safety has always been the focus of attention. For FWID EV, in order to ensure the reliability of braking, in addition to using the distributed drive motor to generate anti-drag braking torque, generally we need equipping a hydraulic braking system at the same time.

Many recent studies of electro-hydraulic composite braking particularly focused on improving energy recovery efficiency under the regenerative braking conditions [1][2], overlooking the importance of braking efficiency under critical driving situations. The core of electro-hydraulic composite braking is to determine the distribution ratio of the electro power to hydraulic power. C. Wang et al. formulated the optimal braking force distribution method of hydraulic braking and regenerative braking to ensure the driver obtains the same braking sense [3]. J. Wang et al. proposed a nonlinear variable voltage charging control law to balance the performance difference of every cell in the power battery pack, meanwhile used compensation control to calculate the hydraulic braking torque [4]. Yuan et al. designed a motor braking system to meet the total braking demand preferentially and compensated the remaining part of braking demand by hydraulic brake by a tri-level close-loop control architecture [5]. Kumar et al. adjusted the proportions of regenerative braking and friction braking by the displacement of master cylinder plunger and brake power in demand[6].

However, due to the non-zero average braking torque of the motor, and the influence of the motor power and battery charging reverse power, the motor can not provide enough anti-drag braking torque
under emergency braking conditions. Therefore, the existing electro-hydraulic composite braking scheme will generally switch to the single mechanical braking mode in case of emergency braking to ensure safety[7].

In this paper, a novel electro-hydraulic composite braking system based on frequency reconstruction is proposed. Such a control strategy for parallel system can not only improve the maximum braking torque capacity but also further improve the performance of the vehicle braking system under emergency braking conditions.

2. Electro-hydraulic composite braking system

2.1. Structure of the composite braking system

Figure 1 shows a simple diagram of an electro-hydraulic composite braking system designed for FWID EVs. When the driver steps down the pedal, both pedal velocity and pedal position signals are transmitted to the ECU. Then ECU can obtain the driver’s intention according to the given thresholds of pedal velocity and displacement.

![Figure 1. Schematic diagram of the electro-hydraulic composite braking system](image)

The braking torque commands allocated to the wheel drive motor and the wheel cylinder pressure of the hydraulic braking system are directly calculated by the ECU in combination with the current braking mode of the vehicle. The motor braking system is the in-wheel motors corresponding to the wheels. The hydraulic brake is composed of a brake master cylinder, pressure boost solenoid valves, pressure reducing solenoid valves, brake motor (brake hydraulic motor), and brake wheel cylinders of each wheel.

2.2. Math model of Electro-hydraulic composite braking system

From a macro perspective, $T_h$ is the output torque of the motor brake, $T_{h\text{\,desire}}$ is the control input signal of the motor brake, $T_{h\text{\,max}}$ is the maximum output torque of the hydraulic brake. Supposing $T_{h\text{\,desire}}$ is the input of the hydraulic braking system. Then the hydraulic brake can be simplified as the product of a first-order delay system and a saturation link:

$$
T_h = \begin{cases} 
G_h(s) \times T_{h\text{\,max}}, & T_h \geq T_{h\text{\,max}} \\
G_h(s) \times T_{h\text{\,desire}}, & T_h \geq T_{h\text{\,desire}} \geq 0 \\
0, & T_{h\text{\,desire}} < 0
\end{cases}
$$

(1)
where $G_h(s)$ is a simplified hydraulic brake first-order delay system$^{[3]}$:

$$G_h(s) = \frac{T_h}{T_{h_{desire}}} = \frac{1}{\tau_h s + 1} e^{-\tau_{D1}s}$$  \hspace{1cm} (2)

where $\tau_h$ is the time constant of the hydraulic braking system and represents the dead time of the hydraulic valve; $\tau_{D1}$ is the delay time of hydraulic braking system, which indicates the delay of CAN signal and other mechanical systems.

For the motor part, the maximum motor braking torque is subject to its torque-speed characteristics$^{[8]}$. Accordingly, The upper bound of motor output torque $T_{m}^{max}$ can be expressed as:

$$T_{m}^{max}(r) = \min\{T_{m}^{R}, \frac{P_{m}^{R}}{r}\}$$  \hspace{1cm} (3)

where $T_{m}^{R}$ is the rated torque of the drive motor, and $P_{m}^{R}$ is the rated power of the motor, these two parameters depend on the motor lectotype; $r$ stands for the rotor angular velocity, which can be represented to the product of the wheel velocity and corresponding gain $K_w \cdot v_w$. Subsequently, the motor brake can be simplified as the product of another first-order delay system and that saturation link.

The simplified motor brake first-order delay system is:

$$G_m(s) = \frac{T_m}{T_{m_{desire}}} = \frac{1}{\tau_m s + 1} e^{-\tau_{D2}s}$$  \hspace{1cm} (4)

where $\tau_m$ is the time constant of the motor and represents the reaction time of the motor, $\tau_{D2}$ is the delay time of the motor brake control system, which represents the delay of can signal and other mechanical systems.

3. Adaptive frequency reconstruction control

3.1. Frequency description of desired braking torque

Assume that the braking condition will not change in a short time, thus we can take the braking torque in the previous period as a reference. The total desired braking torque $T_{desire}^{desire}$ is a time-domain signal. To study its characteristics and design adaptive frequency rules, it is necessary to transform it from time-domain to frequency-domain by using discrete Fourier transform (DFT).

If $x = \{x(0), x(1), \ldots, x(N-1)\}$ are $N$ sampling points and the sampling frequency is $f_s$, the desired torque $T_{desire}^{desire}$ signal of $N$ steps before the current moment, and its frequency domain signal $X(f)$'s expression is as follows:

$$X(f) = \sum_{k=0}^{N-1} x(k)e^{-j2\pi f_k}$$  \hspace{1cm} (5)

where the analysis frequencies are:

$$f = \{f_0, f_1, \cdots, f_{N-1}\} = \frac{k f_s}{N}, \; k = \{0, 1, \cdots, N-1\}$$  \hspace{1cm} (6)

The single-sided spectrum of desired baking torque after DFT is shown in figure 2. To make better use of the fast response characteristics of motor brake and reduce the impact of hydraulic system delay on the composite braking system under emergency braking condition, the analysis frequencies in equation(6) can be divided into three parts:

$$f = \{0, \frac{f_s}{N}, \cdots, \frac{k_f f_s}{N}, \frac{(k_f + 1) f_s}{N}, \cdots, \frac{k_n f_s}{N}, \frac{(k_n + 1) f_s}{N}, \cdots, \frac{(N-1) f_s}{N}\}$$  \hspace{1cm} (7)

hydraulic part  motor part  noise part
where \( \frac{k_c f_c}{N} = f_c \) is the cut-off frequency of the electro-hydraulic composite system. \( \frac{k_n f_n}{N} = f_n \) is the noise frequency. Then the analysis frequencies of desired braking torque are divided into three parts: (1) desired braking torque on the frequencies lower than the \( f_c \) are allocated to the hydraulic brake. (2) Signals in the frequency band between \( f_c \) and \( f_n \) are charged to the motor brake. (3) Signals on the frequencies exceed \( f_n \) where even the motor fails to meet the response requirements, is the noise part.

![Figure 2. The single-sided spectrum of desired baking torque](image)

**3.2. Optimal cut-off frequency**

The optimal cut-off frequency \( f_c \) is determined by designing the cost function from the energy point of view in each analysis frequency. For the hydraulic braking part:

\[
J_h(f_k) = \int_{t_1(f_k)}^{t_2(f_k)} P_1(f_k)[\sin(2\pi f_c t) - \sin(2\pi f_c (t - C_t))] dt
\]

where \( f_k \) is the \( k \)-th analysis frequencies of the set \( f \); \( P_1(f_k) \) is the single-sided power at \( f_k \) frequency; \( C_t \) is a given constant, the delay time of hydraulic brake response is generally selected. \( J_h(\cdot) \) is called delay cost function, also known as hydraulic cost function. Here:

\[
\begin{align*}
    t_1(f_k) &= \frac{\pi + 2\pi f_k C_t}{4\pi f_k} \\
    t_2(f_k) &= \frac{-\pi + 2\pi f_k C_t}{4\pi f_k}
\end{align*}
\]

Thus the meaning of delay cost function can be seen as the intersection area of curve \( P_1(f_k) \sin(2\pi f_c t) \) and curve \( P_1(f_k)[\sin(2\pi f_c (t - C_t))] \), \( t_1(f_k) \) and \( t_2(f_k) \) are exactly the abscissa values of the intersection points.

For the motor modulation part:

\[
J_m(f_k) = \int_{t_1(f_k)}^{t_2(f_k)} \max[P_1(f_k) \sin(2\pi f_c t) - C_L, 0] dt
\]

where \( C_L \) is overload constant and determined by the maximum torque of the motor. \( J_m(\cdot) \) is called the overload cost function, also known as the motor cost function. The cost function is a positive semidefinite value, when the power at the current analysis frequency \( f_k \) is less than the overload constant \( C_L \) the motor cost function is 0. Here:

\[
\begin{align*}
    t_1(f_k) &= \frac{1}{2\pi f_k} \arcsin \frac{C_L}{P_1(f_k)} \\
    t_2(f_k) &= \frac{1}{2\pi f_k} \left[ \pi - \arcsin \frac{C_L}{P_1(f_k)} \right]
\end{align*}
\]
By calculating the area of $P_1(f_k)\sin(2\pi f_k t)$ exceeding the overload constant $C_L$, within one cycle at the current analysis frequency $f_k$, the overload cost function indicates the influence of motor overload on the whole system at the current analysis frequency $f_k$. Thus $t_1(f_k)$ and $t_2(f_k)$ are exactly the abscissa values of the intersection points when $t \in (0, \pi)$.

In all analysis frequencies, the delay cost function of hydraulic braking part, and the overload cost function of motor braking part are accumulated. Thus the cut-off frequency $f_c = \frac{k_c f_k}{N}$ is determined by:

$$k_c = \arg\min \left[ \sum_{k=0}^{k_k} J_h(f_k) + \sum_{k=k_k}^{k_m} J_m(f_k) \right]$$ (12)

To solve the equation(12), the delay cost function and overload function are calculated at every analysis frequency considered. Then the values obtained are stored in the matrix $J_h \in \mathbb{R}_{1 \times k_k}$ and $J_m \in \mathbb{R}_{1 \times k_k}$. Thus the equation(12) can be converted to an integer linear programming problem:

$$\begin{align*}
\min & \quad J_h x + J_m (1 - x) \\
\text{s.t.} & \quad x_1 - x_2 \geq 0, \\
& \quad x_2 - x_1 \geq 0, \\
& \quad \vdots \\
& \quad x_{k_k-1} - x_{k_k} \geq 0, \\
& \quad 1 \geq x_1, x_2, \ldots, x_{k_k} \geq 0, \\
& \quad x_1, x_2, \ldots, x_{k_k} \in \mathbb{Z}
\end{align*}$$ (13)

The last element in vector $x = [x_1 \ x_2 \ \cdots \ x_{k_k}]^T$ that is not 1, is the index number $k_c$ of the cut-off frequency. Then the desire braking torque can be allocated to hydraulic braking subsystem and motor braking subsystem by reconstructing at cut-off frequency $f_c$. And the low-pass filter and high-pass filter in frequency reconstruction are achieved in this work by a first order inertial link:

$$T_{desire} = T_{desire}^h + T_{desire}^m$$

$$= \frac{f_c}{s + f_c} T_{desire}^h + \left( \frac{s}{s + f_c} \right) T_{desire}^m$$ (14)

4. Simulation results

In this section, a 2-axles school bus full vehicle model in TruckSim software is used to verify the effectiveness and working principle of the proposed frequency reconstruction control of electro-hydraulic composite braking system. The vehicle parameters used in the simulations are presented in table 1. The simulation step is 0.001s. Set the initial speed of the vehicle at 16.67m/s (60km/h) and make emergency braking after 2s driving. The desired torque will be decided by logic threshold control, the upper threshold of slip rate is 0.2 while the lower threshold is 0.15. When the speed is lower than 1.67m/s (6km/h), the braking strategy will switch to pure hydraulic braking and maintain a fixed wheel cylinder pressure. The tire-road coefficient of the working condition is set as 0.85. The results are as follows:

As shown in figure 3(a), the high-frequency part of desired braking torque is distributed to the motor brake while the hydraulic brake handles the high-frequency part. In figure 3(b), the cut-off frequency of left front wheel is adjusted constantly according to the desired braking torque signals. As is shown in figure 4(a), the vehicle with composite braking system decelerates to 1.67m/s in 6.082s stably, and switches to pure hydraulic braking. It can be found that the wheel velocity of the left front wheel is slightly lower than the vehicle velocity, and the vibration caused by the logic threshold algorithm
appears. According to figure 4(b), it can be found that the slip rate of the left-front wheel can be controlled around 0.15 ~ 0.2 to achieve the maximum tire braking force.

Table 1. Parameters used in the Simulation

| Parameter name                                   | value   | Parameter name                                    | value   |
|-------------------------------------------------|---------|---------------------------------------------------|---------|
| Vehicle sprung mass                             | 6000 kg | Master cylinder pressure                          | 14 MPa  |
| Front axle unsprung mass                        | 480 kg  | Time constant of the hydraulic braking system $\tau_h$ | 0.06    |
| Front axle left and right side spin moment of inertia | 10 kg·m² | Delay time of the hydraulic brake control system $\tau_{D1}$ | 0.02    |
| Rear axle unsprung mass                         | 735 kg  | Maximum output torque of the hydraulic brake $T_{h\text{max}}$ | 15000 N·m |
| Front axle left and right side spin moment of inertia | 20 kg·m² | Time constant of the motor $\tau_w$               | 0.025   |
| Distance of mass center from front axle         | 2.9 m   | Delay time of the motor brake $\tau_{D2}$         | 0.02    |
| Distance of mass center from front axle         | 3.55 m  | Rated torque of the drive motor $T_{m\text{max}}$ | ±5000 N·m |
| Four tires effective rolling radius             | 0.51 m  | Time delay coefficient                            | 0.06    |
| Tire spin moment of inertia                     | 14 kg·m²| overload constant $C_L$                           | 1       |

Figure 3. Simulation results of electro-hydraulic composite braking system: (a) braking torque distribution of left front wheel; (b) cut-off frequency of left front wheel

Figure 4. Comparison between composite braking and pure hydraulic braking: (a) vehicle velocity; (b) slip rate of left front wheel
In order to verify the advantages of the electro-hydraulic composite braking strategy compared with traditional ABS, the control effect of pure hydraulic braking system based on logic threshold is compared with that of frequency reconstruction electro-hydraulic composite braking system proposed in this paper. The desired braking torques of both two braking systems are determined by logic threshold control method, and the braking effect is shown in figure 4. If using the time that braking process starts from the 2nd second until the vehicle speed is zero, to calculate the braking time. Then the braking time of electro-hydraulic composite braking is 26.56% less than that of pure hydraulic braking.

5. Conclusion
We propose a novel electro-hydraulic composite braking strategy based on frequency reconstruction for FWID EV. The key idea is to distribute the desired braking torque to hydraulic brake and motor brake according to their characteristics and their advantages over each other. By constructing a delay cost function and an overload cost function, we quantify the delay characteristic of hydraulic brake and low saturation characteristic of motor brake. And then find an optimal cut-off frequency adaptively to allocate the low-frequency part of desired braking torque to the hydraulic braking system and high-frequency part to the motor braking system. In the simulation results, the composite braking system has an advantage of braking time over the pure hydraulic braking system.

The electro-hydraulic braking system proposed can not only promote the braking effect during emergency braking conditions, but also reduce the cost of ABS's calibration for these FWID EVs with another hydraulic braking system installed. The experiments are still needed to verify the practical effect, and the problem of braking torque distribution under regenerative braking conditions will be studied in our future work.

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