

A New Torque Distribution Strategy for Blended Anti-Lock Braking Systems of Electric Vehicles Based on Road Conditions and Driver's Intentions

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ABSTRACT

This paper proposes a new braking torque distribution strategy for electric vehicles equipped with a hybrid hydraulic braking and regenerative braking system. The braking torque distribution strategy is proposed based on the required braking torque and the regenerative braking system's status. To get the required braking torque, a new strategy is designed based on the road conditions and driver's braking intentions. Through the estimated road surface, a robust wheel slip controller is designed to calculate the overall maximum braking torque required for the anti-lock braking system (ABS) under this road condition. Driver's braking intentions are classified as the emergency braking and the normal braking. In the case of emergency braking, the required braking torque is to be equal to the overall maximum braking torque. In the case of normal braking, the command braking torque is proportional to the pedal stroke. Then the required braking torque is chosen as the smaller one between the overall maximum braking torque and the command braking torque. To acquire both high regenerative efficiency and good braking performance when the required braking torque is smaller than the maximum braking torque of the regenerative braking system, the whole braking torque can be provided by the regenerative braking system; when the required braking torque is bigger than the maximum braking torque of the regenerative braking system, the whole braking torque can be provided by both the hydraulic braking system and the regenerative braking system, while the maximum braking torque of the regenerative braking system is obtained based on the battery state of charge (SOC) and vehicle speed in real-time. Both fuzzy rule-based scheduling and robust control approaches will be applied to achieve the above proposed distribution strategy. The effectiveness of the proposed control system is validated by numerical simulations under various road conditions, vehicle speeds, driver's braking intentions, and the battery SOC's.

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INTRODUCTION

As a crucial part of vehicles, the anti-lock braking system (ABS) plays an important role for improving vehicle braking performance in emergency situations. The ABS not only can keep the wheel directional control but also can reduce the stop distance in severe braking. The key part of ABS is the wheel slip ratio control. In order to acquire the maximum friction force and the shortest braking distance, ABS should be able to keep wheel slip ratio track the optimum of slip ratio under given road conditions. In conventional hydraulic braking (HB) system, the most common ABS algorithm is based on the logic threshold control (LTC) through monitoring deceleration and slip ratio of wheels [1]. In order to track the optimal slip ratio, many other algorithms are also used, such as sliding mode control [2], adaptive control, neural control [3], fuzzy control [4] and so on. However, due to the slow dynamic characteristics of the hydraulic system, the friction braking torque has a significant delay during a transient response and it seriously affects the optimal slip ratio control performance [5].

Electric vehicles are currently received much attention due to their zero carbon dioxide emission and other advantages compared to conventional engine vehicles [6]. Electric vehicles can normally be

equipped with regenerative braking (RB) system [7] in addition to the HB system. As HB system can provide large braking torque and RB system has fast response, which is beneficial to the wheel slip ratio control in an emergency braking, appropriate control of these two braking systems not only can enhance the braking performance but also can improve energy regeneration. Therefore, blended braking system that combines conventional HB system and RB system is becoming more attractive and the coordination between energy recovery and vehicle stability through distributing braking torque between the RB system and conventional HB system have received more attention [3]. In [5], the authors conducted a research on braking performance in emergency braking, where the HB system generates a stable pressure which is equal to the required torque, and the RB system is used as an auxiliary system to improve the dynamic characteristics of the braking system. In [1], the energy recovery of a braking system was focused on. For achieving the maximum energy recovery, the HB system is preferred not to operate until the RB system reaches its maximum availability. This strategy can improve the energy recovery performance. However, in emergency situation, the braking performance will be affected. For getting a good braking performance and high energy recovery, it is necessary to take different distribution strategies based on different conditions. For

example, the braking torque distribution strategy can be changed based on different braking modes [8]. Nevertheless, just considering the braking modes is not enough for accurate control. It is necessary to take driver's braking intention, the relationship between pedal displacement and braking force, road condition and vehicle status into consideration.

In this paper, we design a new torque distribution strategy for blended ABS. A three-point prediction method is utilized to estimate the road surface [9] and a robust wheel slip ratio control [5] is used to calculate the overall maximum braking force based on road conditions. The driver's braking intention is identified by a fuzzy system [17]. And the relationship between pedal displacement and braking force is also considered in braking condition [10]. The effectiveness of the proposed control system is validated by numerical simulations under various conditions. The main contribution of the paper is to develop a relationship between pedal commander and braking force by combining driver's braking intention and pedal displacement. And a torque distribution strategy is developed based on driver's braking intention, pedal displacement, road condition, battery state, motor and vehicle status.

The rest of the paper is organized as follows. In Section 2, the models of whole system including quarter-vehicle dynamics, HB system, RB system, the fuzzy system which is used to identify driver's braking intention, relationship between pedal stroke and braking force are introduced. In Section 3, the supervisory slip controller is discussed. It includes a three-point prediction method and a robust wheel slip ratio controller. The torque distribution strategy is discussed in Section 4. In Section 5, the simulation results on a quarter-vehicle braking model are presented. Finally, conclusions are given in Section 6.

SYSTEM MODELING

Vehicle Dynamics Model

In this section, a quarter-vehicle model is used in this paper [11]. This model can be presented by a single wheel. As most ABS control strategies [11] did, only a straight-line braking event is considered in this study and the wheel viscous friction forces are ignored. The dynamics of the vehicle and wheel model can be expressed as:

$$J\dot{\omega}_w = -T_b + rF_x \quad (1)$$

$$m\dot{v} = -F_x - F_a \quad (2)$$

$$F_a = f_a v^2 \quad (3)$$

$$F_x = \mu(\lambda)mg \quad (4)$$

$$\lambda = \frac{\omega_v - \omega_w}{\omega_v} \quad (5)$$

where r is the effective wheel radius, J is the moment of inertia of the wheel, T_b is the braking torque, v is the longitudinal velocity of the vehicle, ω_w is the angular velocity of the wheel, ω_v is the angular velocity of the vehicle, which can be measured by dual-antenna global positioning system [5], F_x is the longitudinal tire force, m is total mass of the quarter vehicle, f_a is the aerodynamic drag coefficient, F_a is the aerodynamic drag force, $\mu(\lambda)$ is the tyre longitudinal friction coefficient, which is a function of wheel slip ratio λ . In this paper, the tire model used is based on the tire friction model developed by Burckhardt and Reimpell (1993). It is shown as below [11]:

$$\mu(\lambda) = C_1(1 - e^{-c_2\lambda}) - C_3\lambda \quad (6)$$

where C_1 , C_2 and C_3 are constants for different road conditions. C_1 is the maximum value of the friction curve, C_2 indicates the friction curve shape and C_3 is the difference between the peak value of the friction curve and the value when slip ratio is 1.

Hydraulic Braking System Model

In this paper, HB system is described by the following equation [11]:

$$dp_w/dt = \frac{k_1 C_d A_{vi} k}{V_0} (p_m - p_w)^{\varphi_i} - \frac{k_2 C_d A_{vd} k}{V_0} (p_w - p_0)^{\varphi_d} \quad (7)$$

where k_1 , k_2 represent the control signals to simulate the on-off feature of the pressure modulation for the wheel cylinder, where the states of HB system can be classified into three modes: increasing, decreasing and holding. p_w , p_m , p_0 are wheel cylinder pressure, master cylinder pressure and reservoir pressure. $C_d A_{vi}$, $A_{vd} k$ and V_0 are the flow coefficient of the valves, the cross-sectional area of the inlet valves, the cross-sectional area of the outlet valves, the bulk modulus of the brake oil and the brake oil volume under no pressure, respectively. φ_i and φ_d are the coefficients for pressure increasing and decreasing modes, respectively, and they can be calibrated from experimental data [2] as following:

$$\frac{dp_w}{dt} = \begin{cases} 37.5416 * (p_m - p_w)^{0.59} & \text{Increasing} \\ 0 & \text{Holding} \\ -38.3128 * (p_w - p_0)^{0.93} & \text{Decreasing} \end{cases} \quad (8)$$

The HB torque can then be calculated using the following equation:

$$T_{hb} = 2P_w A_w f_{hb} r_{hb} \quad (9)$$

where T_{hb} is the hydraulic brake torque, A_w is wheel cylinder cross sectional area, f_{hb} is the friction coefficient, r_{hb} is the effective brake radius.

Regenerative Braking System

In this paper, the RB system consists of in-wheel motors and battery pack. Detailed characteristics descriptions of the RB system can be found in reference [6]. The maximum braking torque of motor can be described by following equation [6]:

$$T_{max} = \begin{cases} T_{mN} & \omega_w < \omega_{mN} \\ 9550P_{mN}/\omega_m & \omega_w > \omega_{mN} \end{cases} \quad (10)$$

where T_{max} is the maximum motor torque, T_{mN} is the motor rated torque, ω_w is the motor speed, ω_{mN} is the motor base speed and P_{mN} is the motor rated power.

Considering battery overcharging may shorten the battery's life when the battery SOC is high, a weighting factor W_{SOC} is employed to protect the battery. So the motor available braking torque is written as follow:

$$T_{avail} = T_{max}W_{SOC} \quad (11)$$

The weighting factor for the battery SOC can be written as follows:

$$W_{SOC} = \begin{cases} 1 & SOC_{act} < 0.8 \\ 1 - SOC_{act} & 0.8 \leq SOC_{act} < 0.9 \\ 0 & SOC_{act} \geq 0.9 \end{cases} \quad (12)$$

where SOC_{act} can be written as follows:

$$SOC_{act} = SOC_{ini} - \frac{100 \int_0^t I(\tau) d\tau}{Q_c} \quad (13)$$

where SOC_{act} is the actual battery SOC value; SOC_{ini} is the initial battery SOC value; Q_c is the total battery energy capacity; I is the motor current:

$$I = \frac{U}{R_{chg}} \quad (14)$$

where U is the battery charging voltage, and it can be written as follows:

$$U = \frac{V_{oc} - \sqrt{V_{oc}^2 + 4P_{motor}R_{chg}}}{2} \quad (15)$$

where V_{oc} is the open-circuit voltage which is the terminal voltage at zero current flowing in or out the battery. R_{chg} is internal resistance value, it depends on the state of the battery. The power into the motor P_{motor} can be calculated as

$$P_{motor} = T_{rb}\omega_w\rho \quad (16)$$

where P_{motor} is the power generated by regenerative braking system; T_{rb} is the actual regenerative braking torque; ρ is the power efficiency.

Pedal Simulator

In the brake-by-wire system, the brake pedal is not directly connected to the actuators. So a pedal simulator is needed to simulate the traditional braking system. In this paper, the model of pedal simulator is described by the following functions [10]:

$$P_m = a_1V^2 + a_2V \quad (17)$$

where V is the volume of brake fluid pushed; a_1, a_2 are fitting factors.

$$V = A_p x \quad (18)$$

where A_p is the piston area; x is the piston displacement. The relationship between piston displacement x and the piston force N can be described as:

$$N = a_1A_p^3x^2 + a_2A_p^2x \quad (19)$$

The relationship between pedal stroke L and piston displacement x can be written as:

$$L = \beta x \quad (20)$$

where L is pedal stroke; β is leverage ratio; L_{max} is the maximum stroke of pedal. Finally, the relationship between pedal stroke L and the master cylinder pressure P_m can be written as:

$$P_m = \frac{a_1A_p^2L^2}{\beta^2} + \frac{a_2A_pL}{\beta} \quad (21)$$

when $L = L_{max}$:

$$P_{mmax} = \frac{a_1A_p^2L_{max}^2}{\beta^2} + \frac{a_2A_pL_{max}}{\beta} \quad (22)$$

$$T_{cb} = T_{hmax} = 2P_{mmax}A_w f_{hb} r_{hb} \quad (23)$$

where T_{hmax} is the maximum braking torque of the HB system. In this paper, T_{hmax} is 400Nm. T_{cb} is the command braking torque. T_{cb} can be calculated by following equation:

$$T_{cb} = 2P_m A_w f_{hb} r_{hb} \quad (24)$$

Driver's Braking Intentions

In this paper, the driver's braking intentions are classified as emergency braking and normal braking. In [17], the ANFIS has been used to identify driver's braking intentions through 1200 sets of data. Referring to [17], we build a TS-fuzzy system to identify driver's braking intentions. The inputs of the fuzzy system are pedal stroke and the change rate of the pedal stroke. Three linguistic values are