

Robust Control of Regenerative and Hydraulic Brakes for Enhancing Directional Stability of an Electric Vehicle During Straight-Line Braking

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ABSTRACT

Thanks to the actuation flexibility of their systems, electric vehicles with individual powertrains, including in-wheel and on-board motors, are a very popular research topic amongst various types of electrified powertrain architectures. The introduction of the individual electric powertrain provides great capacity for improvement of the vehicle's energy efficiency and control performance. However, it also poses tremendous challenges concerning vehicle safety, due to the complex system dynamics and cooperation mechanisms between multiactuators.

For an electric vehicle with independently controlled motors, because of design and manufacturing factors, the steady-state error of each motor output torque, and the flexibilities and nonlinear backlash of left and right drivetrains, can be different. This results in asymmetrical output characteristics of electric powertrain systems on the same axle. Therefore, during a normal straight-line deceleration, an unexpected yaw moment would be generated, affecting vehicle's directional stability.

In this study, a novel method of directional stability enhancement through robust control of blended braking of an electric vehicle equipped with four individual on-board motors during normal straight-line deceleration are studied. System models, including the vehicle dynamics, tire, electric powertrain, and hydraulic brake models, are developed. Mechanisms of directional instability of the electric vehicle during straight-line braking are analyzed. To further improve the electric vehicle's safety and performance, robust control algorithm of blended braking is developed using nonlinear sliding mode approach. Simulations of the proposed control strategy is carried out under straight-line braking. The results demonstrate that the developed approach is advantageous over the baseline, with respect to both the directional stability and regeneration efficiency, thus validating the feasibility and effectiveness of the controller synthesis.

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INTRODUCTION

Increasing global environmental concern requires that automobiles be cleaner and more fuel-efficient. In this context, vehicle electrification is quite promising because of the high-efficiency of powertrain systems and reduced, or zero, emissions [1,2]. Because of the actuation flexibility of their systems, over-actuated electric vehicles with individual driving systems, including in-wheel and on-board motors, are a very popular research topic amongst various types of electrified powertrain architectures [3, 4, 5]. The introduction of the individual electric powertrain provides great capacity for

improvement of the vehicle's energy efficiency and control performance. However, it also poses tremendous challenges concerning vehicle safety, since the system dynamics of the electric powertrain are very different from those of conventional vehicles. Furthermore, cooperation mechanisms between multi-actuators, including motors, brakes, and steering, are quite complex. Therefore, the safety factor for various types of driving conditions is of great significance in the design and control of over-actuated electric vehicles.

Existing studies of over-actuated electrified vehicles with individual in-wheel or on-board motors mainly focus on dynamics control during critical driving maneuvers [3,4,6,7]. However, issues of vehicle stability are of interest not only in emergency driving situations, but also during normal driving maneuvers, because they affect vehicle safety and performance. For example, consider the case of directional instability during normal straight-line braking.

For an electric vehicle with independently controlled motors, because of the design and manufacturing factors, the steady-state error of each motor output torque and the torsional characteristics of left and right drivetrains can be different. This results in asymmetrical output characteristics of electric powertrain systems on the same axle [8,9]. Therefore, during a normal straight-line deceleration, an unexpected yaw moment would be generated, affecting the directional stability of the vehicle. If not corrected by the driver, the vehicle may pull towards one side of the road. This phenomenon is referred to as brake pull. To eliminate the unexpected brake pull, approaches have been investigated through optimization of drivetrain layout and active steering control for conventional ICE vehicles [10]. However, directional stability enhancement during regenerative deceleration for an electric vehicle with individually controlled motors can be even more difficult, because of the complex cooperation mechanisms between multi-actuators, which include motors and hydraulic brakes.

In this study, novel methods of directional stability enhancement through blended braking control of an over-actuated electric vehicle equipped with four individual on-board motors during normal straight-line deceleration are studied. System models, including the electric powertrain, and hydraulic brake models, are developed. Mechanisms of directional instability of the electric vehicle during straight-line braking are analyzed. To further improve the electric vehicle's safety and performance, a robust controller of regenerative and hydraulic blended brakes is developed based on sliding-mode control scheme. Simulations of the proposed control algorithms are carried out during normal straight-line braking processes. Simulation results validate the feasibility and effectiveness of the proposed approach.

BLENDED BRAKING SYSTEM MODELLING

Figure 1 presents the overall structure of the regenerative and hydraulic blended braking system considered in this study. Four onboard electric motors are installed individually on the left and right sides of the two axles of the vehicle. During deceleration, regenerative braking torques, which are transmitted by the drivelines, are exerted onto the axle. In the meantime, hydraulic braking torque of each wheel is modulated by the hydraulic modulator. Blended braking torque governs the overall braking operation.

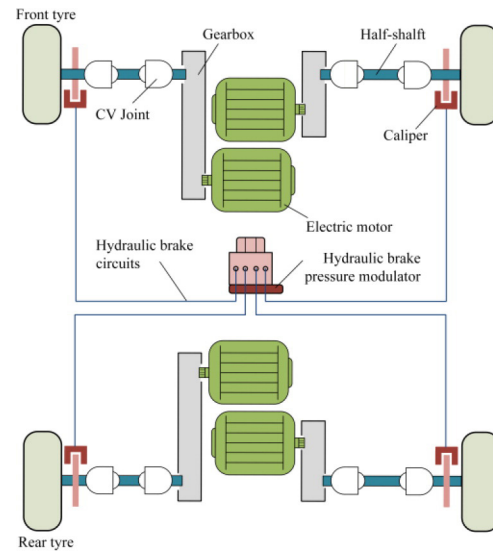


Figure 1. Structure of the regenerative and hydraulic blended braking system.

Electric Powertrain System Model

The electric powertrain is comprised of an electric motor, a gearbox, a final drive, a differential, and half-shafts. In this study, the powertrain system is simplified to a two-inertia model [8,9]. The first inertia is represented by the electric motor, and the other corresponds to the wheel contribution. The drivetrain flexibility, represented by the stiffness and damping properties of the half-shafts, is taken into consideration. Besides, the backlash nonlinearities throughout the powertrain are also considered. And the reference value of the motor torque is the control input to the overall system.

The motor torque is modelled as a first-order reaction with a small time constant τ_m [2], as shown in equation (1).

$$\tau_m \dot{T}_m + T_m = T_{m,ref} \quad (1)$$

where, T_m is the actual value of the motor torque, and $T_{m,ref}$ is the reference value of the motor torque.

The dynamic equation for the transmission of torque from the motor output shaft to the half-shafts is as follows:

$$J_m \ddot{\theta}_m + b_m \dot{\theta}_m = T_m - T_{hs} / i_g \quad (2)$$

where, J_m is the moment of inertia of the electric motor, b_m is the viscous friction of the motor, and $\dot{\theta}_m$ is the angular speed of the electric motor. T_{hs} is the half-shaft torque, and i_g is the transmission ratio.

A flexible half-shaft with nonlinear backlash connects the gearbox and the wheel inertia. The nonlinear model for the half-shaft torque can be written as [9]:

$$T_{hs} = k_{hs}\theta_s + c_{hs}\dot{\theta}_s \quad (3)$$

$$\theta_s = \theta_m / i_g - \theta_w - \theta_b \quad (4)$$

where, k_{hs} and c_{hs} are the stiffness coefficient and damping coefficient of the half-shaft, respectively. θ_s , θ_m , θ_w , and θ_b are the angular position of the shaft, the motor, the wheel, and the backlash, respectively.

The nonlinear model for the backlash position is described as [11]:

$$\dot{\theta}_b = \begin{cases} \max\left(0, \dot{\theta}_d + \frac{k_{hs}}{c_{hs}}(\theta_d - \theta_b)\right), & \theta_b = -\alpha \\ \dot{\theta}_d + \frac{k_{hs}}{c_{hs}}(\theta_d - \theta_b), & |\theta_b| < \alpha \\ \min\left(0, \dot{\theta}_d + \frac{k_{hs}}{c_{hs}}(\theta_d - \theta_b)\right), & \theta_b = \alpha \end{cases} \quad (5)$$

where, 2α is the backlash gap size.

The dynamic equation for a driven wheel is as follows:

$$J_w \ddot{\theta}_w + b_w \dot{\theta}_w = T_l + T_{hb} - T_{hs} \quad (6)$$

where, J_w is the moment of inertia of the tyre, and b_w is the friction coefficient of the tyre. $\dot{\theta}_w$ is the angular speed of the front wheel. T_l is the torque contributed by the longitudinal tyre force, and T_{hb} is the hydraulic brake torque generated by the mechanical brake system.

Hydraulic Brake System Model

In order to simulate the modulation procedure for the wheel cylinder pressures, the hydraulic braking system should be modelled. A schematic diagram of the hydraulic braking system is shown in Figure 2. p_{Ha} and p_{La} denote the pressures of the high-pressure accumulator and the low-pressure accumulator, respectively. The inlet and outlet valves are respectively PWM controlled. The structure of the wheel cylinder is simplified to a piston and spring. k_w is the equivalent stiffness of the spring and x_w is the displacement of the spring.

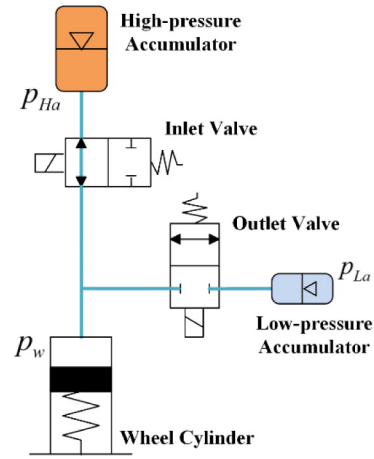


Figure 2. Schematic diagram of the hydraulic braking system.

The hydraulic fluid flow of a valve can be given by:

$$Q = C_d A_v \sqrt{\frac{2 \cdot \Delta p}{\rho}} \quad (7)$$

where, C_d is the flow coefficient of the valve inlet, A_v is the cross-sectional area of the valve opening, and ρ is the density of the hydraulic fluid. Δp is the pressure difference between the inlet and outlet of the valve, and it can be calculated using the following equations:

for the pressure increase process:

$$\Delta p = p_{Ha} - p_w \quad (8)$$

for the pressure decrease process:

$$\Delta p = p_w - p_{La} \quad (9)$$

for the wheel brake cylinder:

$$\pi r_w^2 dx_w = Q dt \quad (10)$$

$$dp_w = \frac{k_w}{\pi^2 r_w^4} dx_w \quad (11)$$

Combining equations (7), (10) and (11), the wheel pressure dynamics can be represented as follows [2]:

$$\dot{p}_w = \frac{k_w}{\pi^2 r_w^4} C_d A_v \sqrt{\frac{2 \cdot \Delta p}{\rho}} \quad (12)$$

where, r_w is the radius of the wheel cylinder.