

Function-Based Architecture Design for Next-Generation Automotive Brake Controls

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ABSTRACT

This paper presents a unified novel function-based brake control architecture, which is designed based on a top-down approach with functional abstraction and modularity. The proposed control architecture includes a commands interpreter module, including a driver commands interpreter to interpret driver intention, and a command integration to integrate the driver intention with sensor-guided active driving command, state observers for estimation of vehicle sideslip, vehicle speed, tire lateral and longitudinal slips, tire-road friction coefficient, etc., a commands integrated control allocation module which aims to generate braking force and yaw moment commands and provide optimal distribution among four wheels without body instability and wheel lock or slip, a low-level control module includes four wheel pressure control modules, each of which regulates wheel pressure by fast and accurate tracking commanded wheel pressure. For electric vehicles, a distribution module considered regenerative braking will also be added into the architecture (low-level control). Then, based on the above novel control architecture and simulation platform, some extensive simulations have been conducted, which demonstrate the validity and effectiveness of the proposed concept and method.

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INTRODUCTION

Brake control system plays a critical role on vehicle safety, stability and handling performance. Conventional brake control systems, such as ABS(Antilock Braking System), TCS(Traction Control System), ESC(Electronic Stability Control), etc., are typically developed independently of each other and are incorporated into the overall brake control system one at a time to achieve their individual objectives. Such bottom-up based control architecture often results in resource and objective conflicts and functional overlap between different features that requires arbitrated control. While arbitration often leads to performance trade-offs or even conflict, such control architecture tends to be more hardware dependent, lack of coordination and sharing of resources, lack of reusability among common functionalities and lack of flexibility for fault tolerance.

Modern brake control system has been widely used not only for handling and stability, but also more and more towards handling driver assistance and active safety features, such as ACC(Adaptive Cruise Control), and Regenerative Braking Systems (RBS) for EVs(Electric Vehicles). As a result of the increasing number and complexity of brake control features, the functional overlap and interaction of each subsystem become inevitable with multiple objectives ranging from enhanced safety and stability, comfort and convenience, to energy saving. Under the conventional control architecture, since each of these features or subsystems may be developed by an individual supplier independently from each other, the integration and coordination among them become more and more complex. The ability to interchange one supplier's subsystem with another becomes problematic, which often causes inevitable duplications.

The conventional brake actuator, such as the Hydraulic Control Unit of ESC, ABS, etc. that difficultly to meet the demands of modern brake control system. In recent years, the brake-bywire system has been rapidly developed, as Electro-Hydraulic-Brake system (EHB), Electro-Mechanical-Brake system (EMB), and the four wheel braking force can be achieved continuously and accurately, which can make up for the deficiency of the traditional system. However, researches based on the novel braking system, the traditional bottom-up control architecture is usually used to achieve the demands of vehicle control[1].

Lots of scholars studied in different control structures, based on the Integrated Vehicle Dynamics Control (IVDC), such as the decentralized control structure[2], the centralized control structure[3], the multi-layered control structure[4], then the integrated control of the various subsystems of vehicle used control allocation can be achieved. Referred to the control architecture of IVDC, in terms of longitudinal control subsystem, B.-C.CHEN, etc. presented a three-level control architecture of electronic stability control (ESC) [5], contains the yaw moment command decision(1st level), the distribution of desired longitudinal tire force based control allocation(2nd level) and the slip ratio control(3rd level). Cheng Lin, etc. presented a two-level control structure of ESC for the four-wheel independently actuated electric ground vehicles[6], contains the yaw moment calculation based on SMC(Sliding Mode Controller) controller(1st level), the wheel torque distribution and control(2nd level), and the desired yaw moment is directly allocated to tires' slip ratio in the two-level. Liang Li, etc. presented a control algorithm based on the hierarchical strategy[7], contains feedback control and

feed forward control. For the path-following and safety problem of the vehicles, Fei Wang, etc. presented a constrained H coordinated control approach[8] and a nonlinear coordinated motion controller[9].

Based on the previous studies and the related issues, this paper presents a unified novel function-based brake control architecture, which is designed based on a top-down approach with functional abstraction and modularity. The proposed control architecture includes a commands interpreter module, state observers for estimation of vehicle sideslip, vehicle speed, tire lateral and longitudinal slips, tire-road friction coefficient, etc., a commands integrated control allocation module, and a low-level control module includes four-wheel pressure control modules. For electric vehicles, a distribution module considered regenerative braking will also be added into the architecture (low-level control). Then, based on the above novel control architecture and a simulation platform, some extensive simulations have been conducted, which demonstrate the validity and effectiveness of the proposed concept and method.

THE NOVEL BRAKE CONTROL ARCHITECTURE

As shown in Figure1, the proposed control architecture includes a commands interpreter module, including a driver commands interpreter to interpret driver intention, and a command integration to integrate the driver intention with sensor-guided active driving command, state observers for estimation of vehicle sideslip, vehicle speed, tire lateral and longitudinal slips, tire-road friction coefficient, etc., a commands integrated control allocation module which aims to generate braking force and yaw moment commands and provide optimal distribution among four wheels without body instability and wheel lock or slip, a low-level control module includes four wheel pressure control modules, each of which regulates wheel pressure by fast and accurate tracking commanded wheel pressure. For electric vehicles, a distribution module considered regenerative braking will also be added into the architecture (low-level control).

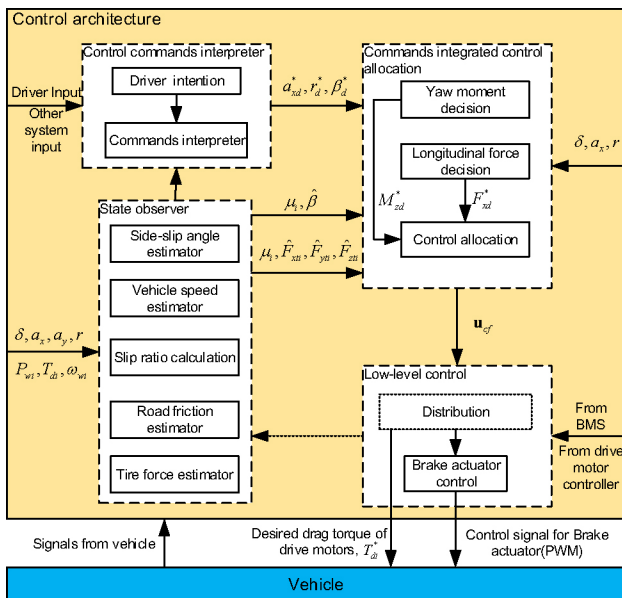


Figure 1. The proposed novel brake control architecture.

Commands Interpreter

Since the control commands are mainly from the driver operation and active control (ADAS(Advanced Driver Assistance Systems), etc.) system, the control commands parsing are a kind of multi- input system control decision-making problems, the main methods generally contain the following ways:

1. Considering the relationship between driver and active control systems, the control commands can be decided with the game theory, and the motion state of vehicle should be obtained according to the ideal model[10].
2. The driver's intention and behavior prediction can be studied based on lots of experimental data, then the semiautomatic smooth transformation of man-machine control mode and the coordination between the driver operation and automatic control can be realized. The ideal response of vehicle can be achieved by automatically adjusting the parameters of the electric control systems, or by building various characteristics of the ideal reference model[11].
3. Dominated by driver input, the control commands from active control systems will be ignored when the driver operations have been detected, the motion state of vehicle can be obtained based on the steering angle, acceleration/braking pedal input, through the ideal reference model.

In order to simplify the problems, the third way is used in this paper.

Longitudinal

The desired deceleration of vehicle is usually proportional to the pedal input in recent research, which is difficult to obtain a balance in both low and high speeds. In this paper, the pedal input and the brake ability determine the desired vehicle deceleration. The desired deceleration can be expressed by:

$$a_{xd} = -f(\hat{F}_{brake}), (\hat{F}_{brake} > F_{ini})$$

$$\hat{F}_{brake} = \frac{P_{mc} A_{mc} + k_s (x_{spring} + x_{ini})}{R_{pedal}}, \quad x_{spring} = P_{brake} / R_{pedal} \quad (1)$$

where, $\hat{F}_{brake}$ is the estimated brake force, P_{mc} , A_{mc} is the pressure and diameter of master cylinder, k_s is stiffness of spring, R_{pedal} is the ratio of brake pedal, and P_{brake} is the travel of brake pedal. The relationship between brake pedal force and brake deceleration can be obtained by field experiment, as shown as Fig 2, and the initial little deceleration is mainly caused by air resistance and rolling resistance of wheels.

The maximum deceleration is largely determined by the vehicle weight, the tire-road adhesion, etc. In addition, the tire-road adhesion is taken into account as well. Then:

$$a_{xd}^* = \min(a_{xd}, \mu g) \cdot \text{sgn}(a_{xd}) \quad (2)$$

where, $\mu = \sum_{i=1}^4 \mu_i / 4$ is the road friction coefficient.

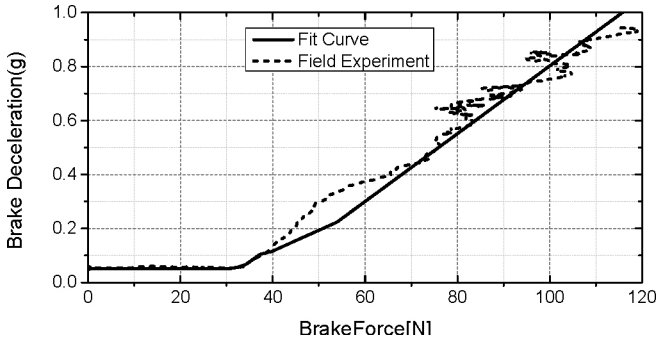


Figure 2. The relationship between the brake pedal force and brake deceleration.

Lateral

A bicycle model with steady-state characteristics is used to represent the desired vehicle lateral responses. The desired yaw rate and sideslip angle at vehicle center of gravity (C.G.) is expressed by:

$$r_d = \frac{v_x / L}{1 + K v_x^2} \delta, \quad (v_x \geq 0) \quad (3)$$

$$\beta_d = \frac{b - \frac{m a v_x^2}{k_r L}}{L(1 + K v_x^2)} \delta, \quad (v_x \geq 0) \quad (4)$$

where, δ is the steering angle at the front tire, m is the vehicle total mass, $L = a + b$ is the wheelbase, k_r is the equivalent cornering stiffness of rear axle, K is the understeer characteristic. As shown in Fig 3, due to the change of the road friction coefficient μ and front wheel angle δ , the correction of the desired yaw rate can be expressed as follows:

$$r_d^* = \min \left\{ |r_d|, \left| \frac{\zeta \mu g}{v_x} \right| \right\} \text{sgn}(\delta) \quad (5)$$

where, ζ is a factor to compensate the error caused by the absence of longitudinal acceleration and the differential of lateral velocity[12]. Similar to the yaw rate, the correction of the desired side-slip angle due to the road friction coefficient and characteristic value of side-slip angle is given as:

$$\beta_d^* = \min \left\{ |\beta_d|, |\beta_{d\max}| \right\} \text{sgn}(\beta_d) \quad (6)$$

$$\beta_d^* = r_d^* \frac{1}{v_x} \left(b - \frac{m a v_x^2}{k_r L} \right), \quad \beta_{d\max}^* = \frac{\mu g}{v_x^2} \left(b - \frac{m a v_x^2}{k_r \cdot L} \right)$$

where, β_T is the characteristic value of side-slip angle, for the ice road is $\pm 2^\circ$ and dried asphalt pavement is $\pm 12^\circ$ [13]. The value of desired side-slip angle can also be defined as zero in some studies. As shown in Fig 4, when the vehicle side-slip angle is large, the yaw moment and lateral force are small, making it difficult to control the vehicle.

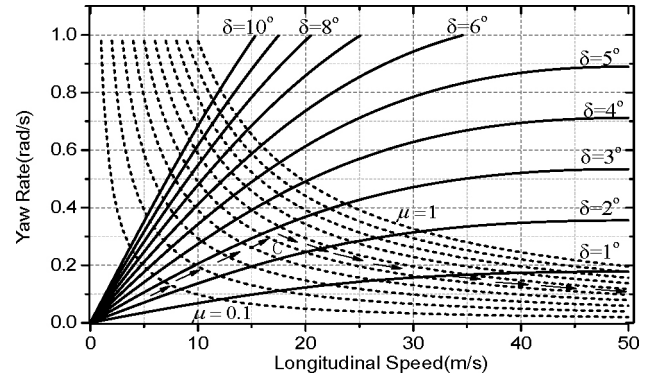


Figure 3. The relationship between the desired yaw rate and velocity in steady-state condition.

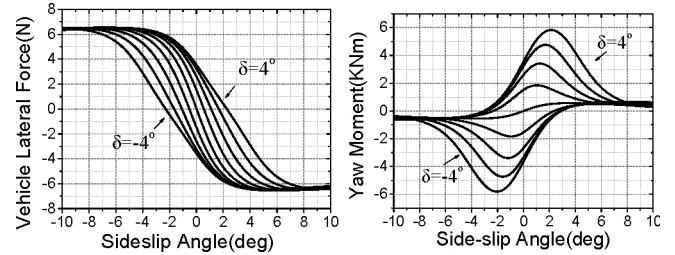


Figure 4. The effects of side-slip angle for vehicle lateral stability.

Commands Integrated Control Allocation

This part mainly contains longitudinal force decision, yaw moment decision and control allocation among the 4 wheels. While the actual road usually has both longitudinal and lateral slope, as shown in Fig 5. As a result of the existence of comprehensive road slope, the component force of gravity in the vehicle coordinate system and the wheel load will be changed. Therefore, the component of gravity in the vehicle coordinate system should be calculated before discussing the commands integrated control allocation.

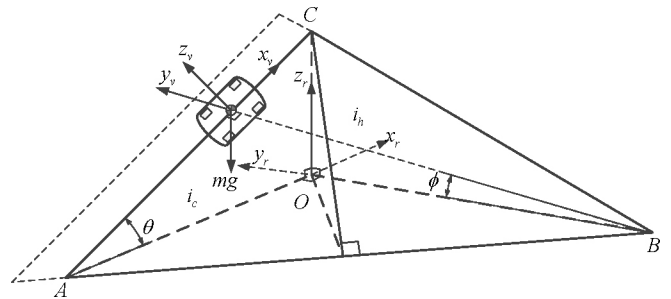


Figure 5. Vehicle on the road with longitudinal and lateral slope

According to the Fig.5, the coordinate system transformation matrix from x_p, y_p, z_p to x_v, y_v, z_v can be expressed as follows.

$$T_{r2v} = Z \cdot Y \cdot X = \begin{bmatrix} \cos \theta & -\sin \theta \sin \phi & -\sin \theta \cos \phi \\ 0 & \cos \phi & -\sin \phi \\ \sin \theta & \cos \theta \sin \phi & \cos \theta \cos \phi \end{bmatrix}$$

$$Z = I, Y = \begin{bmatrix} \cos(-\theta) & 0 & \sin(-\theta) \\ 0 & 1 & 0 \\ -\sin(-\theta) & 0 & \cos(-\theta) \end{bmatrix}, X = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \phi & \sin \phi \\ 0 & \sin \phi & \cos \phi \end{bmatrix} \quad (7)$$