

Article

Actuator-Oriented Hierarchical Coordinated Control of Electromechanical Braking for Corner-Module Electric Vehicles During Braking-in-Turn Maneuvers

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Abstract

Corner-module electric vehicles equipped with four-wheel independent drive, four-wheel independent steering, and electromechanical braking (EMB) actuators provide a flexible platform for software-defined chassis control, but braking-in-turn maneuvers impose severe longitudinal–lateral coupling and competition for tire adhesion resources. This paper proposes an actuator-oriented hierarchical coordinated control strategy for EMB-based corner-module vehicles. At the upper level, a Model Predictive Controller optimizes lateral tire force allocation under a tire-friction-ellipse hard constraint and coordinates the four-wheel steering response. At the lower level, a three-intensity adaptive braking-force distribution algorithm converts the vehicle-level demand into wheel-level EMB clamping-force commands while considering braking intensity, steering intensity, load transfer, and yaw stability. To improve actuator tracking accuracy, the EMB subsystem combines nonlinear actuator modeling, offline parameter identification, online recursive-least-squares correction, and force–speed–position cascade control. MATLAB (R2025b)/Simulink-CarSim co-simulation and EMB hardware-in-the-loop (HIL) tests verify the proposed strategy under fixed-angle emergency braking and lane-change braking conditions with high, low, and variable-adhesion roads. The results show improved trajectory tracking and yaw stability, reduced braking-torque fluctuation, and faster EMB clamping-force response, demonstrating the suitability of the proposed actuator-level coordination method for intelligent electric chassis applications.

Keywords: EMB actuator; corner-module electric vehicle; braking-in-turn maneuver; hierarchical coordinated control; MPC



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1. Introduction

With the evolution of intelligent electric vehicles toward a software-defined chassis, vehicle motion performance increasingly depends on the coordinated control of distributed chassis actuators rather than on a single steering or braking subsystem [1]. As the key path to realizing global chassis control, the distributed drive technology has evolved from traditional in-wheel motor drive to the corner module architecture [2,3]. This kind of physical form that highly integrates the power unit, the steer-by-wire actuator (SBW), and the EMB into an independent unit gives the four 'corners' of the vehicle independent driving, rotating, braking, and suspension capabilities, forming a typical multi-input and

multi-output (MIMO) over-actuated system [4,5]. To fully exploit the over-actuation advantages of distributed-drive vehicles, integrated chassis control strategies and advanced control allocation methods are extensively investigated to optimally coordinate multiple active chassis subsystems [6,7]. Furthermore, brake blending techniques that intelligently coordinate regenerative braking and electro-mechanical friction braking serve as a critical foundation for improving overall vehicle stability and braking efficiency [8]. However, the increase in physical integration also poses unprecedented control challenges, especially under the typical combined conditions of braking-in-turn maneuvers. The complexity of this working condition stems from its significant longitudinal–lateral dynamic strong coupling characteristics, and there is a fundamental competitive contradiction between the longitudinal braking force and the lateral force under the constraint of the ‘friction ellipse’ [9]. The compact design of the corner module often leads to unexpected dynamic interference between the braking and steering systems, which aggravates control difficulty [10,11].

Research on coordinated control of CAV braking-in-turn maneuver can be divided into three areas: longitudinal and lateral decoupling control, dynamic coordinated distribution of braking force, and precise control of the EMB system [12]. The longitudinal and lateral decoupling control is used to address longitudinal and lateral force coupling interference and to adapt to multi-steering modes during braking-in-turn maneuver. The longitudinal and lateral control targets are decoupled using algorithms to improve control accuracy and response speed. Yan et al. proposed a hierarchical control framework for the CAV that distributes the front and rear axle rotation angles via compound control and achieves internal decoupling of the steering system. However, the strong coupling constraints on longitudinal and lateral forces under braking conditions are not fully integrated, and the general decoupling adaptation between 2WS (two-wheel steering) and 4WS (four-wheel steering) modes is not achieved [13]. By leveraging the linear parameter-varying system as the theoretical foundation, Hang et al. developed a dedicated path-tracking controller for 4WS vehicles via a synergistic combination of the LQR algorithm and feedforward control strategies. Partial decoupling is achieved through parameter adaptation, which adapts to different longitudinal speeds and friction coefficients but does not decouple longitudinal and lateral dynamics under braking conditions [14]. Using sliding mode control (SMC) design and hierarchical 4WS path tracking control, Li et al. recast path tracking as a yaw-rate tracking problem and integrated it with lateral stability control to realize lateral internal decoupling, but did not consider the influence of the integration characteristics of the corner module on the decoupling effect [15,16]. Nonlinear, multidimensional coupling is significantly enhanced under extreme conditions. Although existing decoupling strategies have made progress, their simplifications are inadequate for extreme braking-in-turn maneuvers. Specifically, controllers lacking explicit tire-friction-ellipse hard constraints often command physically infeasible tire forces when adhesion is severely limited. Under critical conditions (e.g., emergency obstacle avoidance on slippery roads), these methods cannot prevent premature tire saturation. Consequently, the vehicle is highly prone to losing lateral tracking capability, leading to directional instability, spin-outs, and related traffic accidents. To mitigate these safety risks, the over-actuation advantages of CAVs and their adaptability to variable road adhesion must be fully utilized [17,18].

The focus of the dynamic, coordinated distribution of braking force is to adapt to the corner module’s over-actuation characteristics and balance braking efficiency and steering stability through longitudinal and lateral decoupling. Based on a two-degrees-of-freedom vehicle model, Wu et al. combined the friction-adhesion-ellipse constraint to compute the maximum longitudinal force online and achieve dynamic braking-force distribution. They did not optimize the distribution coefficient in combination with the difference between the front and rear wheel angles in 4WS mode, nor did they fully adapt to dynamic load changes

after decoupling [19]. Zhang Z et al. proposed a coordinated control strategy to optimize the performance of the vehicle braking-in-turn maneuver. Still, they did not introduce a lateral stability correction mechanism, and the decoupled braking force distribution and steering system were not coordinated [20]. The traditional braking force distribution method is based chiefly on ideal I-curve or static constraints, which makes it challenging to address the nonlinear coupling problem of the braking-in-turn maneuver after decoupling, and the over-actuation advantage of the corner module has not been fully realized [21].

The precise control of the EMB system aims to improve the actuator's response speed and control accuracy, and to provide reliable execution guarantees for longitudinal and lateral decoupling control and dynamic distribution of braking force. Mou et al. designed an EMB three-stage closed-loop control strategy, combined with sliding-mode control, to enhance anti-interference performance. Still, they did not fully consider the real-time impact of dynamic load changes on clamping force requirements during the braking-in-turn maneuver, nor did they account for the cooperative control requirements for decoupled steering and braking [22]. Lindvai-Soos et al. designed the EMB controller based on sliding-mode variable-structure control to enhance the system's robustness, but did not incorporate a decoupled coordinated control logic that meets the dynamic requirements of the braking-in-turn maneuver [23].

For corner-module electric vehicles, the key challenge is therefore the application of a real-time motion, force, and acceleration measurement feedback system integrated with the model-informed control of the corner module actuators under braking with variable speed in a turning maneuver. To address this complex task, a coordinated controller must explicitly connect the upper-level lateral-force demand, lower-level braking-force redistribution, and wheel-level clamping-force tracking. Such a connection is essential for explaining how actuator output participates in tire-force coordination and for improving the physical interpretability of braking-in-turn control.

To address these issues, this study proposes an actuator-oriented hierarchical EMB coordinated control strategy for a 4WID-4WIS corner-module electric vehicle. The method combines friction-ellipse constrained Model Predictive Control (MPC), adaptive braking-force distribution, and precise EMB clamping-force control in a unified hierarchical architecture. The upper controller optimizes lateral force and steering allocation, whereas the lower controller redistributes braking demand and converts it into EMB clamping-force commands. The EMB actuator model and cascade controller are incorporated to improve command tracking under dynamic braking loads.

Based on this architecture, the core hypothesis of this study is that the proposed actuator-oriented hierarchical strategy outperforms traditional braking force distribution methods in terms of directional stability and trajectory tracking accuracy under braking-in-turn maneuvers. The main contributions of this paper are as follows:

- To improve the coordination between chassis stability and braking efficiency, a three-intensity adaptive braking-force distribution algorithm is proposed. The algorithm dynamically adjusts front-rear and inner-outer braking weights according to braking intensity, steering intensity, load transfer, and yaw-state deviation, and then maps the coordinated wheel braking demand to the EMB target clamping force.
- To enhance EMB actuator tracking under dynamic loads, a precise clamping-force control method is constructed by combining nonlinear EMB modeling, offline identification, online recursive-least-squares (RLS) correction, and force-speed-position cascade control with sliding-mode speed regulation. This design provides an actuator-level execution basis for the proposed hierarchical braking-in-turn coordination strategy.

The remainder of this paper is organized as follows: Section 2 introduces the 4WID-4WIS CAV structure and the hierarchical coordinated control framework for braking-

in-turn. Section 3 elaborates on system models. Section 4 presents the design of the hierarchical coordinated control strategy. Section 5 conducts experimental verification via MATLAB/Simulink-CarSim co-simulation and EMB HIL tests under multiple conditions, along with result analysis. Finally, Section 6 concludes the research and outlines future work.

2. Research Object and Control Architecture

2.1. 4WID-4WIS CAV Structure and EMB Actuator Structure

The typical CAV is equipped with four-wheel independent drive (4WID), four-wheel independent steering (4WIS), and EMB, and its structure is shown in Figure 1a. The CAV controller unit (VCU) receives steering, driving, brake, battery, and other sensor signals from the vehicle. The VCU and the corner module control unit (MCU) are connected via a wiring harness to enable the exchange of vehicle information and the execution of vehicle algorithms. The MCU acts as the domain controller for the drive, steering, and braking modules, while the brake control unit (BCU) is integrated with each EMB actuator. Figure 1b further illustrates the EMB actuator principle used in this study: the BCU receives the target clamping-force command, drives the PMSM through the inverter, converts motor torque into linear displacement through the reducer/ball-screw mechanism, and generates brake-disk clamping force through the caliper. Current, speed, position, and clamping-force feedback close the force–speed–position cascade loop, providing the actuator-level basis for the coordinated braking strategy developed in the following sections. To support the EMB mathematical model and control design, Table 1 lists the main parameters and output characteristics of the adopted PMSM (model: B20).

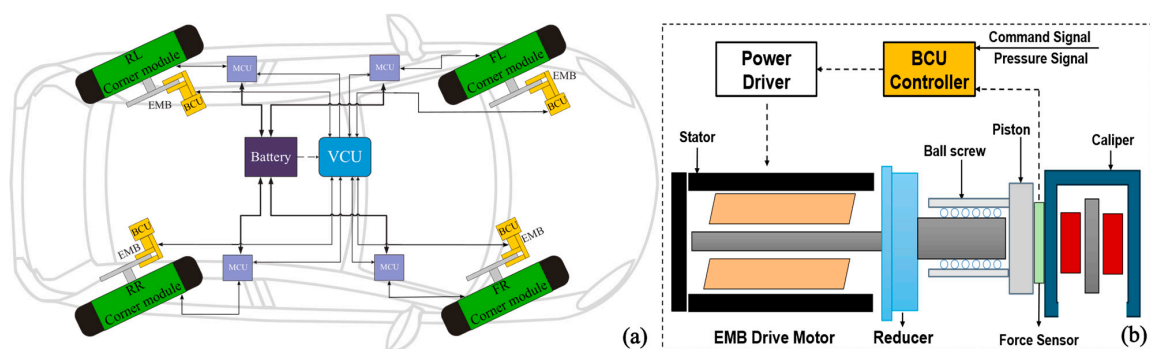


Figure 1. Configuration of the corner-module vehicle and EMB actuator principle: (a) CAV architecture; (b) EMB actuator structure and control signal flow.

Table 1. Key parameters and output characteristics of EMB motor.

Parameters (Unit)	Value	Parameters (Unit)	Value
Rated voltage (V)	24	Rated current (A)	4.1
Voltage Range (V)	12~48	Peak current (A)	8.3
Rated Power (W)	98	Phase resistance (Ω)	0.48
Rated torque (N·m)	1.68	Torque constant (N·m/A)	0.44
Peak torque (N·m)	3.41	Number of pole pairs	7
Phase inductance (mH)	0.31	Maximum clamping force (KN)	10

2.2. EMB Hierarchical Coordination Control Framework for Braking-in-Turn Maneuver

The proposed hierarchical coordinated control system framework for the CAV 4WIS vehicle braking-in-turn maneuver—including the setting of braking-in-turn conditions, four-wheel steering angle allocation, hierarchical coordinated braking force distribution,

and vehicle control—is shown in Figure 2, designed to address the core challenges of unreasonable adhesion resource allocation, under-refined braking force distribution, and steering–braking asynchrony in traditional control via a hierarchical EMB coordinated control strategy integrated with a bidirectional cross-layer cooperative mechanism. In the steering and braking condition setting module, the driver’s peripheral devices, such as steering wheel angle and brake pedal stroke, as well as MATLAB/Simulink simulation signals, can be used as inputs. These inputs provide precise steering and braking conditions for the control algorithm and vehicle dynamics model. These input signals help achieve an effective combination of real driving intentions and the simulation environment. In the rotating/braking hierarchical coordination module, the upper controller considers the difference between longitudinal and lateral speeds, simplifies the vehicle model to a 3DOF single-track nonlinear model, and accounts for the vehicle longitudinal–lateral–lateral coupling dynamics. Based on the tire-adhesion-ellipse constraint, a multi-objective optimization function is constructed. MPC, coupled with the nonlinear vehicle dynamics model, calculates tire lateral force, ensuring steering–braking adhesion and enhancing safety. In combination with the Ackermann Steering Principle (ASP), the composite control strategy is adopted, with yaw-rate feedback used to adjust the rear axle angle to achieve the CAV’s four-wheel angle distribution. The lower controller embeds the dynamic distribution of braking force and the estimation algorithm for sensor-less clamping force, and integrates real-time vehicle information, such as wheel speed and body attitude, to calculate the four-wheel braking clamping force demand. Leveraging EMB’s hydraulic-circuit-free design and fast response, the position–speed–force three-loop control algorithm enables precise braking torque output and closed-loop braking force distribution for braking-in-turn maneuver scenarios.

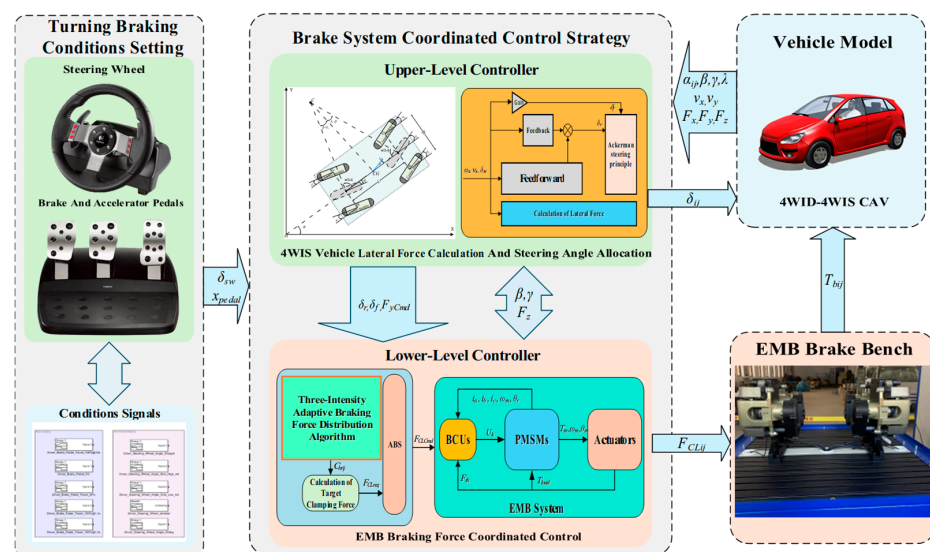


Figure 2. EMB coordination control strategy based on hierarchical architecture.

The hierarchical coordinated control strategy gives full play to the integrated advantages of the ‘drive control rotation’ of the corner module and the independent and accurate control characteristics of EMB. A multi-level closed-loop feedback system spanning vehicle-level (yaw rate, lateral acceleration), corner-module-level (steering angle, wheel speed), and actuator-level (current, position, clamping force) sensors is constructed to compensate for inherent uncertainties, including road adhesion variations, tire friction nonlinearities, and yaw inertia changes caused by active steering. This design effectively shortens the dynamic stable response time, reduces yaw-rate fluctuations, and improves vehicle comfort and steering tracking accuracy.

3. System Modeling

3.1. 4WID-4WIS CAV Model

Given the complexity of real vehicle dynamics, it is necessary to balance model accuracy and computational cost. Therefore, based on the dynamic characteristics of CAV, the influence of rolling, vertical, and pitch motion is ignored, and a 3DOF nonlinear vehicle dynamics model, including longitudinal, lateral, and yaw motion, is established. The force analysis is shown in Figure 3. The simplified model can reflect the longitudinal deceleration, tire vertical load distribution, and the longitudinal and lateral force-coupling effects caused by vehicle braking, providing a dynamic basis for cooperative control of braking-in-turn maneuver. The specific expression is Equation (1).

$$\begin{cases} m\dot{v}_x = mv_y\gamma + 2\left(C_{lf}\lambda_f + C_{lr}\lambda_r + C_{cf}\left(\delta_f - \frac{v_y + a\gamma}{v_x}\right)\delta_f - C_{cr}\left(\delta_r + \frac{b\gamma - v_y}{v_x}\right)\delta_r\right) \\ m\dot{v}_y = -mv_x\gamma + 2\left(C_{cf}\left(\delta_f - \frac{v_y + a\gamma}{v_x}\right) + C_{cr}\left(\delta_r + \frac{b\gamma - v_y}{v_x}\right)\right) \\ I_z\dot{\gamma} = 2\left(aC_{cf}\left(\delta_f - \frac{v_y + a\gamma}{v_x}\right) - bC_{cr}\left(\delta_r + \frac{b\gamma - v_y}{v_x}\right)\right) \end{cases} \quad (1)$$

where m is the curb weight of the vehicle; $\dot{v}_x$ and $\dot{v}_y$ are the longitudinal and lateral accelerations; γ and $\dot{\gamma}$ are the yaw rate and yaw angular acceleration; a and b are the distances from the front and rear axles to the center of mass; C_{lf} and C_{lr} are the longitudinal stiffness of the front and rear wheels; C_{cf} and C_{cr} are the cornering stiffness of the front and rear wheels; λ_f and λ_r are the slip ratios of the front and rear wheels; δ_f is the front axle steering angle; and δ_r is the rear axle steering angle.

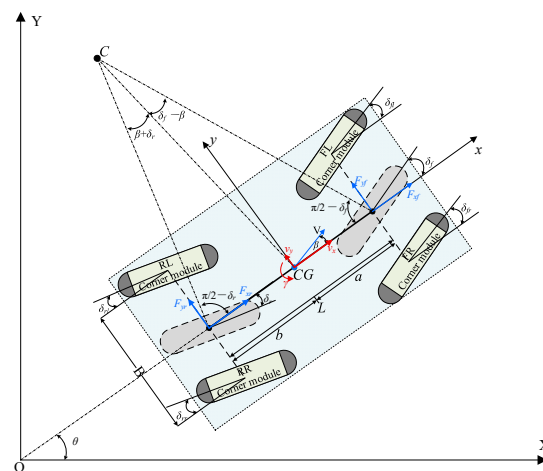


Figure 3. Schematic diagram of integrated 3DOF vehicle dynamics and 4WS distribution of the CAV.

The kinematic model underpins trajectory planning and tracking control. To simplify controller design, the four-wheel independent steering (4WIS) vehicle kinematic model is reduced to a single-track model under the assumptions of pure rolling and small steering angles, as illustrated in Figure 3. The black dots on the front and rear axles denote their respective centers. The blue dot (CG) indicates the vehicle's geometric center, and point C is the vehicle's center of rotation. The heading angle θ is the angle between the vehicle's front direction and the X-axis. The side slip angle of the center of mass β is the angle between the velocity V at the CG and the vehicle's longitudinal axis. According to the ASP, the ideal relationship between the CAV's wheel angles is shown in Figure 3. The steering angle of each wheel is given by Equation (2). Where L is the wheelbase, and B is the track width, $i = f/r$, $j = l/r$.

$$\begin{cases} \tan\delta_{fl} = \frac{\tan\delta_f}{1 - \frac{B}{2L}(\tan\delta_f - \tan\delta_r)} \\ \tan\delta_{rl} = \frac{\tan\delta_r}{1 - \frac{B}{2L}(\tan\delta_f - \tan\delta_r)} \end{cases}, \begin{cases} \tan\delta_{fr} = \frac{\tan\delta_f}{1 + \frac{B}{2L}(\tan\delta_f - \tan\delta_r)} \\ \tan\delta_{rr} = \frac{\tan\delta_r}{1 + \frac{B}{2L}(\tan\delta_f - \tan\delta_r)} \end{cases} \quad (2)$$

3.2. Tire Model

To accurately and concisely describe the tire's mechanical properties, this paper adopts the "Magic Tire Formula" mechanical model, and its calculation formula for the tire's longitudinal force F_x is

$$F_x = D \sin(\text{Carctan}(B_s X_1 - E(B_s X_1 - \arctan(B_s X_1)))) + S_V \quad (3)$$

where X_1 , B_s , C , D , E , S_V are all curve shape parameters of the Magic Tire Formula [24]. To ensure the accuracy of the vehicle dynamics simulation, tires with specifications of 185/65 R15 were referenced from the CarSim database. Under a nominal single-wheel load of 4000 N and dry asphalt road conditions, the empirical parameters are reconstructed using the least-squares method. The specific parameters applied in Equation (3) are $B_s = 9.2$, $C = 1.65$, $D = 4000$ and $E = 0.05$. Furthermore, the normalized friction-ellipse coefficients in Equation (4) are fitted as $C_{xij} = 1.0$ and $C_{yij} = 1.0$.

The longitudinal and lateral sliding speeds of the wheel center are the causes of the longitudinal and lateral forces generated by the tire. Some scholars have shown, through theoretical and experimental analysis, that the tire's longitudinal sliding stiffness is unaffected by lateral forces and tire deformation. In contrast, its lateral stiffness is affected by longitudinal forces and tire deformation [25]. Under the premise of considering the coupling effect of longitudinal and lateral forces of the tire, their simplified relationship is shown in Equation (4):

$$\left(\frac{F_{xij}}{C_{xij}}\right)^2 + \left(\frac{F_{yij}}{C_{yij}}\right)^2 \leq (\mu_{\max} F_{zij})^2 \quad (4)$$

Among them, $F_{xij\max}$ and $F_{yij\max}$ are the maximum longitudinal force and lateral force corresponding to the current tire slip angle α_{ij} of the tire, respectively. C_{xij} and C_{yij} are the friction-ellipse coefficients for the tire's longitudinal force and lateral force, and F_{zij} is the wheel vertical force. $\mu_{\max}$ is the maximum road adhesion coefficient, and $F_{xij\max}$ is the tire vertical load. As shown in Figure 4, in extreme cases, the friction ellipse takes F_{xij} as the semi-major axis and F_{yij} as the semi-minor axis. At any time, the tire force vector $F_{ij} = (F_{xij}, F_{yij})^T$ determined by the actual longitudinal force and actual lateral force of the tire must lie inside the friction ellipse.

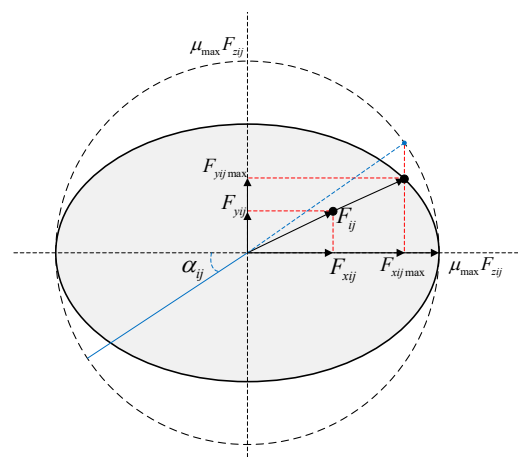


Figure 4. Schematic illustration of tire friction adhesion ellipse.

3.3. EMB Model

The friction characteristics of PMSM directly determine the precision of the braking clamping force. Establishing an accurate and well-calibrated motor friction model is therefore pivotal to optimizing the overall braking performance of the EMB system, with the motor's dynamic governing Equation expressed as Equation (5):

$$\begin{cases} U_a = I_a R_a + L_a \dot{I}_a + K_t \dot{\theta}_m \\ J_m \ddot{\theta}_m = K_t I_a - T_D - T_{load} \end{cases} \quad (5)$$

where U_a , I_a , R_a and L_a represent the armature voltage, current, resistance, and inductance; K_t , θ_m , J_m and T_D represent the torque constant, rotor angle, rotational inertia and friction torque, and T_{load} is the load torque.

To accurately capture EMB nonlinearities and enable dynamic braking command execution, a two-stage parameter identification and estimation method is adopted. First, offline identification is conducted in advance on a dedicated EMB hardware test bench. Using static step-response data spanning the clamping force range of 300–10,000 N, we calibrate the nonlinear characteristic curve between motor rotor angle and clamping force F_{cl} , while identifying key motor parameters. Second, online adaptive estimation operates continuously in real time within the BCU via the RLS algorithm. As specified in Equations (5) and (6), this process directly relies on real-time feedback from the motor's built-in current sensor (I_a) and high-resolution encoder (θ_m) to dynamically estimate and update the actual F_{cl} and braking torque T_{bij} .

$$\begin{cases} F_{cl} = f(x_{emb}^3, x_{emb}^2, x_{emb}), & x_{emb} = \frac{P_h}{2\pi i_p} \theta_m \\ T_{load} = \frac{F_{cl} \cdot P_h}{2\pi \cdot i_p \cdot \eta_s \cdot \eta_p}, & T_{bij} = 2F_{cl} \cdot R_b \cdot \mu_b \end{cases} \quad (6)$$

where x_{emb} is the axial displacement of the ball screw, and P_h , i_p , R_b , μ_b , η_s , η_p are all key parameters of the EMB transmission mechanism.

4. Design of Hierarchical Coordination Control Strategy

4.1. Upper Controller: MPC Lateral Force Optimization with Friction Ellipse Hard Constraint and 4WS Angle Allocation

As shown in Figure 5, the upper controller employs a 'feedforward–feedback' composite control strategy to achieve rapid, stable distribution of front and rear axle steering angles. Feedforward control rooted in the ASP provides the initial steering angle distribution, while yaw-rate feedback adjustment enables dynamic fine-tuning. As the lateral force requirement increases, the rear axle steering angle is corrected in real time to reduce lateral load on the inner wheels, thereby preserving the adhesion margin for longitudinal braking force. Simultaneously, the lateral force target derived via MPC is treated as a hard constraint and transmitted to the lower-layer braking force distribution module, ensuring that the longitudinal and lateral forces do not exceed the limits of the tire friction ellipse.

The controller is based on steering wheel angle input, combined with yaw-rate feedback, to adjust the rear axle angle and ensure vehicle steering stability. The specific distribution relationship, such as the feedforward and feedback coefficients, is derived from the constraint that the center-of-mass sideslip angle is zero in steady-state steering. The expression is Equation (7).

$$\begin{cases} \delta_r = k_1 \delta_f + k_2 \gamma \\ k_1 = -\frac{C_{cf}}{C_{cr}}, k_2 = \frac{mv_x^2 + 2(aC_{cf} - bC_{cr})}{2C_{cr}v_x} \end{cases} \quad (7)$$

where k_1 is a constant less than zero, used to enhance vehicle maneuverability; k_2 is a function of vehicle speed, which can dynamically adjust the rear wheel angle through real-time feedback of yaw rate γ to enhance stability during high-speed braking and steering.

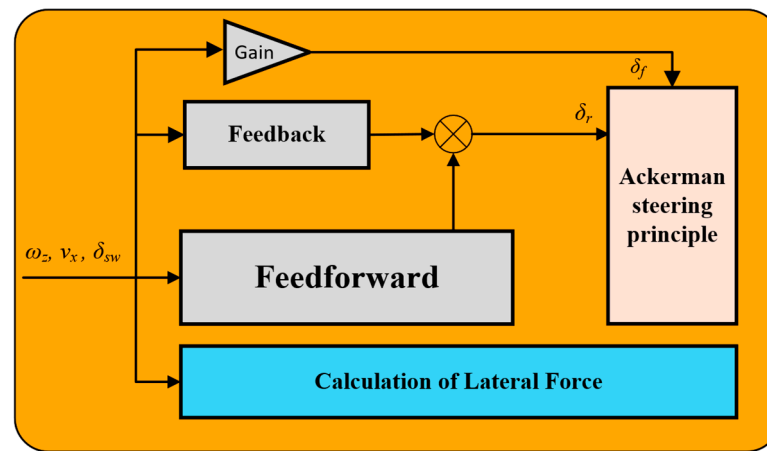


Figure 5. Four-wheel angle distribution and lateral force solution based on ASP.

Combining the lateral force balance relationship of the 3DOF vehicle dynamics model, the calculation expression for the lateral forces of the front and rear wheels is given by Equation (8), where F_{yf} and F_{yr} represent the lateral forces of the front and rear wheels, respectively. The total required lateral force F_{yCmd} for steering is the sum of the lateral forces of the front and rear wheels. By inputting the real-time vehicle states ($v_x/v_y/\gamma$) and steering wheel angle δ_{sw} into the above Equation, and combining the tire model in Section 3, a multi-objective optimization function is constructed, and the MPC method is used to solve the lateral tire forces [26].

$$\begin{cases} F_{yf} = C_{cf} \left(\delta_f - \frac{v_y + a\gamma}{v_x} \right) \\ F_{yr} = C_{cr} \left(\delta_r + \frac{b\gamma - v_y}{v_x} \right) \end{cases} \quad (8)$$

Model Predictive Control (MPC) is adopted for lateral force allocation, as it can solve multi-objective optimization problems—tracking the desired yaw rate while minimizing the sideslip angle—subject to hard physical constraints such as tire-friction-ellipse boundaries.

To balance real-time performance and computational efficiency, the nonlinear vehicle dynamics model is locally linearized via successive linearization, forming a discrete-time linear time-varying (LTV) predictive model, $\mathbf{x}(k+1) = \mathbf{A}_k \mathbf{x}(k) + \mathbf{B}_k \mathbf{u}(k)$, where the state vector is $\mathbf{x} = [\beta, \gamma]^T$, and the control input is $\mathbf{u} = [F_{yf}, F_{yr}]^T$. The system matrix $\mathbf{A}_k$ and input matrix $\mathbf{B}_k$ are updated at each control cycle using real-time longitudinal speed v_x and tire cornering stiffness, adapting to transient braking dynamics.

The optimal lateral forces are derived by minimizing the cost function:

$$J = \sum_{i=1}^{N_p} \|\mathbf{y}(k+i|k) - \mathbf{y}_{ref}(k+i|k)\|_{\mathbf{Q}}^2 + \sum_{i=0}^{N_c-1} \|\Delta \mathbf{u}(k+i|k)\|_{\mathbf{R}}^2 + \rho \epsilon^2 \quad (9)$$

where $\mathbf{y}_{ref} = [0, \gamma_d]^T$ denotes the reference state; $N_p = 30$ (prediction horizon) and $N_c = 3$ (control horizon); $\mathbf{Q}$ and $\mathbf{R}$ are weighting matrices for tracking errors and control increments; ϵ is the relaxation factor weighted by ρ . The optimization is constrained by $|F_{yi}| \leq \mu F_{zi}$, derived from the tire friction ellipse.

4.2. Lower Controller: Three-Intensity Adaptive Braking Force Distribution and EMB Precise Control

4.2.1. Adaptive Braking Force Distribution

Under braking-in-turn maneuver conditions, the vehicle's longitudinal and lateral force coupling is significant, and load transfer is obvious. The lower-level controller combines the ideal braking force distribution curve and the ECE regulations constraint to propose an adaptive braking force distribution algorithm. The algorithm, based on the tire-adhesion-ellipse limit, introduces a lateral stability correction function, constructs a hierarchical adaptation function, and addresses the insufficient adaptability of the traditional distribution method during braking-in-turn maneuvers. The solution of the dynamic distribution coefficient C_{brij} is shown in Equation (10).

$$C_{brij} = \frac{\mu F_{zij} \sqrt{1 - \left(\frac{F_{yij}}{\mu F_{zij}} \right)^2} \cdot \psi(z, \sigma) \cdot \xi(\beta, \gamma)}{\sum_{i,j} \left[\mu F_{zij} \sqrt{1 - \left(\frac{F_{yij}}{\mu F_{zij}} \right)^2} \cdot \psi(z, \sigma) \cdot \xi(\beta, \gamma) \right]} \quad (10)$$

$$\psi(z, \sigma) = \begin{cases} 1.0 & (z \leq 0.1 \& \sigma \leq 0.3) \\ 1.2 - k_{z1}z + k_{\sigma 1}\sigma & (0.1 < z \leq 0.61 | 0.3 < \sigma \leq 0.7) \\ 0.8 + k_{z2}z + k_{\sigma 2}\sigma & (z > 0.61 \& \sigma > 0.7) \end{cases}$$

$$\xi(\beta, \gamma) = 1 - k_{\beta} \cdot \frac{|\beta|}{\beta_{\max}} - k_{\gamma} \cdot \frac{|\gamma - \gamma_d|}{\gamma_d}$$

$$\sigma = \frac{|\delta_f| + |\gamma - \gamma_d|}{|\delta_{f\max}| + |\gamma_{\max} - \gamma_d|}$$

Among them, $\psi(z, \sigma)$ is the hierarchical adaptation function. Combined with ECE regulations, it implements three-level dynamic adaptation (weak, medium, and strong) based on braking intensity and steering intensity. Under weak working conditions, the distribution weight is balanced to avoid excessive braking force affecting steering stability; under medium working conditions, it balances braking efficiency and trajectory-tracking requirements; under strong working conditions, it prioritizes lateral stability. In the formula, k_{z1} , k_{z2} , $k_{\sigma 1}$ and $k_{\sigma 2}$ are the braking intensity correction coefficients and steering intensity correction coefficients, respectively. The lateral stability correction function addresses the nonlinear dynamics of turning and braking. When the side slip angle of the center of mass approaches the limit value or the yaw rate deviates significantly from the desired value, it reduces the braking force weight of the corresponding wheels to prevent tire instability; when the vehicle's lateral state is stable, it maximizes the braking force weight to ensure braking efficiency. In the formula, k_{β} and k_{γ} are the side slip angle correction coefficient and yaw-rate correction coefficient, respectively. σ quantifies the intensity of the steering condition through the change in steering angle and the deviation of steering stability, where $\delta_{f\max}$ is the maximum front wheel steering angle, $\gamma_{\max}$ is the maximum yaw rate, and γ_d is the desired yaw rate. The empirical coefficients (k_{z1} , k_{z2} , $k_{\sigma 1}$, $k_{\sigma 2}$, k_{β} , k_{γ}) and brake intensity thresholds in Equation (10) are calibrated via extensive co-simulations.

As shown in Figure 6, the lower-layer control strategy takes the vehicle's lateral acceleration a_y , δ_f , δ_r , longitudinal vehicle speed v_x , vertical force F_z , slip ratio λ_{ij} , target lateral force F_{ycmd} , and γ as inputs. Via the hierarchical adaptive braking force distribution algorithm, it outputs C_{brij} to the target clamping force calculation sub-module. Combined with EMB parameters such as brake disk radius and friction coefficient, the target clamping force F_{CICmd} is solved. Considering the variation in road adhesion during braking, the

lower-layer controller embeds an SMC-based ABS module, which calibrates the sliding surface and reaching law through k_{abs1} , k_{abs2} and k_{abs3} to achieve tracking control of the wheel slip ratio. This module receives target clamping force and wheel speed feedback, corrects the clamping force in real time, and dynamically adapts to the lateral force F_{yCmd} while ensuring braking efficiency and avoiding excessive occupation of adhesion resources by the longitudinal braking force.

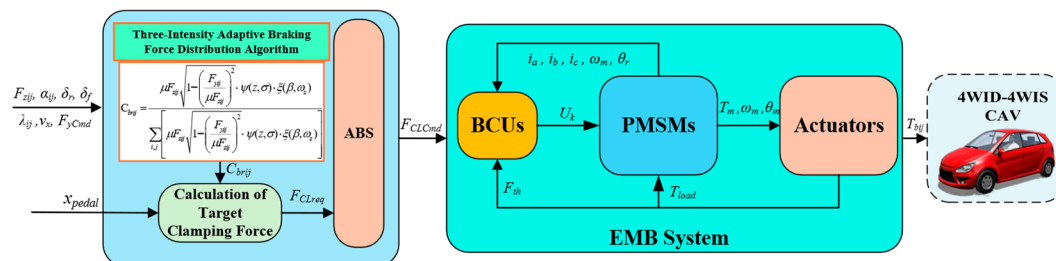


Figure 6. Logic diagram of lower controller for braking distribution coordination control.

4.2.2. EMB Precise Control

The corrected clamping force demand is transmitted to the EMB system, and the BCU initiates the position–speed–force three-loop control algorithm upon receiving the signals [27]. Key parameters of the EMB coordinated control strategy are listed in Table 2, where J_{emb} is the equivalent moment of inertia including the reduction mechanism, and the three-loop control parameters are denoted by k with subscripts.

Table 2. Key parameters of EMB coordination control strategy.

Symbol	Value	Symbol	Value	Symbol	Value
k_β	0.1	k_{z2}	0.3	k_{pFcl}	1365.6
k_ω	0.08	$k_{\sigma1}$	0.3	k_{iFcl}	0.12
k_{z1}	0.7	$k_{\sigma2}$	0.7	k_{dFcl}	0.06
Symbol	Value	Symbol	Value	Symbol	Value
$k_{c\omega}$	3.5	$k_{p\theta}$	0.8	k_{abs1}	8.2
$k_{i\omega}$	0.8	$k_{i\theta}$	0.05	k_{abs2}	9.1
$k_{k\omega}$	3.2	J_{emb}	0.001 kg·m ²	k_{abs3}	0.13

The force loop is the outer loop: A PID controller is employed here for clamping force regulation due to its high reliability, low computational overhead, and excellent capability to eliminate steady-state errors for low-frequency target force tracking. It takes the deviation between the target clamping force and the sensor-less estimated clamping force as input and outputs the target motor speed ω_{ref} . This estimated value is based on signals such as motor current $i_a/i_b/i_c$ and speed ω_r , synchronously compensating for nonlinear friction disturbances [28].

The speed loop (middle loop) serves as the core dynamic regulator. An integral sliding mode controller (ISM) is adopted here due to its strong robustness to matched uncertainties, such as nonlinear friction drift and dynamic load transfer under cornering-braking conditions. Defining the motor speed tracking error as $e_\omega = \omega_{ref} - \omega_r$, the sliding surface s is formulated as $s = e_\omega + c \int e_\omega dt$, where c is the sliding surface parameter. Combined with an exponential reaching law, the SMC dynamically adjusts the target current to suppress speed fluctuations and reject external disturbances.

The position loop is the inner loop; it is based on the target current from the speed loop and the rotor position signal fed back by the encoder, and it generates the driving

voltage U_k via SVPWM to drive the motor to output torque. Combined with the feedback compensation of the load torque T_{load} , the motor torque is converted into braking clamping force through the ball screw, ultimately realizing precise EMB control by generating the four-wheel braking torque T_{bij} .

5. Simulation Analysis and Experimental Verification

5.1. Co-Simulation Platform Settings

Based on the proposed mathematical model, this paper establishes a MATLAB/Simulink and CarSim co-simulation platform for algorithm verification and debugging, as shown in Figure 7.

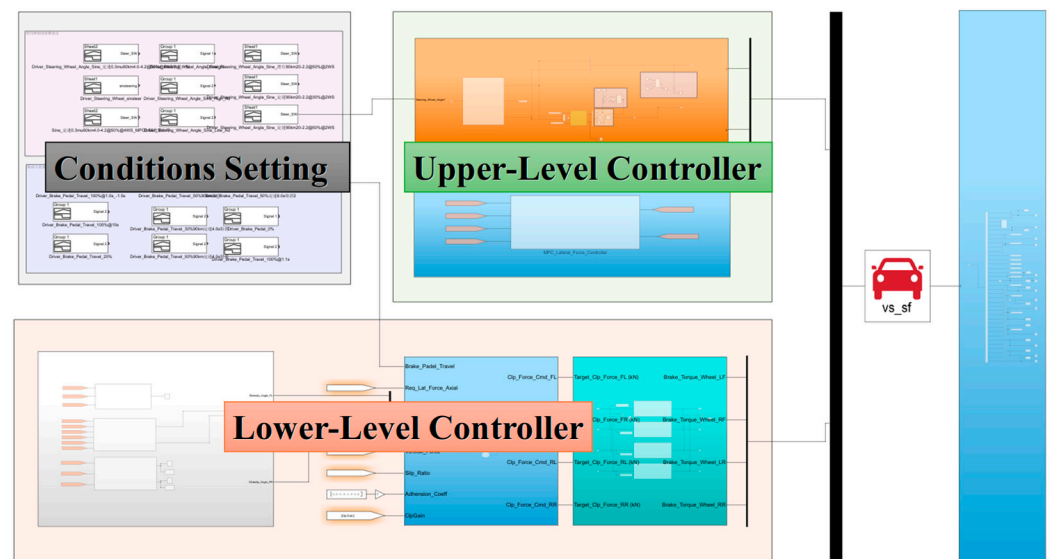


Figure 7. Co-simulation platform of coordinated control strategy.

The co-simulation platform follows the overall logic of the control strategy framework, divided into four parts: working condition setting, upper-level controller, lower-level controller, and CarSim vehicle state. This paper selects a distributed-drive B-class vehicle from the CarSim vehicle library for simulation, with the steering and braking systems set as external inputs. The key parameters of the whole vehicle dynamic model are shown in Table 3.

Table 3. Vehicle simulation model parameters.

Symbol	Value	Unit	Symbol	Value	Unit
m	1236	kg	L	2600	mm
m_s	1110	kg	r_b	287	mm
h_{CG}	540	mm	F_z	5500	N
B	1563	mm	I_z	1343.1	kg·m ²
a	1040	mm	I_x	440.6	kg·m ²
b	1560	mm	I_w	1.3	kg·m ²

Among them, m_s refers to the sprung mass, h_{CG} is the height of the center of mass, r_b denotes the tire rolling radius, I_x and I_z represent the moments of inertia about the x-axis and z-axis, and I_w is the longitudinal moment of inertia of the tire. The meanings of other symbols are consistent with those mentioned earlier.

5.2. Co-Simulation Settings and Result Analysis

5.2.1. Setting and Result Analysis of Fixed Angle Emergency Braking Scenarios

Simulation of emergency braking under a steering angle emergency braking condition, where the steering wheel angle increases from 0 to 60° within 500 ms, and the brake pedal

is pressed to 100% within 300 ms, as shown in Figure 8a. The setup of the variable adhesion road surface is shown in Figure 8b, with the road adhesion coefficient suddenly dropping from 0.85 to 0.3 at a longitudinal distance of 40 m, simulating the high-rise ice surface scenario on actual roads.

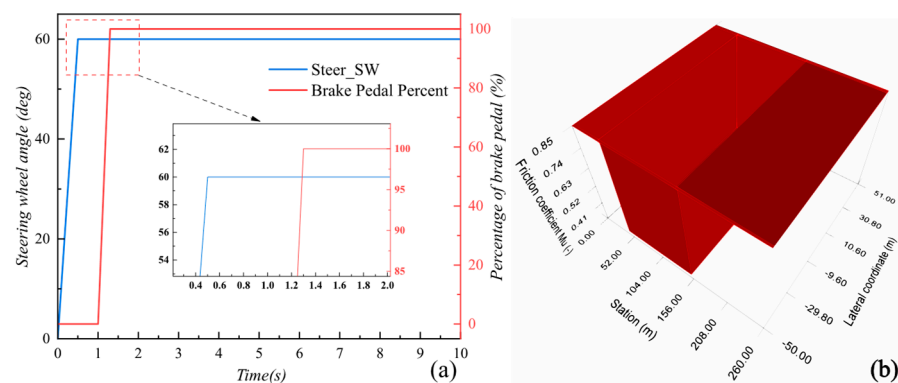


Figure 8. (a) Steering wheel and brake pedal settings in the emergency braking scenario with a fixed steering angle; (b) emergency braking scenario with variable adhesion road surface settings.

This article presents three typical working conditions for fixed-angle emergency braking scenarios. The parameters are shown in Table 4, and the three working conditions test the two-wheel steering (2WS) and 4WS conditions, respectively, while covering the corner braking conditions on high-adhesion, low-adhesion, and variable-adhesion road surfaces. Such a setting can comprehensively examine the robustness of the coordinated control strategy. In the simulation results of the lateral force and torque curves, the black and red solid lines represent the left and right front wheels, respectively, and the blue and green dashed lines represent the left and right rear wheels, respectively. In the yaw-rate and longitudinal acceleration curves, the red solid line represents the proposed MPC-AD strategy, while the blue and green dashed lines represent Baseline 1 (MPC-I) and Baseline 2 (LQR-AD), respectively. To rigorously evaluate the performance of the proposed strategy, two state-of-the-art control schemes are adopted as benchmarks. (1) Baseline 1 (MPC-I): an MPC-based upper controller integrated with the traditional ideal I-curve braking force distribution. (2) Baseline 2 (LQR-Adaptive): a standard LQR-based upper controller combined with the proposed adaptive distribution strategy. To guarantee a fair comparison, the MPC and LQR frameworks share identical settings, including vehicle parameters, reference models and test scenarios.

Table 4. Simulation working condition parameters for fixed angle emergency braking scenarios.

No.	Brake Pedal (%)	δ_{sw}	V (km/h)	μ
1	0–100@0.3 s	60	90	0.85
2	0–100@0.3 s	60	60	0.3
3	0–100@0.3 s	60	60	0.85 to 0.3

Figure 9a depicts the vehicle's lateral force variation curve, where front-wheel lateral force exceeds rear-wheel lateral force under both steering conditions, and right-wheel lateral force is generally larger than left-wheel lateral force, consistent with the vehicle's load transfer characteristics during left braking-in-turn. During the 2–6 s interval, the lateral force of the vehicle under 4WS is more stable than under 2WS, indicating that the steering-intensity term of the braking-force distribution algorithm helps stabilize steering.

Figure 9c presents the yaw-rate responses under three control strategies. The proposed MPC-Adaptive strategy exhibits smaller fluctuations than Baseline 1 (MPC-I) and Baseline 2 (LQR-Adaptive), indicating better lateral stability during braking-in-turn maneuvers.

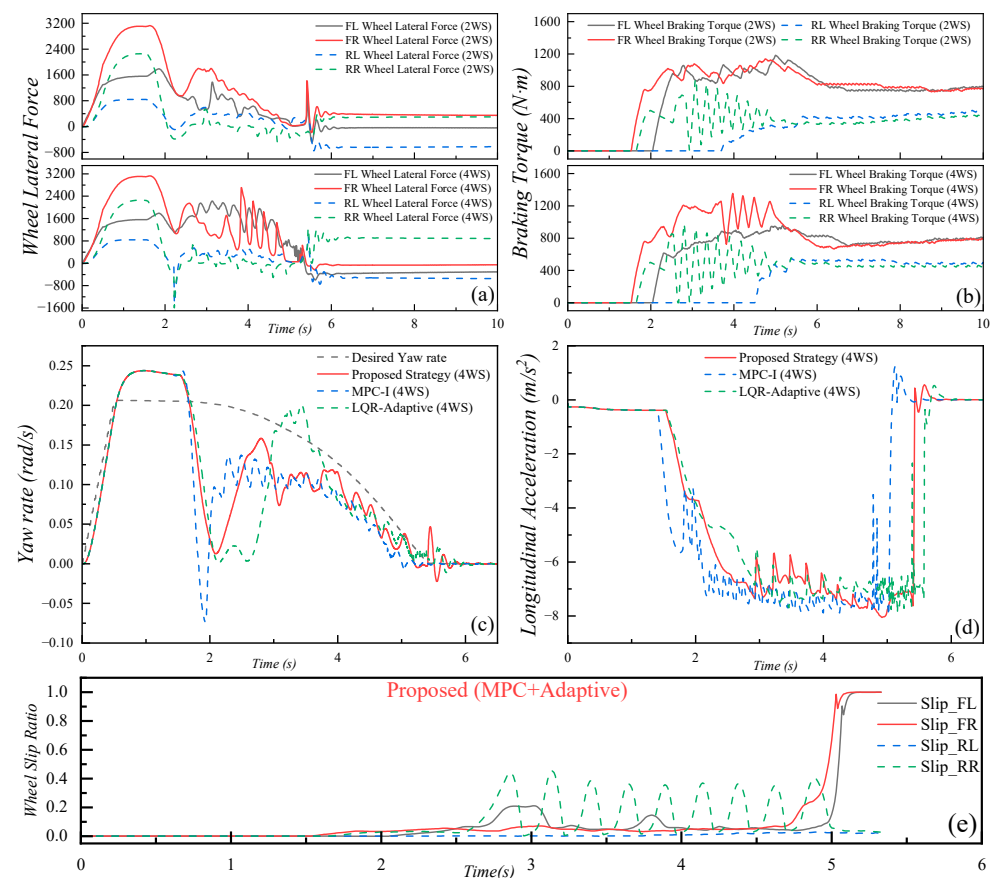


Figure 9. Curves of emergency braking under 90 km/h high adhesion road surface and fixed steering angle; (a) wheel lateral force curves; (b) wheel braking torque curves; (c) yaw-rate curves; (d) longitudinal acceleration curves; (e) four-wheel longitudinal slip ratio curves of the proposed MPC-Adaptive strategy.

Figure 9b is the vehicle braking torque curve; the right wheel, due to load transfer, triggers the ABS function, and oscillating changes occur. The dynamic braking force distribution strategy is effective for both 2WS and 4WS conditions, with the EMB torque response delay of approximately 55 ms, verifying EMB's precise control. Specifically, the mean difference in braking torque between the front and rear axles under 4WS is approximately 19.7% lower than that under 2WS, which can effectively suppress braking pitching; Figure 9d illustrates the longitudinal deceleration curves of the three control methods. All strategies maintain stable braking, while the proposed strategy achieves smoother deceleration and better driving comfort. Furthermore, Figure 9e illustrates the four-wheel longitudinal slip ratios of the proposed MPC-AD strategy. Benefiting from the high-adhesion road surface, the slip ratios are strictly confined within the optimal adhesion region (with the front axle mean slip ratio at approximately 0.098). This demonstrates that the coordinated controller effectively utilizes the longitudinal adhesion without risking wheel lockup, thereby reserving sufficient lateral margin for steering.

The simulation results for the 60 km/h low-adhesion road condition are shown in Figure 10; the performance of each sub-fig. of the low-adhesion road is consistent with the overall law of the high-adhesion road. From Figure 10a,b, it can be seen that the coordinated control strategy has a stable control effect under the low-adhesion road, the

oscillation amplitude of the 4WS condition is smaller compared to the 2WS condition, and the difference in the mean of the lateral forces and braking torque between the front and rear axles is smaller. Figure 10c,d indicate that the proposed MPC-AD strategy achieves better lateral stability and longitudinal deceleration performance than Baseline 1 (MPC-I) and Baseline 2 (LQR-AD), further verifying the robustness of the coordinated control algorithm. As shown in Figure 10e, under low-adhesion conditions ($\mu = 0.3$), tire forces are highly prone to saturation. With the proposed lower-layer anti-lock braking (ABS) module, the excessive wheel slip tendency is rapidly suppressed. The mean slip ratios of the front and rear axles are stabilized at approximately 0.362 and 0.266, respectively, effectively preventing complete wheel lockup while fully utilizing the available braking force under low-adhesion limits.

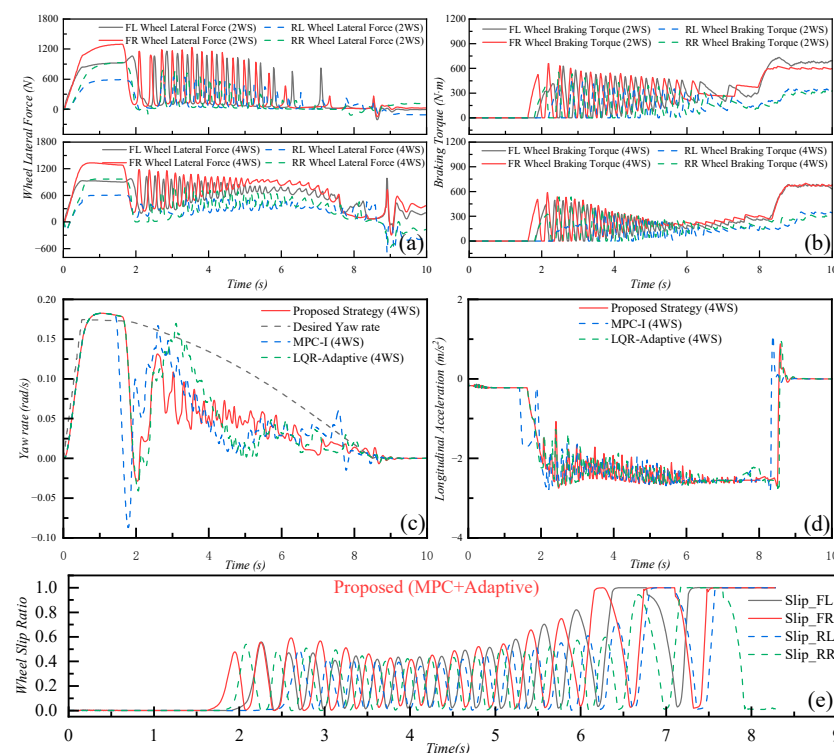


Figure 10. Curves of emergency braking under 60 km/h low-adhesion road surface, fixed steering angle; (a) wheel lateral force curves; (b) wheel braking torque curves; (c) yaw-rate curves; (d) longitudinal acceleration curves; (e) four-wheel longitudinal slip ratio curves of the proposed MPC-Adaptive strategy.

The simulation results for the 60 km/h variable adhesion road condition are shown in Figure 11. The vehicle transitions from a high-adhesion road surface (coefficient of 0.85) to a low-adhesion road surface (coefficient of 0.3). From Figure 11b,d, the lateral wheel force and braking torque respond rapidly. With coordinated regulation between the inner and outer wheels, vehicle deceleration drops obviously after 2.4 s and maintains an average value of approximately 2.4 m/s^2 , which satisfies the emergency braking requirements on low-adhesion roads. Figure 11c shows that the yaw rate of all three control strategies stays within a reasonable range. Compared with Baseline 1 (MPC-I) and Baseline 2 (LQR-AD), the proposed MPC-AD strategy achieves better steering stability, which fully reflects the advantages of the presented adaptive braking force distribution algorithm over conventional control methods. Figure 11e validates the dynamic adaptive capability of the slip ratio control. When the vehicle abruptly transitions onto the low-adhesion road patch at $t = 2.4 \text{ s}$, the wheel slip ratios initially surge but are immediately restrained by the active

anti-lock intervention. This rapid slip regulation prevents tire instability during extreme road variations.

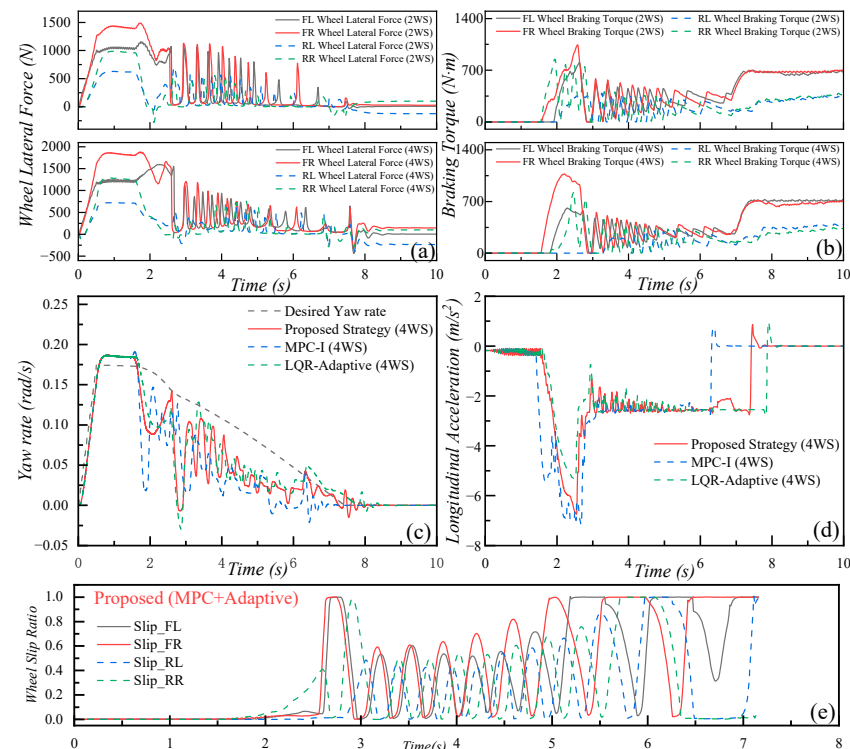


Figure 11. Curves of emergency braking under the condition of 60 km/h variable adhesion road surface and fixed steering angle; (a) wheel lateral force curves; (b) wheel braking torque curves; (c) yaw-rate curves; (d) longitudinal acceleration curves; (e) four-wheel longitudinal slip ratio curves of the proposed MPC-Adaptive strategy.

5.2.2. Setting and Result Analysis of Lane Change Braking Scenarios

The lane-change braking scenarios define three typical working conditions, as shown in Table 5. The adhesion coefficients of Conditions No. 1 and No. 2 are uniform road surfaces, while Condition No. 3 has variable adhesion on the road surface. Taking Condition No. 3 as an example, the steering wheel angle and brake pedal percentage are shown in Figure 12a and used to simulate the driver's braking during the lane change. After completing the lane change, the brake pedal returns to 0%, and the vehicle continues to travel at the original speed; the variable adhesion road surface is set as shown in Figure 12b, with 70–100 m being the lane change section, where the road adhesion coefficient abruptly changes from 0.85 to 0.3 starting from 85 m.

Table 5. Lane change braking scenarios simulation working condition parameters.

No.	Brake Pedal (%)	δ_{sw}	V (km/h)	μ
4	0–50@0.2 s	SLC	90	0.85
5	0–30@0.2 s	SLC	60	0.3
6	0–30@0.2 s	SLC	60	0.85 to 0.3

Considering the length of the paper and that variable road adhesion conditions can fully represent emergency braking during cornering. Therefore, the discussion and analysis focus on the No. 3 scenario of lane-change braking on a variable-adhesion road surface at 60 km/h, and the simulation results are shown in Figure 13.

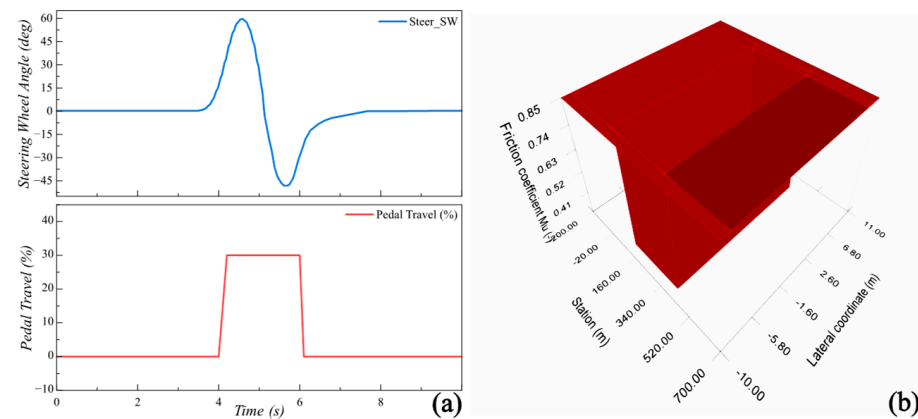


Figure 12. (a) The 60 km/h variable adhesion road surface lane change braking steering wheel and brake pedal settings; (b) 60 km/h variable adhesion road surface lane change braking road settings.

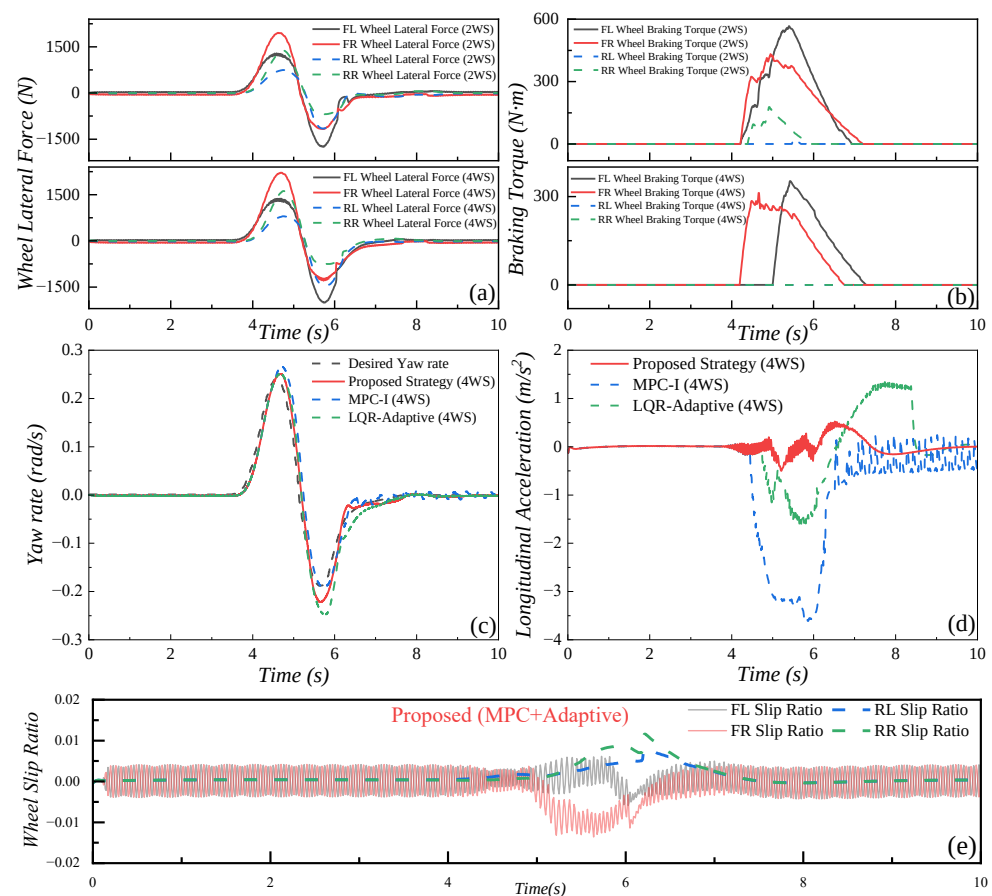


Figure 13. The 60 km/h variable adhesion road braking-in-turn maneuver condition curves; (a) wheel lateral force curves; (b) wheel braking torque curves; (c) yaw-rate curves; (d) longitudinal acceleration curves; (e) four-wheel longitudinal slip ratio curves of the proposed MPC-Adaptive strategy.

The steering and braking strength in the variable-adhesion lane-change braking condition are lower. Figure 13a shows that during the vehicle's single-lane change, the outer wheel experiences a significant lateral force when steering and braking simultaneously. In contrast, lateral force varies smoothly under both 2WS and 4WS conditions, validating the effectiveness of the dynamic distribution strategy. As observed from the 2WS condition curves in Figure 13b, the right rear wheel produces braking torque at large left steering amplitudes, indicating that the hierarchical adaptive function balances braking efficiency and trajectory-tracking demands under moderate operating conditions. The average brak-

ing torque of the proposed MPC-AD strategy is notably lower than that of Baseline 1 (MPC-I) and Baseline 2 (LQR-AD), which is consistent with the trend of the longitudinal acceleration curve in Figure 13d. Combined with the yaw-rate curve in Figure 13c, the results demonstrate that the hierarchical coordinated control strategy delivers better lateral–longitudinal cooperative performance during lane-change braking on variable-adhesion roads. Figure 13e presents the wheel slip ratios during the lane-change braking maneuver. With moderate braking intensity (0–30% brake pedal travel), the longitudinal slip ratios remain extremely low, with mean values below 0.005. The hierarchical controller intentionally restricts longitudinal adhesion utilization to reserve most of the friction-ellipse capacity for the large lateral forces required for obstacle avoidance lane changes, ensuring excellent directional stability and tracking safety.

Table 6 summarizes the comprehensive statistical metrics across all test scenarios, covering both trajectory tracking accuracy and braking stability. Notably, to characterize the statistical distribution of dynamic responses, the four-wheel slip ratios are presented as “Mean $\pm$ Standard Deviation (Std)”. The comparative results fully demonstrate the advantages of the proposed method. For the severe lane-change braking condition (No. 6), the MPC-AD strategy achieves a maximum lateral tracking error of 0.932 m, outperforming MPC-I (2.571 m) and LQR-AD (1.372 m) by a considerable margin.

Table 6. Comprehensive statistical metrics across various braking-in-turn scenarios.

Scenarios	Control Strategies	Max Lateral Error (m)	Yaw-Rate RMSE (rad/s)	Front Axle Torque Diff (N·m)	Front Axle Slip (Mean $\pm$ Std)	Rear Axle Slip (Mean $\pm$ Std)
Scenario No. 1	Proposed (MPC-AD)	N/A	N/A	109.02	0.098 $\pm$ 0.226	0.048 $\pm$ 0.099
	Baseline 1 (MPC-I)	N/A	N/A	152.86	0.154 $\pm$ 0.208	0.016 $\pm$ 0.014
	Baseline 2 (LQR-AD)	N/A	N/A	138.62	0.042 $\pm$ 0.098	0.080 $\pm$ 0.178
Scenario No. 2	Proposed (MPC-AD)	N/A	N/A	138.54	0.362 $\pm$ 0.364	0.266 $\pm$ 0.322
	Baseline 1 (MPC-I)	N/A	N/A	142.60	0.358 $\pm$ 0.351	0.255 $\pm$ 0.290
	Baseline 2 (LQR-AD)	N/A	N/A	154.43	0.362 $\pm$ 0.360	0.214 $\pm$ 0.275
Scenario No. 3	Proposed (MPC-AD)	N/A	N/A	78.63	0.378 $\pm$ 0.398	0.219 $\pm$ 0.305
	Baseline 1 (MPC-I)	N/A	N/A	57.07	0.437 $\pm$ 0.414	0.237 $\pm$ 0.351
	Baseline 2 (LQR-AD)	N/A	N/A	118.95	0.360 $\pm$ 0.380	0.270 $\pm$ 0.341
Scenario No. 6	Proposed (MPC-AD)	0.932	0.019	116.11	0.005 $\pm$ 0.008	0.002 $\pm$ 0.003
	Baseline 1 (MPC-I)	2.571	0.014	161.36	0.091 $\pm$ 0.157	0.008 $\pm$ 0.009
	Baseline 2 (LQR-AD)	1.372	0.021	242.75	0.003 $\pm$ 0.002	0.001 $\pm$ 0.002

Note: N/A indicates that the metric has no calculation significance for the corresponding fixed-steering-angle braking-in-turn maneuvers, with no predefined tracking path and target yaw rate.

5.3. HIL Experimental Verification

5.3.1. EMB HIL Experimental Platform

A HIL test platform combining a driving simulator with an EMB test bench is built to evaluate the real-time control effect of hierarchical coordination control strategies, as shown in Figure 14. The steering wheel and brake pedal are used as driver operation input devices, injecting steering angle and braking demand signals under the compound working conditions of braking-in-turn maneuver into the test platform; the SIMULINK hierarchical coordination control algorithm is deployed on an NI real-time industrial control computer connected through Transmission Control Protocol (TCP) communication cables and a host computer, ensuring the real-time nature of control decisions and dynamic simulation. The host computer, NI real-time industrial control computer, and EMB dedicated programmable power supply are all integrated in the braking system HIL test control cabinet, which supplies a stable 24 V DC working power to the EMB test bench and, at the same time, establishes a communication link through the CAN bus to ensure the issuance and execution status feedback of control commands. The host computer uses TSMaster with an integrated CarSim-Controller function module in conjunction with NI VeriStand to complete algorithm deployment, while being able to adjust and visualize the output of control algorithms, vehicle dynamics response, EMB clamping force, wheel speed, and slip ratio in real time, and can simultaneously record test process data, providing reliable support for the effectiveness verification of control strategies. Key signal specifications of the test system: driver inputs are sampled at 1 kHz; control commands are transmitted to EMB actuators via CAN bus at 500 Hz; EMB feedback signals (current, position, speed, clamping force) are sampled at 10 kHz. Notably, the current HIL platform only integrates EMB actuator hardware; other corner module components (steer-by-wire, in-wheel motor, suspension) are simulated using high-fidelity dynamic models.

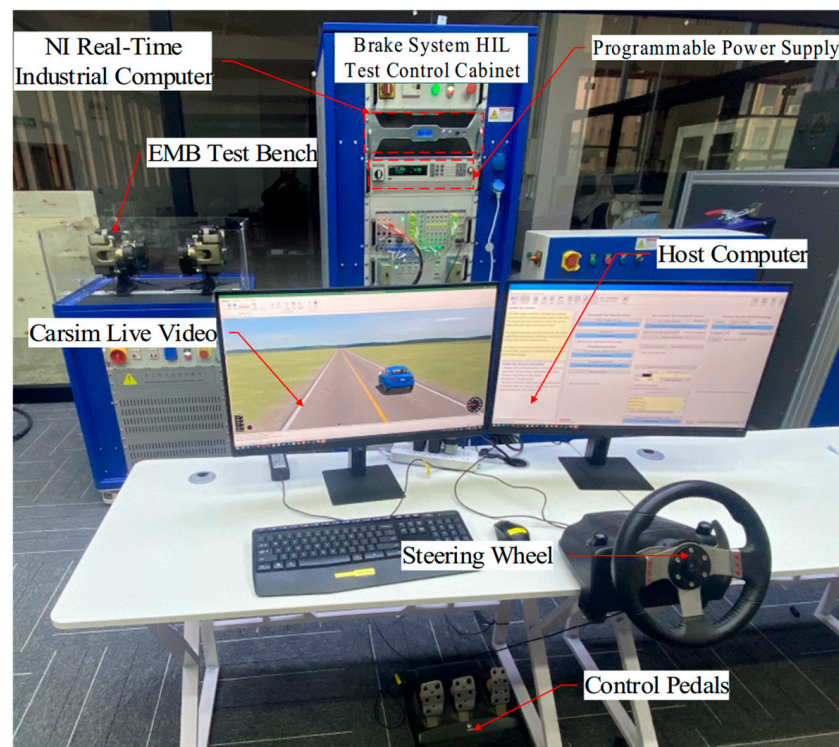


Figure 14. EMB system HIL test platform.

5.3.2. Results Discussion of Fixed Angle Emergency Braking Scenarios

Firstly, the three working conditions of the emergency braking scenario from a fixed steering angle are tested and verified. The steering wheel and pedal signals are consistent with the simulation. The test results on the variable adhesion road surface are shown in Figure 15, where the color of the four-wheel curves in Figure 15a corresponds to the co-simulation test, the light purple curve represents the vehicle speed, the trajectory curve for the 2WS condition is a red solid line, and the blue trajectory curve corresponds to the 4WS condition. In Figure 15b, the red vehicle corresponds to the 2WS condition, and the blue vehicle corresponds to the 4WS condition.

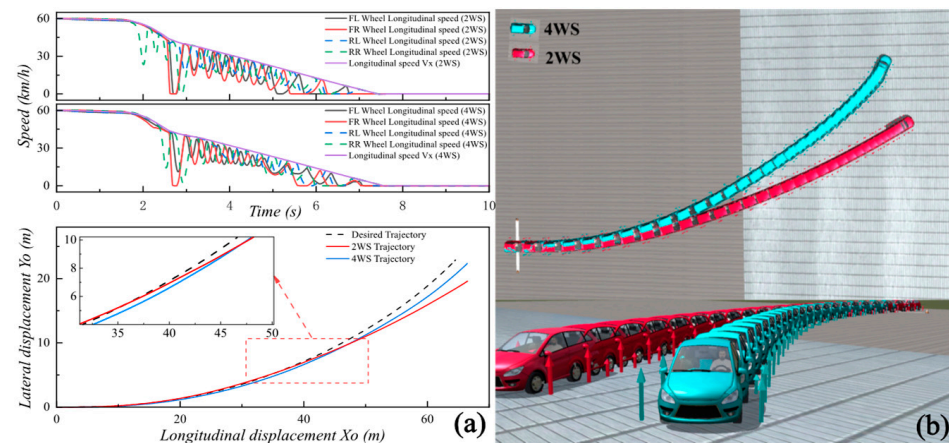


Figure 15. Comparison of emergency braking and variable adhesion conditions in vehicle speed and trajectory; (a) vehicle speed and trajectory curves on variable adhesion road surface at 60 km/h; (b) trajectory results from HIL test platform on variable adhesion road surface at 60 km/h.

Figure 15a shows the relationship between vehicle speed and wheel speed under two steering conditions. The oscillating changes in wheel speed indicate the effectiveness of the dynamic distribution of the control strategy and the ABS function, with the vehicle speed changing smoothly and basically consistent with the trend of simulated braking torque and lateral force changes; observing the trajectory curve in Figure 15a and the CarSim scenarios screenshot in Figure 15b of the HIL test platform, under the 4WS steering condition, the vehicle utilizes more lateral force and has better tracking performance.

Compared with the expected trajectory, the lateral error corresponding to the 2WS working condition can reduce 10.9%, and at the same time, the braking distance is relatively longer than that of the 2WS condition, with the braking time approximately 5.1 s; the vehicle achieves rapid braking force response on variable adhesion road surfaces, with smooth vehicle speed changes and good tracking effect of the braking-in-turn maneuver trajectory.

5.3.3. Results Discussion of Lane Change Braking Scenarios

During the lane change braking scenario test verification, the braking-in-turn maneuver parameter settings are consistent with the co-simulation. The test results for the variable adhesion road surface are shown in Figure 16, and the legend definitions are consistent with the previous text. From the comparison of the velocity curves in Figure 16a, the speed change range in the 4WS condition is about 3.4 km/h, and the speed change range in the 2WS condition is about 8.4 km/h. The control effect is stable under both steering conditions, and the speed change is smooth and continuous. As shown in Figure 16b, at the end of the test, the vehicle trajectory of the 2WS is slightly behind that of the 4WS, and the trace effect of the 4WS is better than that of the 2WS; the vehicle achieves smooth lane change braking under the coordinated control algorithm.

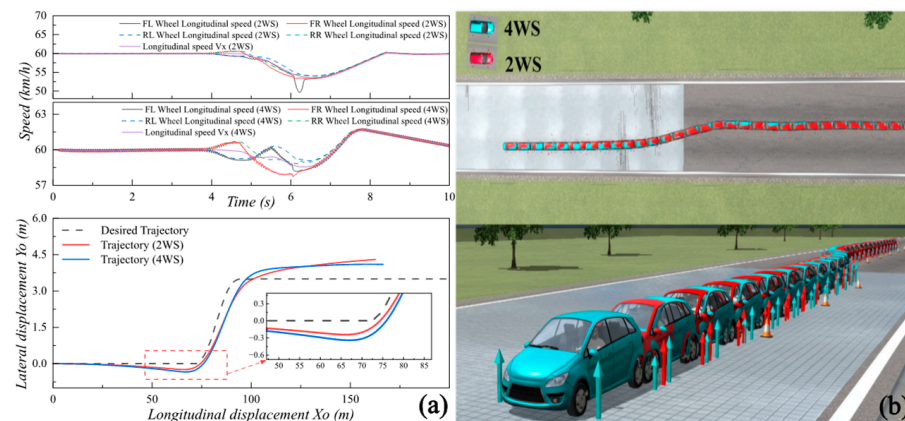


Figure 16. Comparison of vehicle speed and trajectory in lane change braking scenarios and variable adhesion conditions; (a) vehicle speed and trajectory curves on variable adhesion road surface at 60 km/h; (b) trajectory results from HIL test platform on variable adhesion road surface at 60 km/h.

6. Conclusions

This paper takes the 4WID-4WIS CAV as the research object. It proposes a novel hierarchical EMB coordination control strategy for the strong coupling between longitudinal and lateral dynamics in braking-in-turn maneuver conditions and the competition for tire adhesion resources. The feasibility and advantages of the strategy have been demonstrated through co-simulation in MATLAB/Simulink and CarSim under multiple operating conditions, as well as through HIL tests.

- (1) Under both 2WS and 4WS steering modes, the strategy can suppress yaw-rate fluctuations and stabilize lateral force changes under all operating conditions. Under emergency braking conditions with variable adhesion and a fixed steering angle, the tracking error in the 4WS mode is reduced by up to 10.9% compared to the 2WS mode, and the yaw rate is better controlled within the expected range than the reference methods, supporting the braking-in-turn requirements of vehicles with different steering configurations.
- (2) As a typical 4WS platform, the CAV serves as an ideal physical carrier for precise longitudinal-lateral coordinated control. Combined with the three-intensity adaptive braking force distribution algorithm, it reduces the front-rear axle braking torque fluctuation difference by 19.7% under 4WS mode, fully exploiting the over-actuated advantages.
- (3) To address the insufficient consideration of EMB dynamic characteristics in existing studies, the proposed two-stage identification method accurately captures EMB hysteresis and torque delay. Combined with the three-loop control incorporating improved ISMC, the response delay is reduced to 55 ms, enabling reliable execution of refined braking force distribution.

Subsequent research will focus on balancing tracking ability and braking distance, further advancing the refinement of braking force distribution coordination, realizing precise prediction, identification, and tracking control of braking force. Furthermore, the robustness of these empirical parameters against variations in vehicle mass, center of gravity, and road conditions will be systematically evaluated. Exploring redundant control strategies under system-failure scenarios and combining data-driven methods to refine tire modeling to improve control robustness further and support the large-scale application of corner module vehicles.

Author Contributions: Z.S. contributed to the conceptualization, methodology, software, visualization, and design of the study, as well as to the drafting and revision of the manuscript. M.C. and Y.Y. contributed to the conceptualization and design of the study, as well as to the drafting and revision of the manuscript. S.Z. assisted in the completion of the experiments and was involved in the drafting and revision of the manuscript. All authors have read and agreed to the published version of the manuscript.

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