

Slip ratio control based on adaptive fuzzy sliding mode for vehicle with an electromechanical brake system

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Abstract. The development of vehicle intelligence has driven the evolution of brake-by-wire systems, with electromechanical braking (EMB) emerging as a crucial development in intelligent vehicle braking. To enhance the braking safety of EMB-equipped vehicles, this paper proposes a slip ratio control method based on adaptive fuzzy sliding mode control (AFSMC) to more effectively achieve wheel slip ratio tracking during braking. The approach involves establishing the mathematical model of the planetary gear-type EMB system and the vehicle's longitudinal dynamics model. Additionally, a hierarchical collaborative control strategy is introduced, where the bottom layer employs the EMB clamping force control algorithm based on cascaded proportional-integral (PI), and the top layer integrates AFSMC to regulate the slip ratio. The simulation results, validated using the joint simulation platform of Simulink and Carsim under various conditions, illustrate that the AFSMC, compared to the conventional sliding mode controller (SMC), attains more precise control of wheel slip ratio while mitigating the chattering phenomenon. These findings suggest the potential of AFSMC for practical engineering applications.

Keywords: Slip ratio / electromechanical brake system / fuzzy sliding mode control / braking force control

1 Introduction

In recent years, the evolution of automotive intelligence and electrification has concurrently propelled advancements in braking systems, including Electro-Mechanical Braking (EMB), Electro-Hydraulic Braking (EHB) systems, and regenerative braking systems, among others, all of which have been extensively explored [1]. The EMB system, distinguished by its rapid response, enhanced execution efficiency, space-efficient design, and seamless integration with other functional modules, proves more adept for modern automotive integrated control strategies compared to conventional braking systems. This makes it evident that an increasing number of vehicles will incorporate EMB in the future, enhancing the driving experience and safety for both drivers and passengers. Braking performance stands as a critical metric for evaluating vehicle driving safety. However, the direct influence of slip ratio control on a vehicle's braking performance makes slip ratio control in vehicles equipped with EMB an intriguing and formidable challenge.

Typically, researchers actively adjust wheel torque by designing a control algorithm that incorporates both braking and driving torques. This is done to maintain

the wheel's slip ratio close to the anticipated value for the current road surface. Such an approach maximizes the utilization of road surface adhesion characteristics, aiming to achieve heightened longitudinal adhesion, prevent wheel lockups, and ultimately enhance automobile brake safety. Slip ratio control poses a challenge due to the inherent high nonlinearity and uncertainty in the braking system. Systems with anticipated braking torque as input exhibit strong nonlinear and time-varying characteristics, making accurate modeling a formidable task. Furthermore, uncertainties arise from variations in parameters like vehicle condition and road surface adhesion, adding to the complexity. Early scholars predominantly employed logic switching rules for robust tracking control of wheel slip ratio [2]. However, this method lacks the requisite flexibility and precision for executing high-precision, stable tracking control on any target slip ratio, falling short of the demands of modern intelligent vehicles. Contrastingly, control methods based on dynamic models simplify the achievement of continuous target slip ratio tracking control, garnering extensive attention from researchers. Noteworthy efforts by Yuan et al. [3] and Yan et al. [4] applied the nonlinear model predictive control (NMPC) algorithm to electric vehicle slip ratio control systems. By simultaneously regulating the moment of all four wheels, the algorithm maintains the slip ratio close to expectations. Simulation results indicate enhanced braking and driving

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forces, ensuring non-locking wheels during braking and controlled deceleration during driving. Similar research has been conducted using the MPC algorithm by other scholars [5,6]. However, the MPC algorithm's computational demands make it less suitable for engineering development. Castro et al. [7,8] introduced an integrated sliding mode algorithm for anti-saturation conditions, estimating the upper bound of the adhesion coefficient uncertainty based on anticipated torque. This enhances robustness and steady-state precision in slip ratio control, albeit with a reduction in dynamic performance due to integral effects. Sun et al. [9] and Zhou et al. [10] explored the effects of fuzzy sliding mode control (FSMC) and SMC in slip ratio control, with simulation results indicating superior effectiveness, albeit with some limitations in complex working conditions. Boopathi et al. [11] proposed a fuzzy sliding mode controller for the quarter car ABS model, demonstrating high accuracy under medium and low-speed conditions but lacking verification under high-speed conditions.

For vehicles equipped with EMB, precise control of the lower braking system is as critical as the upper slip ratio control algorithm. The EMB system, distinguished by its rapid response time and enhanced execution efficiency compared to conventional hydraulic brakes, introduces its own set of uncertainties and nonlinear challenges. Consequently, ensuring the optimal state of the EMB system becomes an imperative challenge in controller design. Chihoon et al. [12] employed an adaptive PID algorithm to govern the EMB clamping force. They dynamically adjusted the proportional gain in response to variations in the input clamping force while simultaneously estimating the clamping force. Experimental findings validated the algorithm's ability to deliver satisfactory clamping force control performance. Lee et al. [13] tackled clamping force control using the nonlinear Model Predictive Control (MPC) algorithm, formulating the control law by minimizing a quadratic performance index. Simulation results revealed a significantly shortened controller response time, albeit with increased demands on hardware storage capacity. In another approach, Park et al. [14] devised an adaptive Sliding Mode Control (SMC) capable of online identification of parameters in the total friction model. Integrating these parameters into the SMC enhanced the robustness of friction tracking, although it considered fewer parameters, limiting its applicability. Peng et al. [15] and Zhao et al. [16] explored EMB clamping force control using Fuzzy Sliding Mode Control (FSMC) and SMC with an improved approach rate, respectively, yielding acceptable simulation results. Furthermore, Eum et al. [17] applied a robust control algorithm to regulate the restraining force in urban public transportation as their research subject.

Motivated by the above works, aiming at vehicles equipped with EMB, combining the EMB system execution characteristics and slip ratio control strategy, this paper proposes a hierarchical control strategy to enhance the vehicle's braking performance. First, the upper sliding mode controller is designed to control the slip ratio, and second, a fuzzy control algorithm is designed innovatively

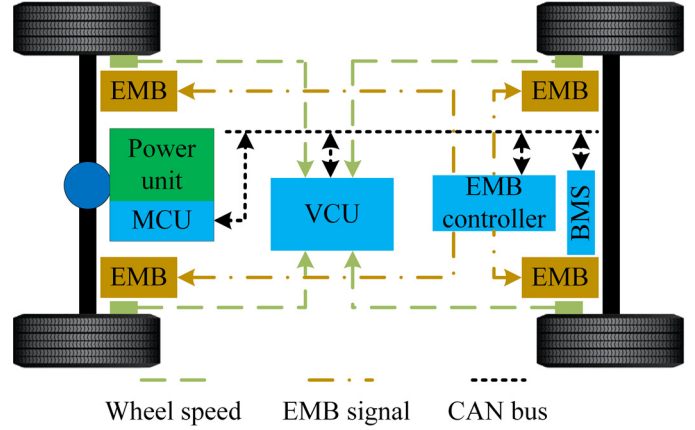


Fig. 1. Simplified vehicle structure diagram with EMB system.

to adaptively modify the switching gain of the sliding control in order to suppress the chattering phenomenon of the system. In order to further enhance the dynamic performance of the EMB system, a cascade PI clamping force control method is proposed for the lower execution layer to efficiently execute the upper layer instructions. Finally, the control effect is validated by the joint simulation platform Simulink and Carsim.

The remaining sections are structured as follows: Section 2 outlines the simplified vehicle structure, the vehicle model, and the mathematical model of the EMB system. Section 3 provides detailed insights into the designed hierarchical controllers, encompassing the upper-layer adaptive fuzzy sliding mode slip ratio controller and the lower-layer EMB cascaded PI clamping force controller. Section 4 showcases the simulation results employed to validate the efficacy of the controllers. Finally, Section 5 encapsulates the key research findings of this paper.

2 System scheme

The focus of this paper is on vehicles equipped with the EMB system. In Figure 1, we present a simplified structure of a front-wheel-drive vehicle featuring an electromechanical braking system. This vehicle is a front-drive electric vehicle, with the power unit and its micro control unit (MCU) situated at the front axle, the vehicle control unit (VCU) overseeing the entire chassis domain, and the battery management system (BMS) efficiently distributing battery energy to various controllers and actuators. As depicted in Figure 1, the integration of the EMB system allows for a more streamlined chassis layout compared to conventional vehicles. The four EMB systems are positioned on each of the four wheels, with the EMB controller receiving commands from the VCU to instruct the EMB actuators. This design offers the advantage of swiftly implementing braking commands, whether initiated by the vehicle's automated braking system or through driver input, enabling independent control of the braking force at each wheel.

Table 1. Parameters of the tyre.

a_0	a_1	a_2	a_3	a_4	a_5	a_6	a_7	a_8
0	1000	1.55	60	300	0.17	0	0	0.8

2.1 Vehicle and Tyre models

As our study is primarily focused on the precise control of the vehicle's slip ratio during braking, we have deliberately chosen to narrow our scope to the longitudinal motion of the vehicle under braking. This deliberate omission of transverse and vertical motions during braking is driven by the need to maintain a focused investigation without over-generalizing or introducing unnecessary complexity. Consequently, this paper exclusively models the longitudinal dynamics of a braking vehicle.

For braking control, the vehicle can be considered a rigid body, and only the longitudinal motion is taken into account. The dynamic equation can be expressed as:

$$m\dot{V}_x = -(F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}). \quad (1)$$

The rotational motions of the wheel can be expressed as:

$$J_i\dot{\omega}_i = F_{xi}R_{wi} - T_{bi} \quad (2)$$

Where, m is vehicle mass, V_x is the longitudinal speed of vehicle, F_{xi} is the longitudinal force of the wheel, ω_i is the angular speed of the wheel, R_{wi} is the radius of the wheel, J_i is the rotational inertia of the wheel, T_{bi} is the braking torque, $i \in \{fl, fr, rl, rr\}$, which stands for left front wheel, right front wheel, left rear wheel and right rear wheel.

Tyres are the only part connecting the body of the vehicle to the road, and the movement of the vehicle depends on the forces on the tyres. However, the tyre's behavior is highly characterized by nonlinearity in complicated road condition [18]. The Magic Formula is used to reflect the transient characteristics of the tyre [19], in this paper. The longitudinal tyre force can be given as:

$$F_{xi} = D \sin\{C \arctan[B\lambda_i(1 - E) + E \arctan(B\lambda_i)]\} \quad (3)$$

where, B is the stiffness factor, C is the shape factor, D is the peak factor and E is the curvature factor, which can be given as:

$$D = a_0 F_{zi}^2 + a_1 F_{zi}. \quad (4)$$

$$C = a_2. \quad (5)$$

$$B = \frac{a_3 F_{zi}^2 + a_4 F_{zi}}{C D e^{a_5 F_{zi}}}. \quad (6)$$

$$E = a_6 F_{zi}^2 + a_7 F_{zi} + a_8. \quad (7)$$

The slip ratio λ_i can be described as:

$$\lambda_i = \begin{cases} \frac{V_x - \omega_i R_{wi}}{V_x} (V_x \neq 0), \text{braking} \\ \frac{\omega_i R_{wi} - V_x}{\omega_i R_{wi}} (\omega_i \neq 0), \text{driving} \end{cases}. \quad (8)$$

The other parameters are shown in Table 1. It should be noted that different brands and models of tyre parameters are not the same.

2.2 EMB model

As illustrated in Figure 2, the examined EMB system in this paper comprises a motor, a planetary-type reduction gear, and a ball screw. During the operation of the EMB system, the rotational motion of the motor is converted into linear motion of the ball screw nut through the planetary-type reduction gear and the ball screw. Subsequently, the ball screw applies pressure to the ball screw nut, generating a clamping force between the friction pad and the brake disk.

To comprehensively study the EMB system, we have developed a dynamic model for the EMB.

2.2.1 Motor

The motor serves as the power source for the EMB system, directly influencing its performance, making it a crucial component. However, as precise motor modeling is not the primary focus of this paper, a simplified mathematical model is provided:

To maintain model feasibility while reducing analytical and computational complexity, the following assumptions are considered: (1) Neglecting the saturation of the motor stator core, omitting the cogging effect and armature reaction. (2) Disregarding eddy current or hysteresis losses. (3) Assuming uniform distribution of armature windings over the inner surface of the stator.

$$U = L\dot{I} + R_m I + E. \quad (9)$$

$$E = K_e \omega_m. \quad (10)$$

$$J\dot{\omega}_m = T_m - T_f - T_L. \quad (11)$$

$$T_m = K_t I. \quad (12)$$

Where, U is the armature voltage. E is the counter electromotive force. L is the armature inductance. I is the armature current. R_m represents the stator resistance.

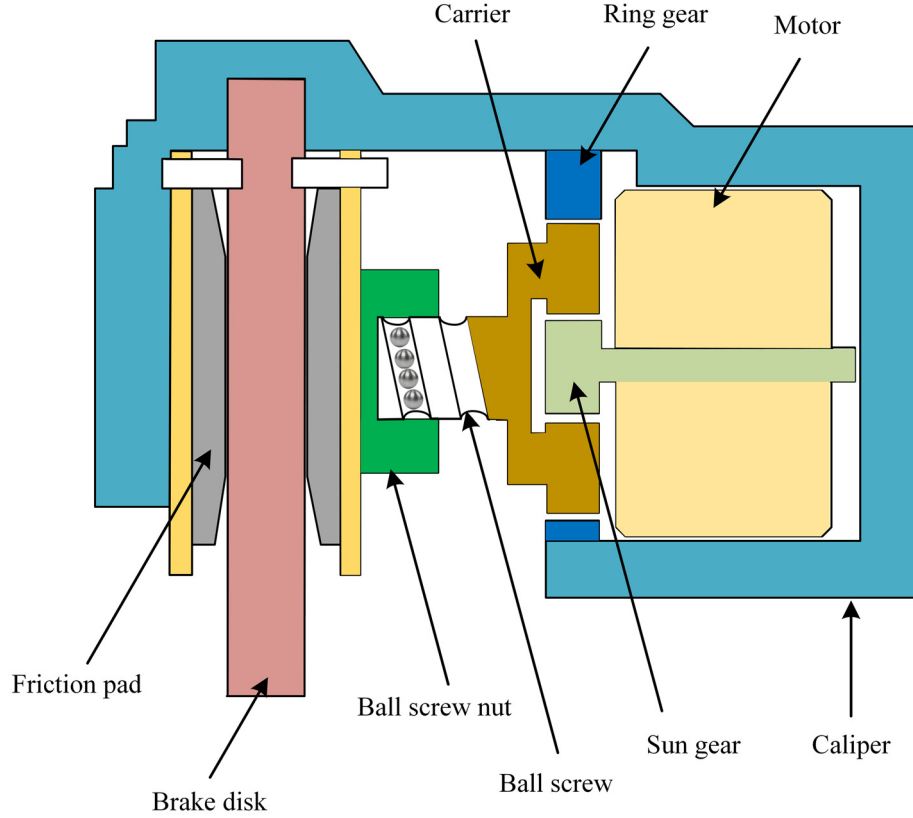


Fig. 2. Schematic diagram of the EMB system.

K_e represents the EMF constant. ω_m represents the actual motor speed. T_m is the motor stall torque. J represents the rotary inertia. K_t represents the torque constant. T_f represents the friction torque. T_L represents the load torque.

Mechanical friction is inherent in the operation of the EMB system, and there exists a direct relationship between the friction torque of the motor and the brake clamping force. This paper adopts a general model of static friction characteristics. When the motor is in operation, its friction torque can be expressed as follows:

$$T_f = \begin{cases} T_e & (\omega_m = 0 \& |T_e| < T_s) \\ T_s \text{sgn}(T_e) & (\omega_m = 0 \& |T_e| \geq T_s) \\ T_c \text{sgn}(\omega_m) + B_v \omega_m & (\omega_m \neq 0) \end{cases} \quad (13)$$

Where, B_v represents the viscous friction coefficient, T_e represents the external torque, T_s represents the maximum static friction torque, T_c represents the coulomb friction torque.

2.2.2 Transmission mechanism model

The current state of motor technology falls short of delivering the necessary braking torque independently. Therefore, a torque-enhancing device is indispensable. This

paper utilizes a planetary-type reduction gear for this purpose. The input end of the planetary gear is linked to the motor's rotor, and the output end is connected to the ball screw pair, as illustrated in Figure 2.

The ball screw converts the planetary gear's rotational motion into axial motion. Regardless of the elastic deformation, the following kinematic equation is constructed:

$$x_E = \frac{\theta_m L_0}{2\pi i_p} \quad (14)$$

Where, x_E is the displacement of the ball screw. θ_m is the rotation angle of the motor shaft. L_0 is the lead of the ball screw. i_p is the reduction ratio of the planetary gear.

There is adequate clearance between the brake disk and friction pad. The approximate mathematical relationship between the displacement of the ball screw and the clamping force obtained by fitting experimental data is as follows [16]:

$$F_E = \begin{cases} K_s(x_E - D_E), & (x_E \geq D_E) \\ 0, & (0 \leq x_E < D_E) \end{cases} \quad (15)$$

Where, F_E is the clamping force, K_s is the clamping force coefficient, D_E is the gap distance between the brake disk and friction pad.

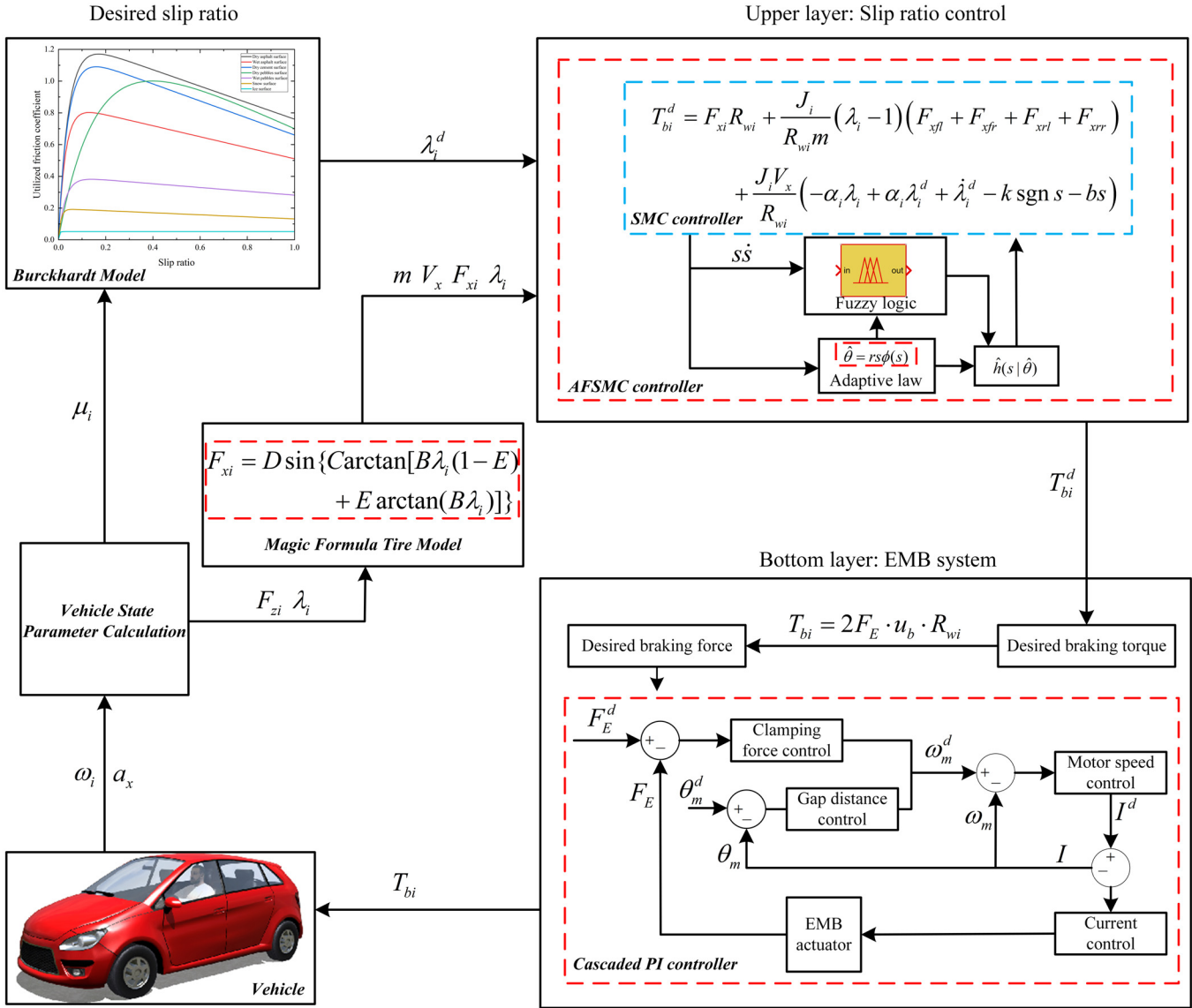


Fig. 3. Sketch of the hierarchical controller.

The load of the drive motor can be expressed as follows:

$$F_{ES} = \frac{2\pi i_p T_L}{L_0}. \quad (16)$$

In this paper, since both sides of the brake disk are subjected to friction when the EMB system is working, the brake model can be expressed as:

$$T_{bi} = 2F_E \cdot u_b \cdot R_{wi}. \quad (17)$$

where, u_b is friction pad coefficient.

3 Hierarchical controller design

As a result of the superior qualities it possesses, hierarchical control has found widespread use in the field of engineering [20]. In the context of this work, it is meant to make the

burden of computation lighter and to promote compatibility with a variety of vehicle configurations [21]. The overall logic of the hierarchical controller designed in this paper is shown in Figure 3.

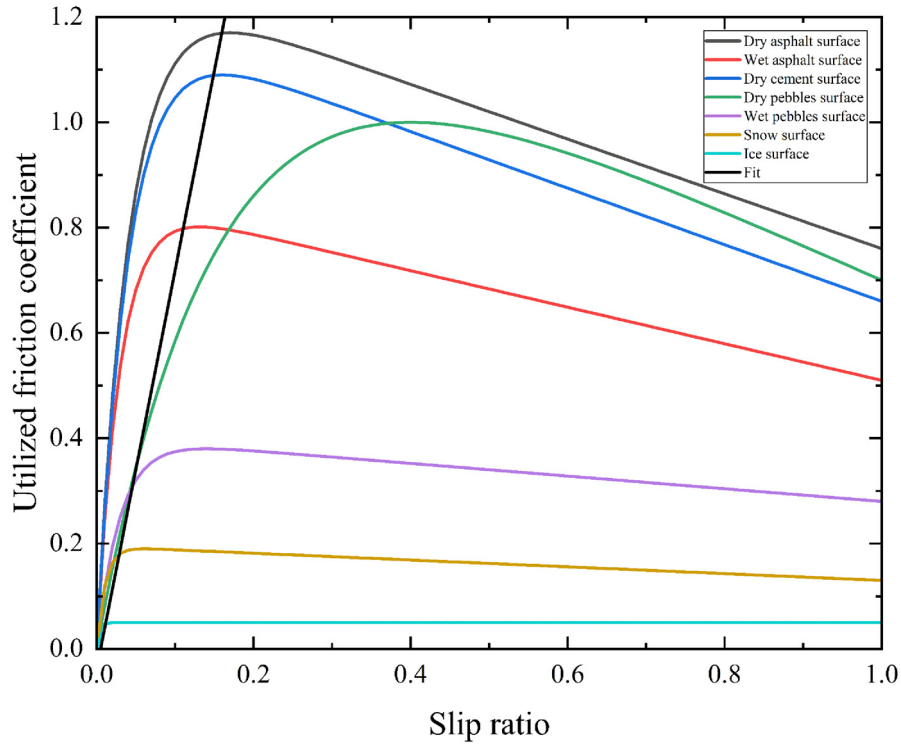
In this paper, the objective of proposed controller is to control slip ratio to track the desired value and to obtain better driving safety by regulating the braking torque provided by the electromechanical brake system. The controller consists of upper fuzzy sliding mode controller and lower execution controller.

3.1 Desired slip ratio estimation

The desired ratio in different conditions is different. When the braking severity is small and the anti-lock braking system (ABS) is not triggered, the control objective is to meet driver's brake demand. When the required braking severity exceeds the maximum value provided by the road

Table 2. The parameters of Burckhardt's model for different road surfaces.

Road surfaces	c_1	c_2	c_3
Dry asphalt surface	1.2801	23.99	0.52
Wet asphalt surface	0.8570	33.822	0.347
Dry cement surface	1.1973	25.168	0.5373
Dry pebbles surface	1.3713	6.4565	0.6691
Wet pebbles surface	0.4004	33.7080	0.1204
Snow surface	0.1946	94.129	0.0646
Ice surface	0.05	306.39	0

**Fig. 4.** Relationship between desired slip ratio and road friction.

friction which studied by this paper, the control objective is to make full use of road friction. In this case, the desired slip ratio can be achieved based on the μ - λ curve as shown in Figure 4.

In fact, this curve is fitted according to the Burckhardt tyre model [22]. Without considering the influence of vehicle speed and tyre load, the formula can be expressed as:

$$\mu(\lambda) = c_1(1 - e^{-c_2\lambda}) - c_3\lambda \quad (18)$$

where, c_1, c_2 and c_3 are the fitting parameters for the experimental data. As shown in Table 2.

It can be seen from Figure 4 that the desired slip ratio has an approximately linear relationship with the road friction. Therefore, the relationship between the peak value

of the road friction coefficient and the desired slip ratio can be expressed as:

$$\lambda^d = c_4\mu + c_5. \quad (19)$$

By approximate fitting, values of c_4 and c_5 can be given as $c_4 = 0.132$ and $c_5 = 0.005$, respectively.

3.2 Top layer: Slip ratio controller

The sliding mode controller has the properties of finite convergence for a highly nonlinear system, insensitivity to parameter changes and disturbances, no need for online system identification, and a low computation burden, allowing it to effectively control the slip ratio. However, it has chattering problems, which affect the control accuracy

and even cause the system to oscillate and become unstable. Therefore, in order to weaken the chattering phenomenon, the fuzzy corrector is designed by fuzzy control method, and the switching control amount is adjusted according to the sliding mode arrival condition. Figure 5 depicts the structure of the adaptive fuzzy sliding mode controller.

The fuzzy sliding mode controller is chosen as the top layer slip ratio controller. It determines the desired braking torque T_{bi}^d to make λ_i track the desired slip ratio λ_i^d .

By differentiating formula (8), λ_i in braking process can be expressed as follows:

$$\dot{\lambda}_i = -\frac{\dot{V}_x(\lambda_i - 1) + \dot{\omega}_i R_{wi}}{V_x} \quad (20)$$

Substituting formula (1) and (2) into formula (20) results in:

$$\dot{\lambda}_i = \frac{(1/m)(1 - \lambda_i)(F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}) - (R_{wi}/J_i)(R_{wi}F_{xi} - T_{bi})}{V_x} \quad (21)$$

The error of actual slip ratio and desired slip ratio can be expressed as:

$$e_i = \lambda_i - \lambda_i^d \quad (22)$$

The proportional-integral form switching surface is used to eliminate the static error of slip ratio, which can be expressed as:

$$s_i = e_i + \alpha_i \int_0^t e_i dt \quad (23)$$

where, α_i is the gain parameters.

Differentiating formula (20) and substituting formula (18) into, $\dot{s}_i$ can be expressed as:

$$\dot{s}_i = \frac{(1/m)(1 - \lambda_i)(F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}) - (R_{wi}/J_i)(R_{wi}F_{xi} - T_{bi})}{V_x} - \dot{\lambda}_i^d + \alpha_i(\lambda_i - \lambda_i^d) \quad (24)$$

The slide mode control law is designed as:

$$\dot{s}_i = -k \operatorname{sgn}(s) - bs \quad (25)$$

where, $k > 0$, $b > 0$, and they are both positive constants.

Substituting formula (25) into (24), the braking torque T_{bi}^d requested to follow desired slip ratio can be expressed as:

$$\begin{aligned} T_{bi}^d = & F_{xi} R_{wi} + \frac{J_i}{R_{wi} m} (\lambda_i - 1) (F_{xfl} + F_{xfr} \\ & + F_{xrl} + F_{xrr}) + \frac{J_i V_x}{R_{wi}} (-\alpha_i \lambda_i + \alpha_i \lambda_i^d \\ & + \dot{\lambda}_i^d - k \operatorname{sgn} s - bs). \end{aligned} \quad (26)$$

Finally, the system's stability is examined using formula (25) as the control law. The stability is demonstrated by *Lyapunov* theory as follows:

The *Lyapunov* function can be expressed as:

$$V = \frac{1}{2} s^2 \quad (27)$$

By substituting the formula (27), it can be obtained as follows:

$$\dot{V} = s \dot{s} = s(-k \operatorname{sgn} s - bs) = -k|s| - bs^2 < 0. \quad (28)$$

However, it can be seen in formula (26) that while the existence of a symbolic function can eliminate interference terms, it inevitably results in chattering and cannot be changed once the value of the switching gain coefficient is determined, which is subject to certain constraints. Therefore, in this paper, adaptive fuzzy control is used to continue the discrete sign function to improve the system's trajectory near the sliding surface, which can effectively reduce the jitter phenomenon.

The fuzzy system is designed by using product inference machine, single value ambiguity and center average ambiguity resolver respectively, and its output $u(x)$ can be expressed as:

$$u(x) = \frac{\sum_{j=1}^m u^j \left(\prod_{i=1}^n \mu_{A_i^j(x_i)} \right)}{\sum_{j=1}^m \left(\prod_{i=1}^n \mu_{A_i^j(x_i)} \right)} \quad (29)$$

In this section, the switching function $s(t)$ is used as the input of the fuzzy system, where A_i^j is its fuzzy set $\{NB \ NS \ ZO \ PS \ PB\}$ and $\mu_{A_i^j(s_i)}$ is the membership function of S_i :

$$\mu_{NB}(s) = \frac{1}{1 + \exp(5(s + 4))}, \quad (30)$$

$$\mu_{NS}(s) = \frac{1}{1 + \exp(5(s + 2))}, \quad (31)$$

$$\mu_{ZO}(s) \exp(-s^2), \quad (32)$$

$$\mu_{PS}(s) = \frac{1}{1 + \exp(5(s - 2))}, \quad (33)$$

$$\mu_{PB}(s) = \frac{1}{1 + \exp(5(s - 4))}. \quad (34)$$

The fuzzy system $\hat{h}(s, \hat{\theta})$ of the following formula (35) is used as the output of the fuzzy system to continuously approximate $k \operatorname{sgn} s$:

$$\hat{h}(s|\hat{\theta}) = \hat{\theta}^T \Phi(s), \quad (35)$$

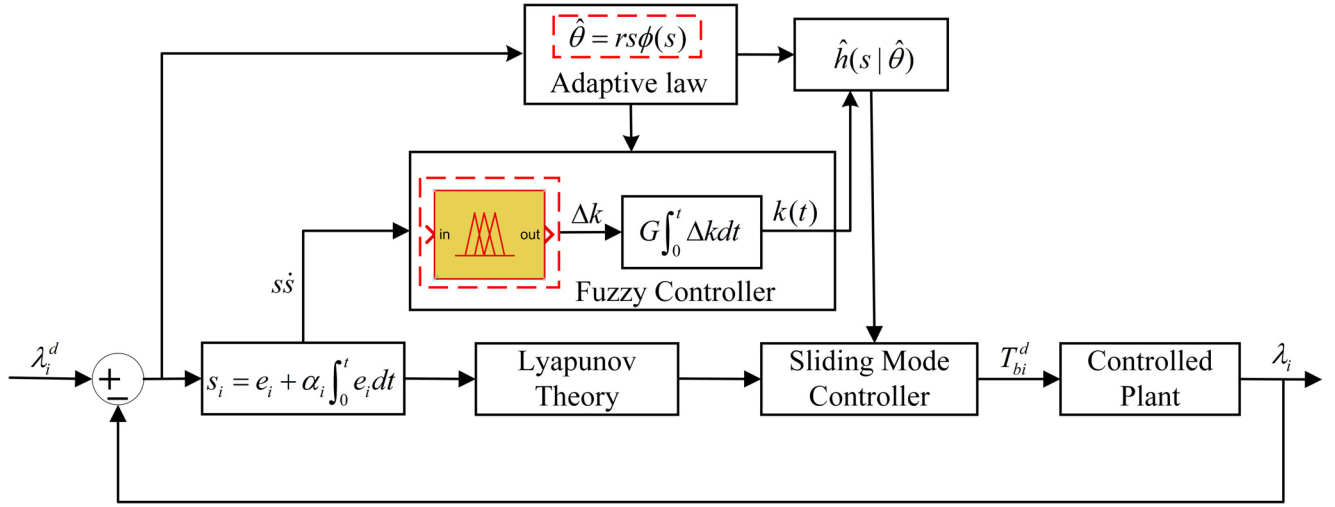


Fig. 5. Diagram of adaptive fuzzy sliding mode controller.

$$\frac{\left(\prod_{i=1}^n \mu_{A_i^j(s_i)} \right)}{\sum_{j=1}^m \left(\prod_{i=1}^n \mu_{A_i^j(s_i)} \right)} \quad (36)$$

Where $\phi(s)$ is the fuzzy vector that satisfies the form of (36), θ^T is the adjustment degree vector, it varies with the adaptive law. Ideally, $\hat{h}(s|\hat{\theta}) = k \operatorname{sgn} s$, however, due to the complex and changeable driving conditions in the actual braking process, it is difficult to ensure that it is in an ideal state for a long time. In order to be able to generate $\hat{h}(S, \hat{\theta})$ online in real time so that it can approach $k \operatorname{sgn} s$ infinitely, the following adaptive law is designed to adjust θ :

$$\dot{\hat{\theta}} = rs\phi(s) \quad (37)$$

Where, r is a positive constant, which is set according to the state of the system.

The optimal adjustment parameter $\hat{\theta}^*$ is determined as follows:

$$\hat{\theta}^* = \arg \min_{\theta \in \Omega} [\sup |\hat{h}(s|\hat{\theta}) - k \operatorname{sgn} s|] \quad (38)$$

Where, Ω is the collection of θ . According to formulas (37) and (38), the $\phi(s)$ should be adjusted in real time following the change of the switching $s(t)$ function, and the change should be made with the minimum regulation error $e_\theta = \hat{\theta} - \hat{\theta}^*$.

Finally, after introducing, $\hat{h}$ formula (26) can be rewritten as:

$$T_{bi}^d = F_{xi} R_{wi} + \frac{J_i}{R_{wi} m} (\lambda_i - 1) (F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}) + \frac{J_i V_x}{R_{wi}} (-\alpha_i \lambda_i + \alpha_i \lambda_i^d + \dot{\lambda}_i^d - \hat{h}(s|\hat{\theta}) - bs). \quad (39)$$

It is important to observe that the coefficient k of $k \operatorname{sgn} s$ should be able to change following the motion state of the system, so that the fuzzy system $\hat{h}(s, \hat{\theta})$ after fuzzy approximation has high adaptability to working conditions. As a result, the variable switching gain is also designed in this paper. It is then changed using fuzzy rules in accordance with the relative location and motion trend of the system and the sliding mode surface.

The parameters and are transformed into the fuzzy sets $s \in [-2, 2]$ of and, $k \in [-2, 2]$ respectively, and is regarded as the fuzzy input and as the output. The corresponding fuzzy linguistic variables are, $ss = \{NB \ NM \ NS \ ZO \ PS \ PM \ PB\}$, $\Delta k = \{NB \ NM \ NS \ ZO \ PS \ PM \ PB\}$. The $ss < 0$ shows that the sliding mode function's state is contrary to the present trend, the system is approaching the sliding mode surface, and the switching gain k should be decreased. The $ss > 0$ shows that the sliding mode function's present state is the same as the changing trend, the sliding mode surface is typically distant from the sliding mode surface, then the switching gain k should be increased.

However, the magnitude of $|ss|$ must also be taken into account in order to rationalize the design of fuzzy rules. When $|ss|$ is large, $|k|$ should have a larger change, and when $|ss|$ is small, $|k|$ should have a smaller change. The membership function diagram of fuzzy rules and fuzzy systems can be derived from the analysis presented above. As shown in Figures 6 and 7.

The design of the fuzzy rules is an essential link in determining the performance of the fuzzy controller. This paper presents seven fuzzy principles for the switching gain coefficient $k(t)$ as follows:

- R1: IF ss is PB THEN Δk is PB;
- R2: IF ss is PM THEN Δk is PM;
- R3: IF ss is PS THEN Δk is PS;
- R4: IF ss is ZO THEN Δk is ZO;
- R5: IF ss is NS THEN Δk is NS;
- R6: IF ss is NM THEN Δk is NM;
- R7: IF ss is NB THEN Δk is NB.

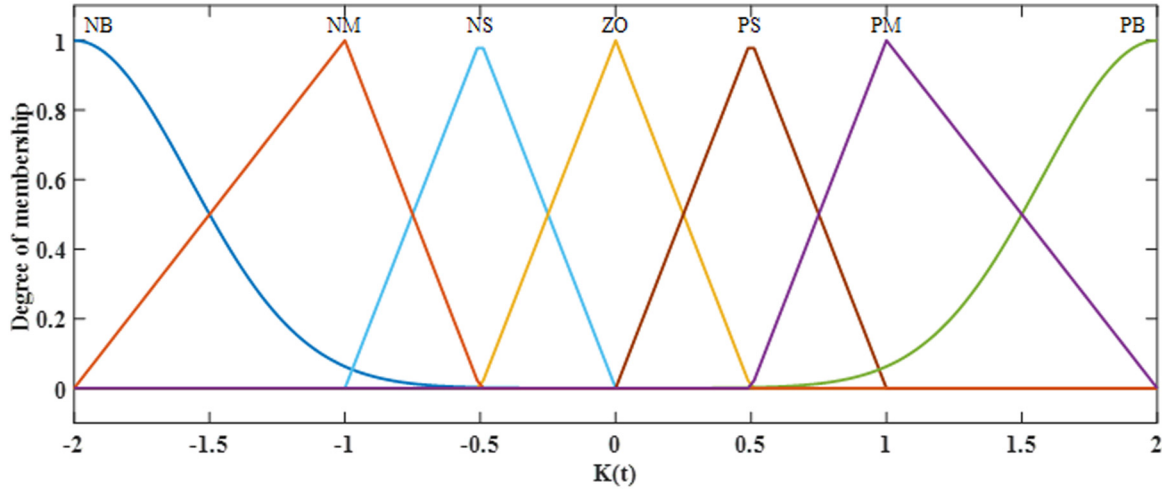


Fig. 6. Membership function of switching gain coefficient $k(t)$.

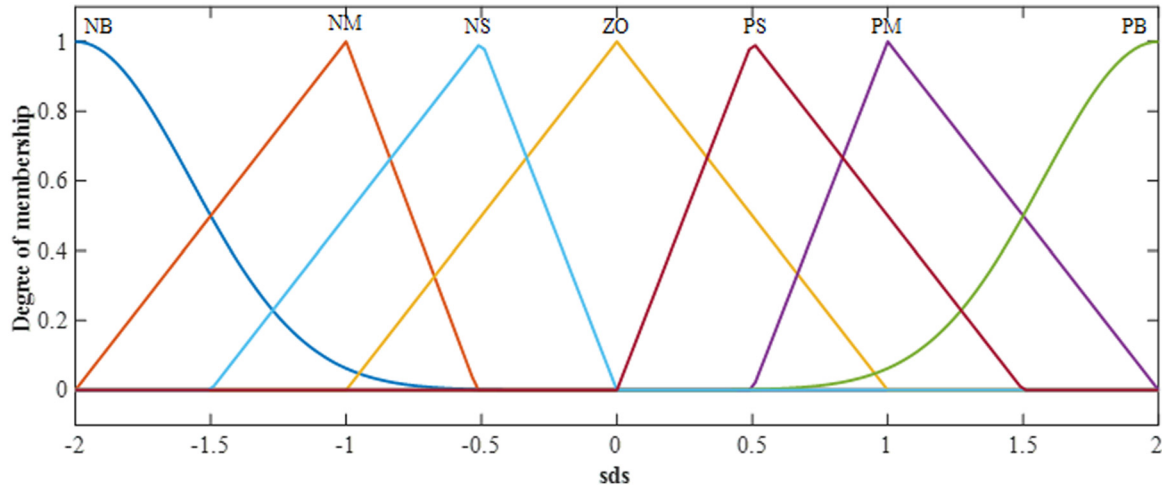


Fig. 7. Membership function of the product of sliding mode surface function and its derivative $\dot{s}$.

The final designed adaptive fuzzy sliding mode controller can make the fuzzy system $\hat{h}(s, \hat{\theta})$ actively approach the sign function with variable gain in real time, and reduce chattering on the basis of self-adaptation.

3.3 Bottom layer: EMB controller

3.3.1 Working process of EMB system

Considering the actual operating conditions of the EMB system, the complete control process, from the moment the EMB controller receives the braking instruction until the fixed gap distance is re-established between the friction pad and the brake disk, can be divided into three main processes: the gap elimination process, the braking pressure following process, and the gap distance adjustment process.

In the gap elimination process, upon receiving the command, the motor operates rapidly, generating thrust through the transmission system to push the ball screw until the brake disk and friction pad come into contact.

During this process, referred to as the gap elimination process, the motor's load is nearly zero. Therefore, a constant torque control based on the conventional PI control algorithm is employed to quickly eliminate the backlash. This control strategy allows the motor to reach its rated speed rapidly, ensuring the completion of this process in the shortest time possible.

However, as the vehicle's mileage increases, the performance of the friction pads and brake disks may deteriorate, causing the gap between them to widen. This deterioration can extend the time required to eliminate the gap. Moreover, the uneven degradation of the brake disks and friction pads on each wheel may result in inconsistent EMB system performance, potentially leading to vehicle instability during braking. To address this situation, a gap distance adjustment strategy needs to be designed. Given that this is not the primary focus of this paper and extensive research by numerous scholars has already been conducted on the subject, it will not be reiterated here.

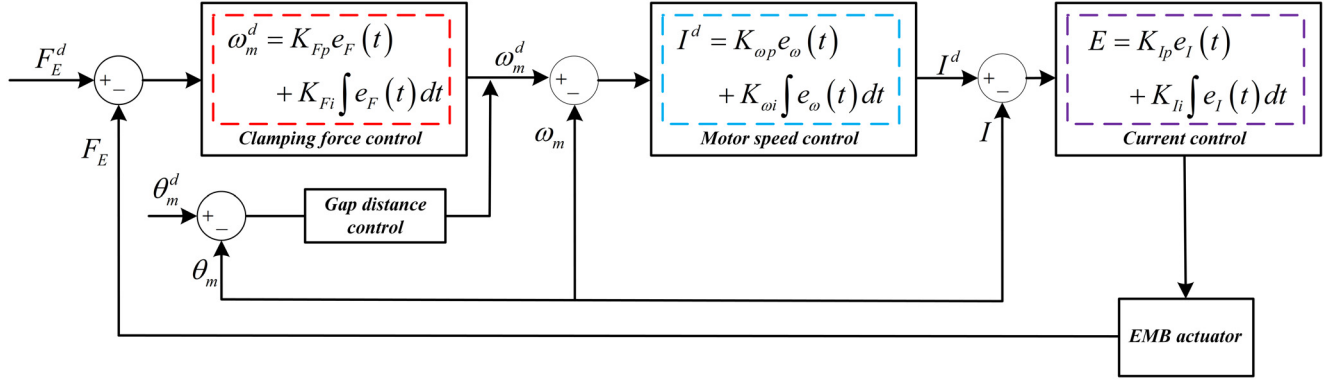


Fig. 8. Cascaded PI controller of EMB.

The braking pressure following process involves the brake disk and friction pad mutually extruding to the specified position from initial contact. During this process, the braking force rapidly increases until it reaches the desired level. Gap distance adjustment is the process of the friction block reverting to its initial position following the conclusion of the braking command.

The characteristics of these three processes in the clamping force control can be referred in Figure 11.

3.3.2 Clamping force controller design

The above introduction in this paper illustrates that the working process of the EMB is comparatively complex, making it challenging to develop an accurate model. Therefore, the clamping force controller of the EMB must possess not only accurate monitoring capabilities but also resistance to external load disturbances and other parameters. In this paper, a cascaded PI controller is employed to design a clamping force controller with high precision that is insensitive to model precision. The control flow is depicted in Figure 8. The control objective is to drive the clamping force F_E track the desired clamping force F_E^d by controlling the motor current.

The formula for the PI controller, which incorporates control error, proportional gain, and integral gain, is as follows:

$$U(t) = K_p e(t) + K_i \int e(t) dt \quad (40)$$

Where, the proportional gain of the PI controller is denoted by K_p . As a result of proportional adjustment, the system's deviation quickly shifts to a decreasing trend. Increasing the proportional gain can speed up the rate of adjustment, but a proportional gain that is too high will cause a large overshoot, reducing the system's stability and possibly causing instability. K_i is the gain integral. The integral adjustment can eliminate system steady-state error and enhance system steady-state precision, but it will slow the system's dynamic response speed.

The clamping force control equation can be expressed as:

$$e_F = F_E - F_E^d, \quad (41)$$

$$\omega_m^d = K_{Fp} e_F(t) + K_{Fi} \int e_F(t) dt, \quad (42)$$

The motor speed control equation can be expressed as:

$$e_\omega = \omega_m - \omega_m^d, \quad (43)$$

$$I^d = K_{\omega p} e_\omega(t) + K_{\omega i} \int e_\omega(t) dt. \quad (44)$$

The current control equation can be expressed as:

$$e_I = I - I^d, \quad (45)$$

$$E = K_{Ip} e_I(t) + K_{Ii} \int e_I(t) dt. \quad (46)$$

4 Simulation results and discussion

To verify the effect of the adaptive fuzzy sliding mode-based slip ratio controller and cascaded PI clamping force controller designed in this paper, based on the Simulink and Carsim joint simulation platform, they were tested under various working conditions. And the schematic structure of the simulation can be referred in Figure 9.

4.1 Simulation analysis of EMB Actuator

Throughout the vehicle's braking process, especially during emergency braking with the involvement of the ABS system, the braking system undergoes frequent operations, undoubtedly requiring enhanced stability and response speed. The frequency response of the EMB is analyzed in Simulink to confirm the effectiveness of the EMB clamping force control algorithm proposed in this paper under such operational conditions. The parameters of the planetary gear electromechanical braking system presented in this paper are detailed in Table 3.

The simulation working conditions designed in this paper are as follows: the desired clamping force of the brakes is specified as 15 kN, the fluctuation range is 30%,

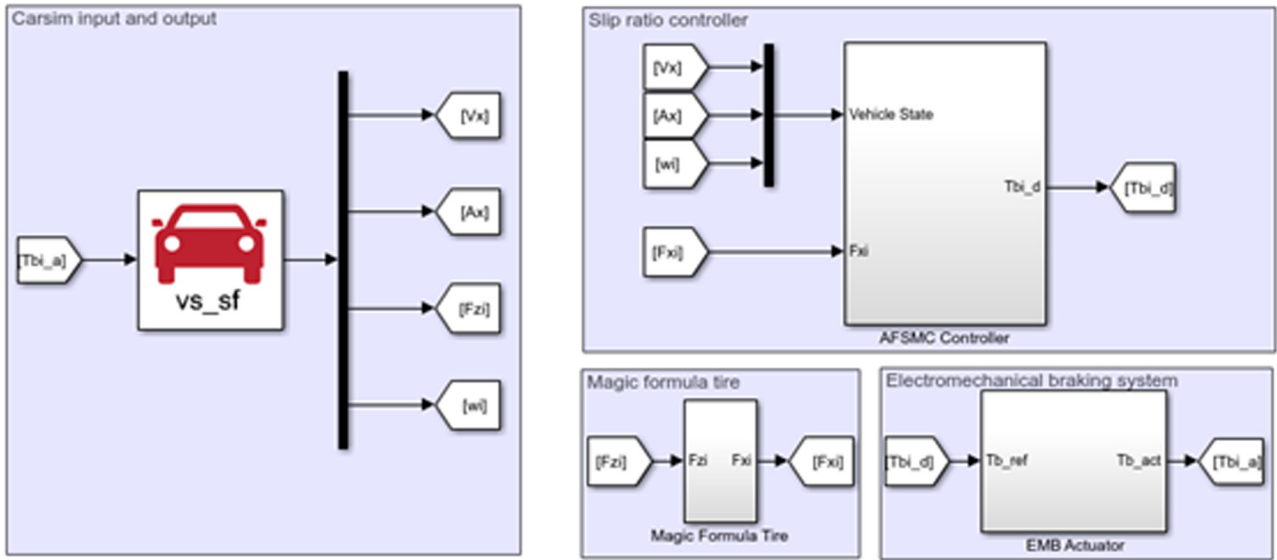


Fig. 9. Schematic structure of the simulation.

Table 3. Main parameters of the EMB system.

Parameters	Notation	Unit	Value
Stator resistance	R_m	Ω	5.55
Stator inductance	L	H	0.00495
EMF coefficient	K_e	V/s/rad	1.294
Torque coefficient	K_t	(N.m)/A	1.177
Rotary inertia	J	Kg/m ²	0.00311
Gear ratio	i_p		5
Ball screw ball	L_0	mm	2
Stiffness coefficient	K_s	N/mm	43000
Rated speed	ω	rpm	350

and the frequency is 3.5 Hz. The response contours for motor current and motor speed are depicted in Figures 10a and 10b, respectively.

Figure 10 demonstrates an excellent tracking effect, especially in terms of speed. The current exhibits some fluctuation in the initial phase of the simulation but quickly and steadily tracks the expected value. The initial rapid rise is attributed to the motor's commencement, wherein the motor speed rapidly increases to the rated speed, causing a corresponding rise in current.

It can be seen that Figure 11 shows the brake clamping force response curve. In the early stages of increasing brake clamping force, the desired clamping force may not be tracked well due to the ongoing elimination of the gap between the brake disk and the friction pad. Concurrently, as depicted in Figure 10b, the motor speed has reached its maximum rated value, influencing the tracking effect. However, after the target brake clamping force enters a cyclic change, the tracking effect becomes satisfactory, meeting the requirements for engineering applications.

4.2 Simulation analysis of slip ratio control

In order to verify the hierarchical control strategy proposed in this paper for controlling the slip ratio, three straight roads (high adhesion coefficient ($\mu = 0.85$), low adhesion coefficient ($\mu = 0.2$), joint ($\mu = 0.8 - \mu = 0.5$)), and select the simulation results of the left front wheel and the left rear wheel of the vehicle for analysis. The vehicle model parameters established in Carsim are shown in Table 4.

4.2.1 Braking conditions on high-adhesion road

At an initial speed of 100 km/h, braking is conducted, and the simulation results for high-adhesion roads are shown in Figure 12. Figure 12a is the longitudinal speed curve, and it is evident that the rear tyre speed is more stable than the front wheel speed. Similarly, the slip ratio fluctuation in Figure 12b can also reflect the stability of the wheel speed. From the commencement of the simulation to approximately 0.1s, no braking force is generated because the braking system must overcome the braking gap. During the initial phase of deceleration, the wheel speed fluctuates significantly and reflects the jitter of the slip ratio.

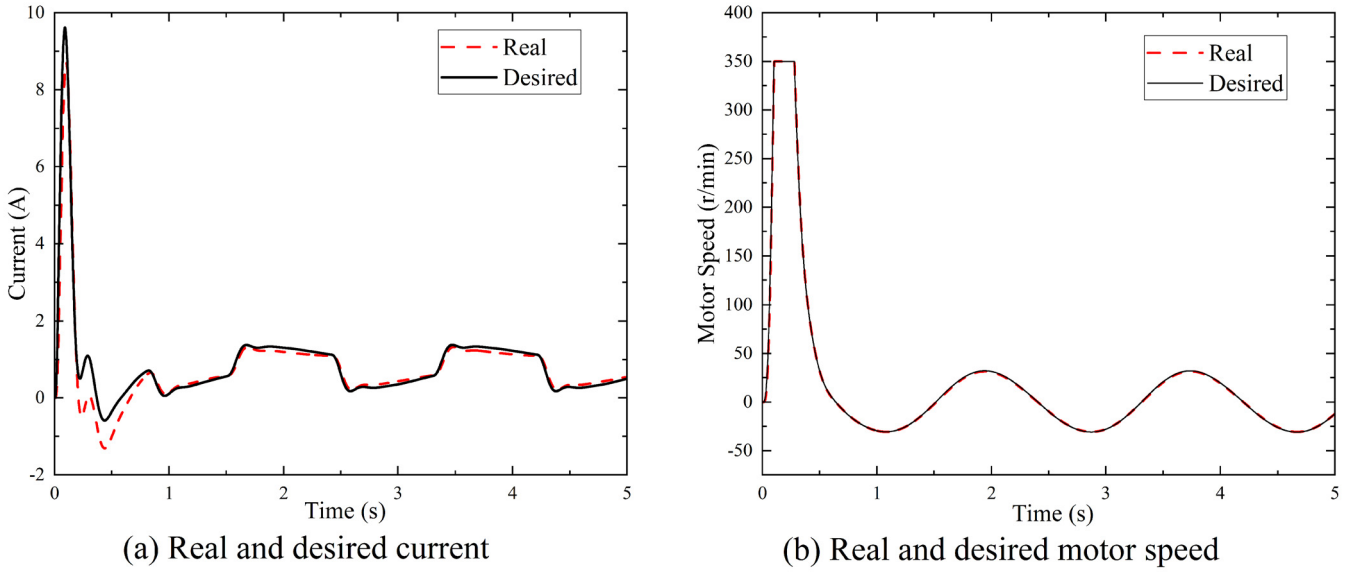


Fig. 10. Current and motor speed.

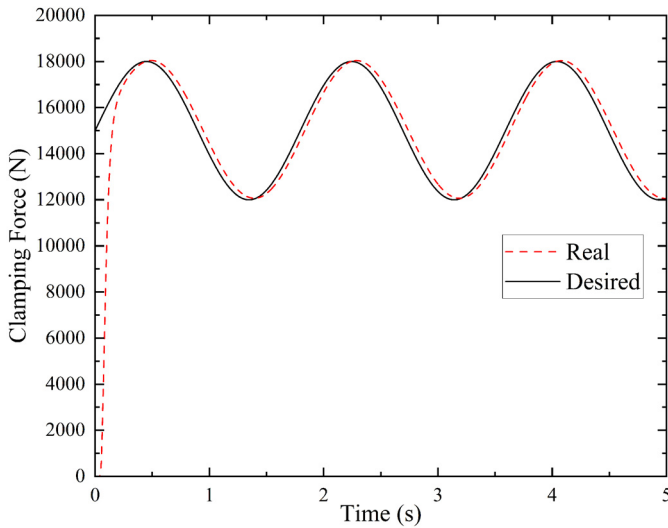


Fig. 11. Actual and desired clamping force.

Figure 13 compares the expected and actual braking torques for the FL wheel and the RL wheel, highlighting the characteristics of the front and rear axles. While the braking torque generally aligns with the expected torque, inherent execution characteristics of EMB and the influence of the clamping force control algorithm introduce some error. The tracking error for FL wheels is smaller than that for RL wheels, resulting in a more effective tracking effect with FL wheels. These characteristics are mirrored on the right wheel. After the vehicle comes to a complete stop, with both wheel speed and vehicle speed at 0, the effective slip ratio cannot be calculated, leading to fluctuations in the target braking torque. As evident in Figures 12 and 13, AFSMC exhibits smaller jitter and steady-state error compared to SMC, signifying its importance for practical applications.

4.2.2 Braking conditions on low-adhesion road

Considering the imperative need for safety in real-world driving scenarios, the vehicle is simulated on a low-adhesion road surface with an initial speed of 75 km/h. Figure 14 illustrates the simulation results on the low-adhesion road surface. Figure 14a portrays the longitudinal vehicle speed and wheel speed curve, while Figure 14b displays the slip ratio curve. In comparison to high adhesion, it is evident that speed jitter is more pronounced, but it manages to stabilize in the lower speed range, reflecting the control effectiveness in slip ratio.

Figure 15 showcases the braking torque control effect curve for the FL wheel and the RL wheel. The jitter in FL wheels is noticeably more pronounced than in RL wheels. This disparity is due to the fact that rear wheel lock is an extremely perilous operating condition. Consequently, restrictions are imposed during algorithm design to avert this situation. Furthermore, the superior control effectiveness of AFSMC is more evident in these conditions.

4.2.3 Braking conditions on joint road

To further substantiate the robustness of the adaptive fuzzy sliding mode control algorithm, a simulation test was conducted on a mixed road surface. Figure 16 illustrates the simulation results on the mixed road surface at an initial speed of 100 km/h, with the road surface adhesion coefficient changing from 0.8 to 0.2 around 1.2 seconds. Figure 16a depicts the longitudinal vehicle speed and wheel speed curve, while Figure 16b displays the slip ratio curve. The wheel speed exhibits pronounced jitter near the moment of the road surface adhesion coefficient change, as evident in the slip ratio control effectiveness.

Figure 17 presents the braking torque control effect curve for the FL wheel and the RL wheel. Due to the abrupt reduction in the road surface adhesion coefficient, the braking torque on the original road surface is excessive, and

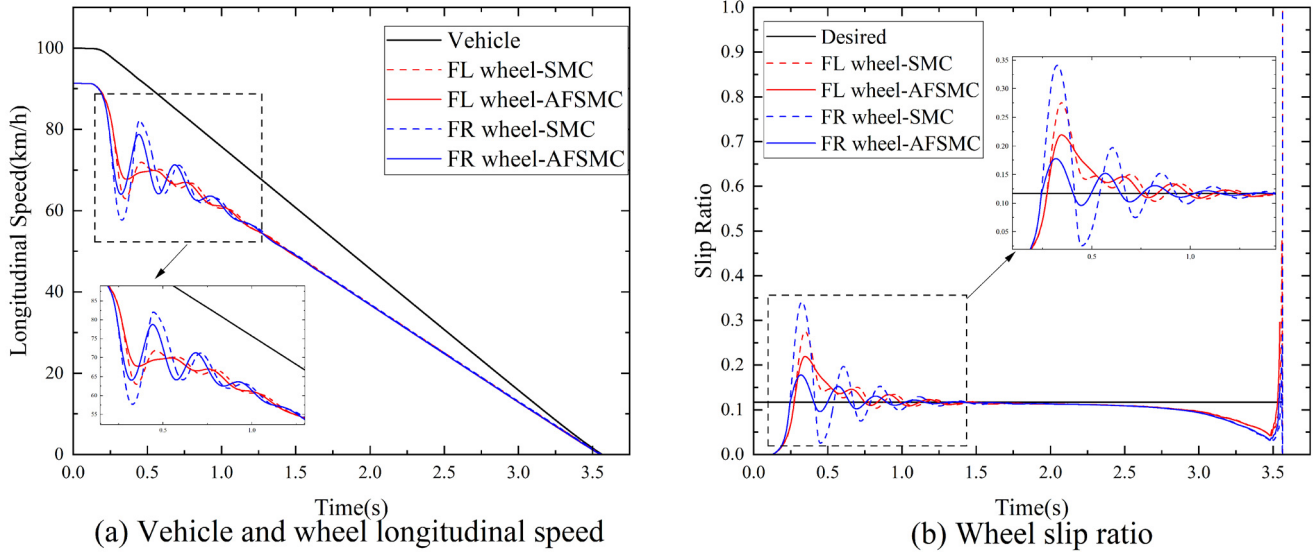


Fig. 12. Results in high adhesion road (longitudinal speed and slip ratio).

Table 4. Main parameters of the vehicle.

Parameters	Notation	Unit	Value
Vehicle mass	m	kg	1370
Distance from the center of mass to the front axle	a	m	1.26
Distance from the center of mass to the rear axle	b	m	1.38
Height of the center of mass	h_g	m	0.48
Effective radius of the tyre	R_w	m	0.304
Vehicle moment of inertia around Z-axis	I_z	kg.m ²	1343.1
Tyre moment of inertia	J_i	kg.m ²	2.1

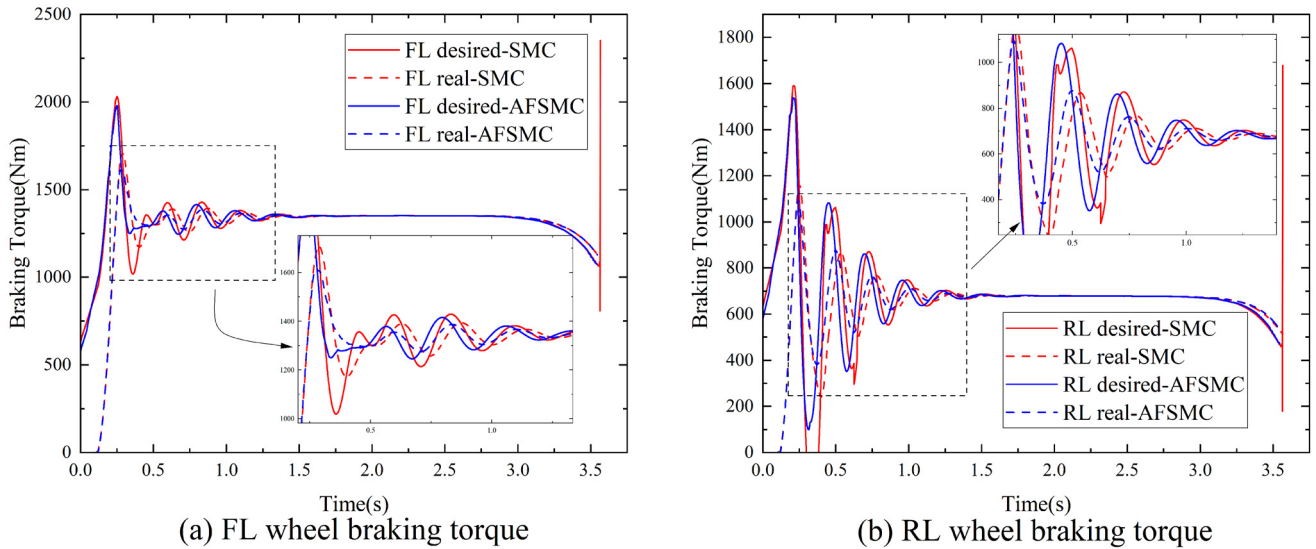


Fig. 13. Results in high adhesion road (braking torque).

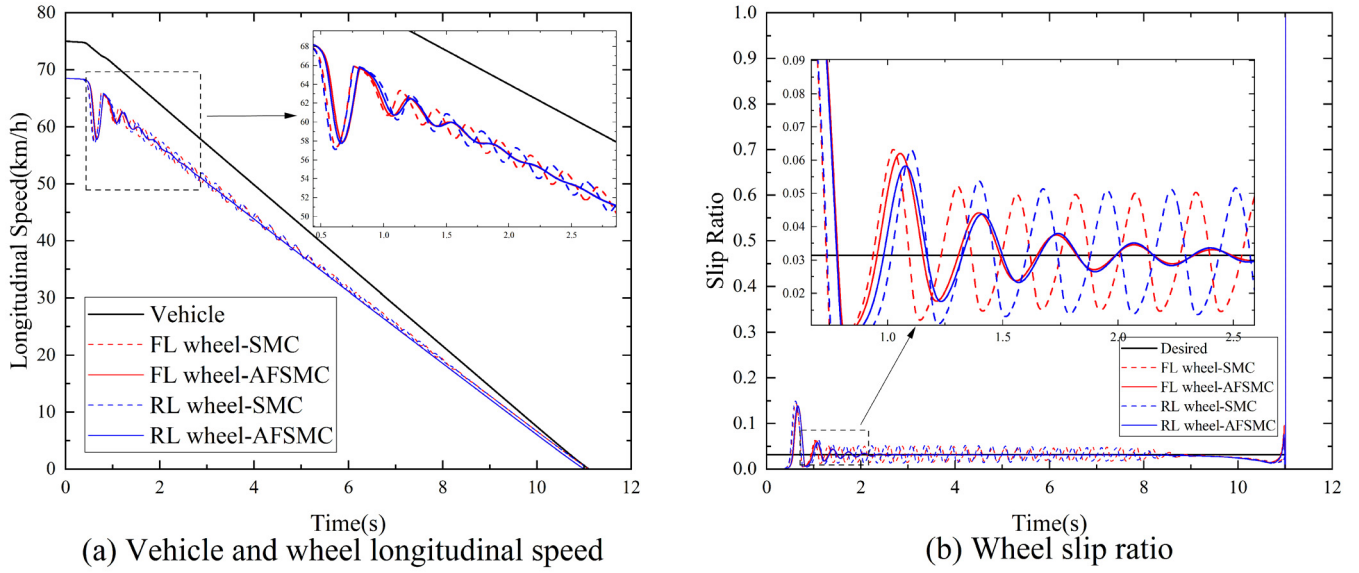


Fig. 14. Results in low adhesion road (longitudinal speed and slip ratio).

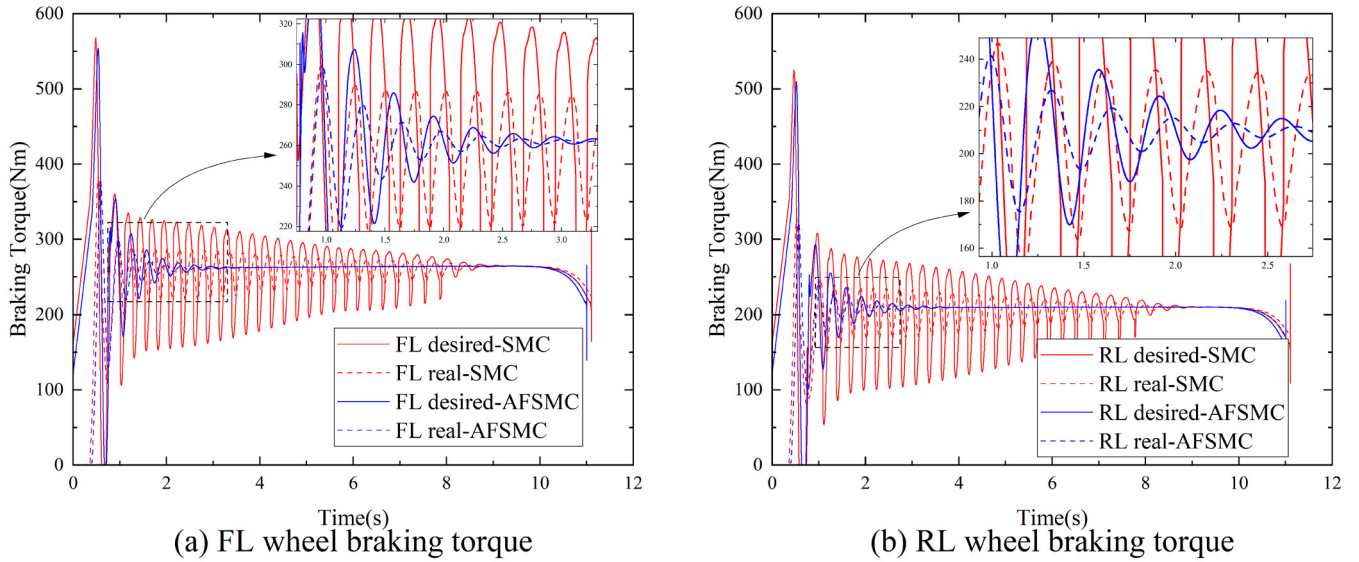


Fig. 15. Results in low adhesion road (braking torque).

the slip ratio cannot be maintained on the low adhesion coefficient road surface. This results in a rapid decrease in wheel speed and a reduction in vehicle deceleration during the road surface transition. As the speed of the front wheels decreases, braking torque is also diminished to maintain the desired slip ratio. In comparison to standard SMC control, AFSMC control exhibits less braking torque jitter, leading to reduced vehicle speed fluctuation, ultimately manifesting in improved slip ratio control effectiveness.

4.3 Results discussion

To provide a quantitative assessment of the derived conclusions, a comparative analysis of error magnitudes between the two controllers is conducted, accompanied by

an evaluation of the parameter braking distance. The results are depicted in Figures 18–20. We use the Root Mean Square Error (RMSE) metric for further discussion, combined with the maximum error “Max” and the braking distance “d”, in order to demonstrate the superiority of the proposed control strategy more intuitively. The specific comparisons are shown in Table 5.

As shown in Table 5, although the braking distance performance is not the best under a certain operating condition, for example (on a low traction road). However, this is the braking control with the objective of controlling the optimal slip ratio of the vehicle wheels, which is one of the optimal solutions to ensure the safety of the vehicle while taking into account the braking comfort.

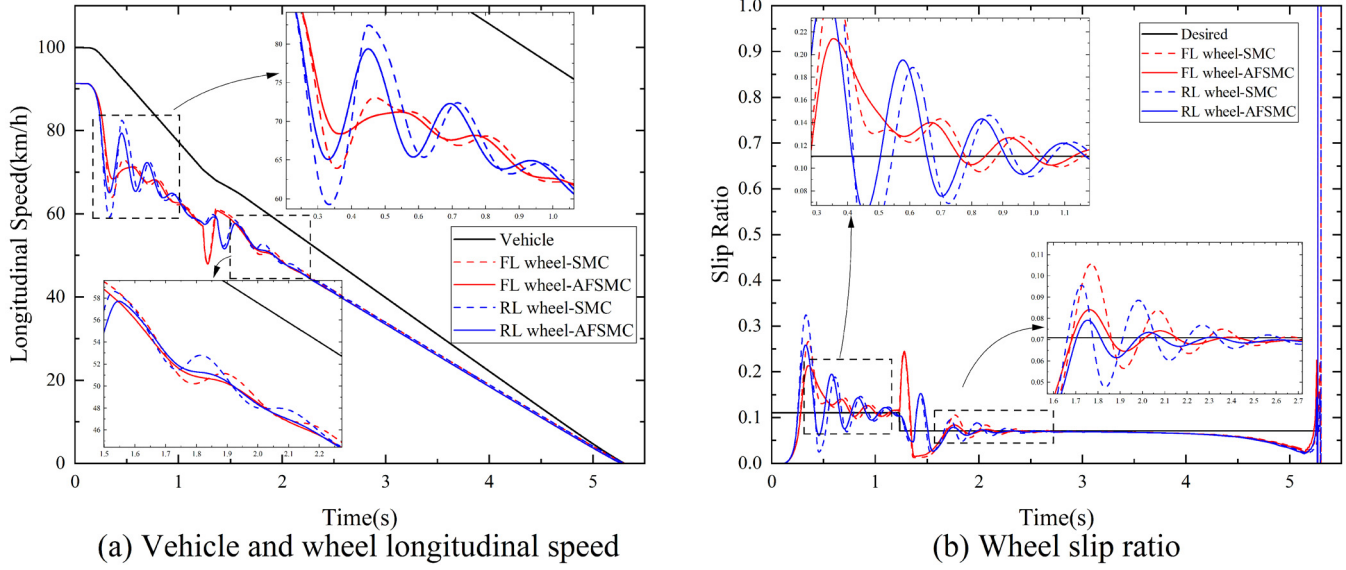


Fig. 16. Results in joint road (longitudinal speed and Wheel slip ratio).

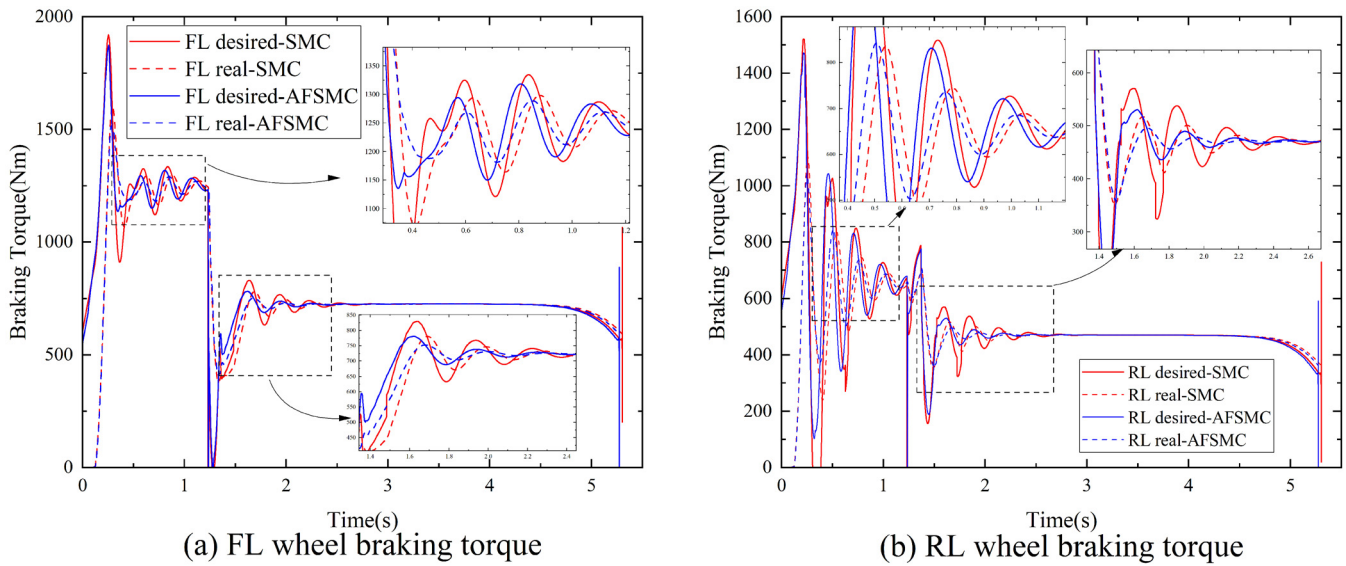


Fig. 17. Results in joint road (braking torque).

Figures 18–20 reveal that the AFSMC proposed in this paper exhibits significantly lower error levels under high adhesion conditions, resulting in a 10.79% reduction in braking distance. While the improvement may not be particularly pronounced during the initial stages of control under extremely low adhesion conditions, the stabilization occurs more rapidly, contributing to a substantial overall enhancement, with a 10.76% reduction in braking distance. In the case of the mixed road surface, the proposed AFSMC demonstrates minimal overshoot and error, notably outperforming the comparison under extremely low adhesion conditions, resulting in a 12.99% reduction in braking distance.

5 Conclusions

Initially, the paper establishes the longitudinal motion model for the vehicle and the mathematical model for the planetary gear EMB. Subsequently, the expected slip ratio is computed based on the Burckhardt tyre model. Finally, an adaptive fuzzy sliding mode controller is devised to regulate the wheel slip ratio by managing the braking torque across all four wheels. The braking torque is generated through the EMB actuator, and a cascaded PI clamping force control algorithm is specifically crafted for this task.

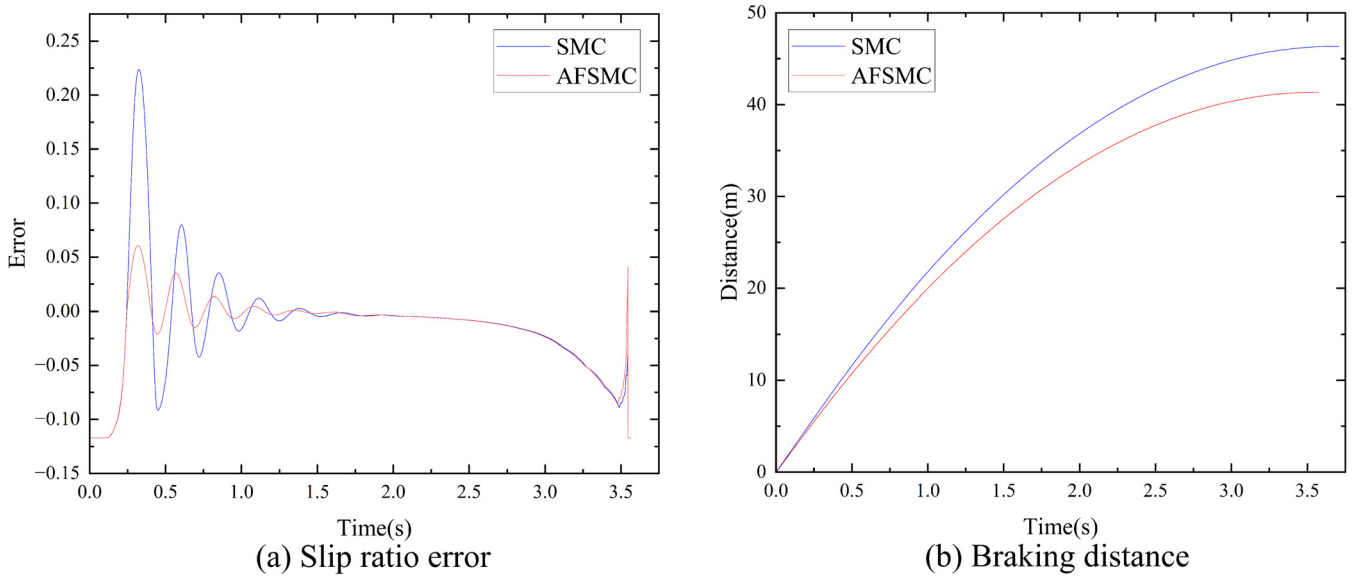


Fig. 18. Results in high adhesion road.

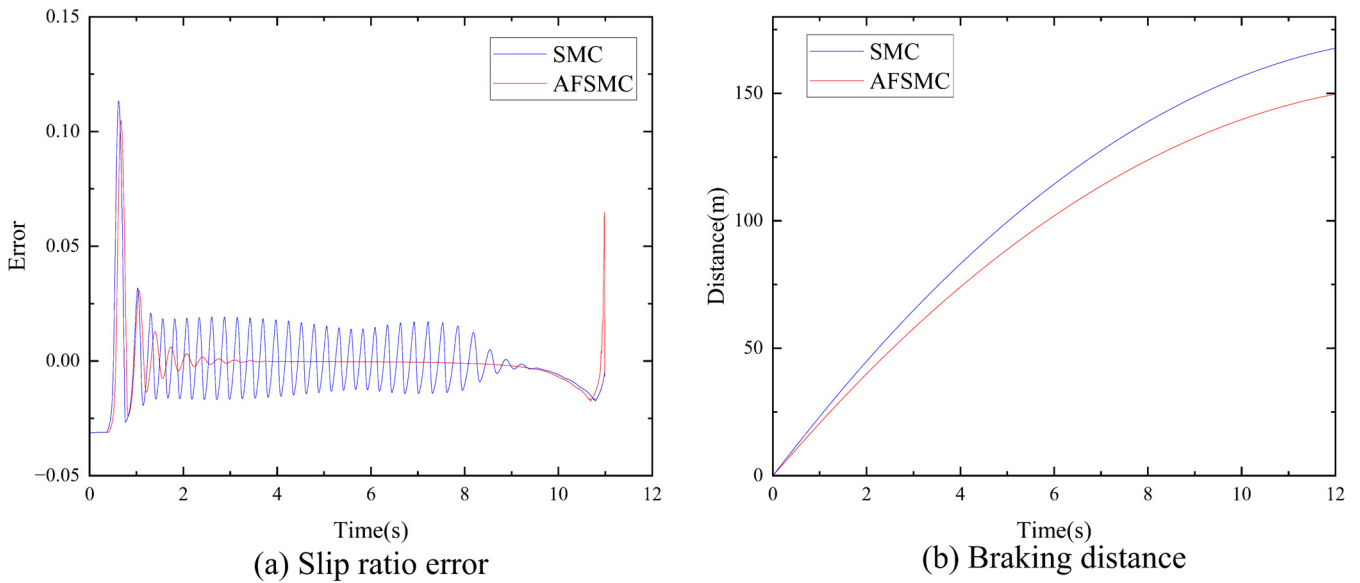
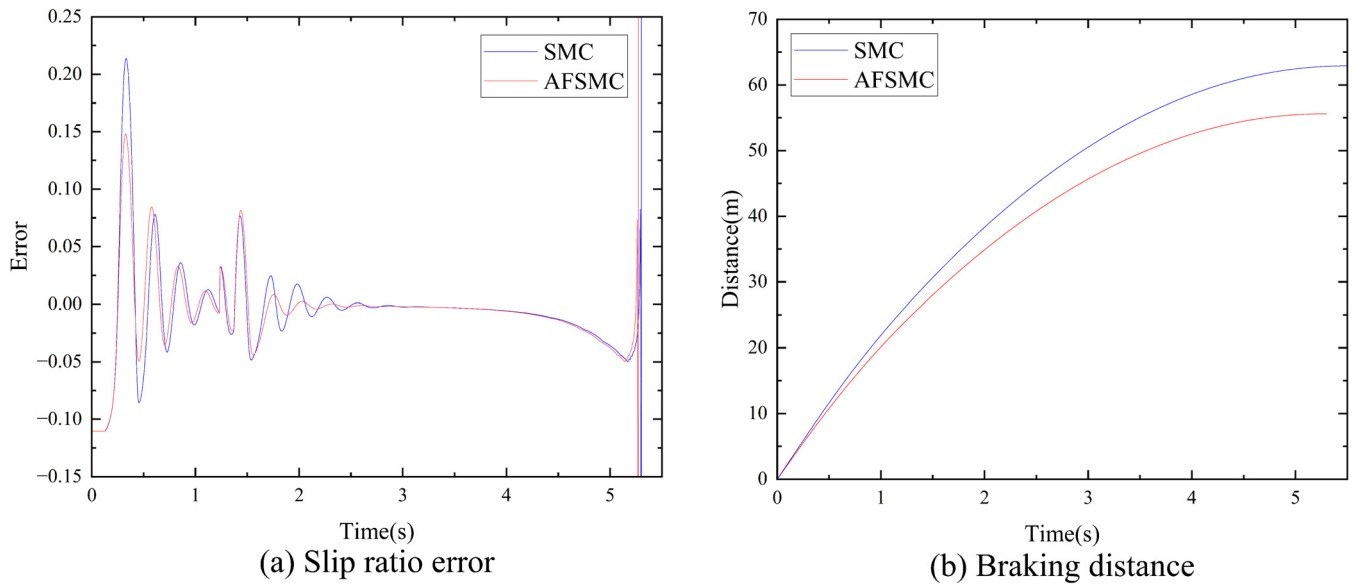


Fig. 19. Results in low adhesion road.

The simulation test results and the control process analysis presented in this paper reveal that traditional SMC is highly sensitive to disturbances in the arrival sliding mode, leading to frequent clattering in the control process. In contrast, the AFSMC proposed in this paper addresses this issue by making the discrete sign function continuous. This is achieved through the design of a fuzzy system that continuously approaches the switching item with a variable gain, mitigating some of the challenges associated with conventional SMC. Upon scrutinizing the control effects, it becomes evident that the performance of the conventional

SMC is inferior to that of the AFSMC control method advocated in this paper. This discrepancy arises from the fact that the traditional SMC control method lacks adaptability once parameters are set, failing to ensure optimal control effects across all conditions. In contrast, the AFSMC, with its variable switching gain determined by fuzzy rules, can dynamically adjust parameters based on the system's relative position, sliding surface, and motion state. This continuous adaptation ensures an optimal control effect under various working conditions, contributing to the stability and convergence of the entire system.

**Fig. 20.** Results in joint road.**Table 5.** Performance comparison of algorithms.

	High adhesion road			Low adhesion road			Joint road		
	Max	RMSE	d(m)	Max	RMSE	d(m)	Max	RMSE	d(m)
SMC	0.23	0.053	46.35	0.14	0.027	167.71	0.21	0.050	63.91
AFSCM	0.06	0.037	41.35	0.13	0.014	149.66	0.15	0.045	55.61

Nevertheless, due to time and resource constraints, this paper confines its analysis to simulation tests on conventional roads without real vehicle testing or comparison with alternative control algorithms. Future endeavors will involve real vehicle tests on diverse road types, offering a comprehensive comparison with other algorithms.

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Conflict of interest

The authors declare no potential conflicts of interest with respect to the research, authorship, and publication of this article.

Data availability statement

The data presented in this study are available on request from the corresponding author.

Author contribution statement

Conceptualization H.Z. and C.Z., methodology H.Z., software, simulation, analysis C.Z. and L.X., validation H.Z., C.Z., and Z.J., writing C.Z.

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