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RESEARCH ARTICLE

Research on Active Rear-Wheel Steering Control Method With Sliding Mode Control Optimized by Model Predictive

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ABSTRACT At present, the rapid development of electric vehicles and the innovation of related technologies have made the active steering control technology of electric vehicles a hot research topic. In this paper, a rear-wheel active steering control strategy based on sliding mode predictive control algorithm was proposed to improve the handling stability and active safety of electric vehicles. Among them, for the jitter problem of the sliding mode control algorithm this paper corrects and optimizes the control rate of the sliding mode control algorithm by adopting the ideas of feedback correction and rolling optimization in the model predictive control algorithm to suppress the jitter problem of the sliding mode control algorithm. At the same time, joint Carsim/Simulink simulation experiments and hardware-in-the-loop experiments were completed under different working conditions, and the simulation results were compared and analyzed with those of front-wheel steering vehicles with the same parameters and active rear-wheel steering vehicles with the sliding mode control algorithm. According to the experimental results, the sliding mode control algorithm optimized based on the model prediction algorithm proposed in this paper reduces the steady-state error of the vehicle yaw rate and the overshoot of the yaw rate by 30.012% and 18.103%, respectively, compared with the sliding mode control algorithm, and the output of the controller eliminates its jitter. The results showed that the proposed sliding mode predictive control algorithm had better control accuracy, better transient response characteristics, and smoother control output. Compared with the front-wheel steering vehicle, the ideal value yaw rate and target trajectory were accurately tracked, and the deviations of yaw rate and sideslip angle were reduced by 28.324% and 68.517%, respectively. The results showed that the proposed active rear-wheel steering control strategy provided better high-speed handling stability and active safety for the vehicle.

INDEX TERMS Electric vehicle, active rear-wheel steering, stability control, sliding mode predictive control, hardware-in-the-loop.

I. INTRODUCTION

The latest figures suggest that global sales of electric vehicles will account for up to 10% of all new sales of fuel, hybrid, and pure electric vehicles in 2022, with sales increasing by 68% year on year and continuing to do so over the next several years [1], [2], [3], [4]. As electric vehicle sales have increased in the global market, it has also driven the continuous innovation of electric vehicle-related technologies, and the control mode of vehicles has gradually changed from passive control to advanced active control and intelligent control [5], [6], [7]. Active control of the vehicle chassis is the core technology that guarantees high lift, high mobility, and reliability of the vehicle, accounting for more than 70%

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of the vehicle development cost. As an important part of the active control of vehicle chassis, the active steering system plays an important role in improving the vehicle's handling stability and active safety. At present, the active steering control technology of electric vehicles has emerged as a research hotspot.

Active steering technology for electric vehicles can be divided into active front-wheel steering (AFS), active four-wheel steering (A4WS), and active rear-wheel steering (ARS) according to the different control methods. AFS improves the steering performance, handling stability, and active safety of the vehicle by modifying the front wheel steering system with additional steering angles to help drivers who lack driving experience [8]. S. A. Saruchi et al. proposed a composite nonlinear feedback controller applied to yaw rate tracking control for the AFS of a vehicle equipped with steer-by-wire. A fast-tracking response of the yaw rate is achieved according to the desired response, and a corrected steering angle input is generated to improve vehicle maneuverability [9]. Jin et al. proposed a nonlinear robust H-infinity control strategy for improving the trajectory tracking performance of an autonomous ground electric vehicle with an AFS system [10]. Jing et al. proposed a novel integrated yaw stability and energy efficiency control strategy based on model predictive control and AFS system to reduce the large additional yaw moment of the vehicle under high-speed turning conditions to achieve vehicle stability improvement [11]. Ma et al. proposed a new Takagi-Sugeno fuzzy based sliding mode control (SMC) strategy for the AFS system to improve the cornering stability of vehicles [12]. Li et al. designed an active front steering sliding mode controller based on ideal variable transmission ratio to improve the maneuverability and stability of the vehicle [13]. Zhang et al. proposed a novel AFS control strategy in which the upper controller used adaptive recursive integral terminal sliding mode control to ensure the convergence performance of the actual side slip angle and yaw rate with strong robustness and fast convergence rate to improve the yaw stability and maneuverability of the steer-by-wire vehicle [14]. Ahmadian et al. proposed a multi-input multi-output integrated adaptive control method for AFS and direct yaw moment control to manage the variation of vehicle mass and tire-road friction coefficient as parameter uncertainties, resulting in improved vehicle stability improved tracking performance in terms of vehicle stability [15]. It can be seen that the core lies in the design of the control algorithm, commonly used control methods mainly include feedback control, robust control, model predictive control, fuzzy control, sliding mode control, and other methods. However, the additional steering angle of AFS will affect the direction of the vehicle, thus invading the driver's behavior. A4WS can improve the handling stability and active safety of the vehicle by actively adding additional steering angles to the steering angles of the front and rear-wheels to track the ideal value of the side slip angle. The common control methods include adaptive fuzzy control [16], model predictive control [17], and sliding mode control [18]. However, due to its high cost, the A4WS system is rarely used in practical engineering, and there is a problem of insufficient steering sensitivity in high-speed A4WS. ARS actively adjusts the rear-wheel steering angle according to the vehicle motion state, so that the vehicle's yaw rate and side slip angle can track its ideal value. It improves the steering sensitivity to a certain extent and can balance the active safety and stability at high speed so as to improve the operation stability and active safety of the vehicle without interfering with the driver's behavior. It can be concluded that ARS is more suitable for the active control of vehicles due to its unique characteristics, then improve and enhance the handling stability and active safety of vehicles.

Early research on ARS control mainly relied on classical control theory for algorithm design and usually used the proportional control method in which the front and rear-wheel turning angles are in constant proportion [19], [20]. However, this method changes the driver's driving habits to a large extent in order to reduce the yaw rate and side slip angle of the vehicle at high-speed steering. With the innovation and development of modern control theory, more and more adaptive and robust control methods have been applied to the active control of vehicles in order to improve and enhance the vibration performance, active safety, and handling stability of vehicles. X.J. Jin et al. proposed a robust vibration controller design for an in-wheel motor-driven electric vehicle active suspension system based on a unified μ-synthesis framework to effectively attenuate negative vibrations of the active suspension [21]. Jin proposed a robust finite-frequency H∞ control strategy with an in-wheel motor active suspension system to improve the vibration performance and ride comfort of electric vehicles [22]. For the study of active rear-wheel steering, Zhang et al. proposed a new pulsed active steering control system to improve the transverse sway stability of the vehicle [23], but since the rear wheel response is manipulated by sending pulsed signals, the yaw rate and the side slip angle of the vehicle fluctuate at a certain frequency. Sahin and Akalin proposed two different active rear-wheel steering controllers: a fuzzy logic controller and a linear model-based predictive controller to reduce the yawing behavior of articulated vehicles [24], but the design of the fuzzy controller lacks systematicity to define the control objectives. Moataz et al. designed a controller based on H∞ control using a linearized bicycle model to achieve active steering of the rear axle of a multi-axis combat vehicle [25] because robust H∞ control is concerned with minimizing the gain in the worst case, which may not be the "worst case" in practice. Changoski et al. designed several cooperative controllers for ARS and AFS system based on sliding mode control theory using the non-linear two-degree-of-freedom (2-DOF) vehicle bicycle model and implemented its control algorithm in MATLAB / Simulink to complete the simulation experiment verification of vehicle stability control and integrated control system [26]. Yuan et al. designed the active rear-wheel steering controller with the control rate of the sliding mode variable structure based on the linear vehicle model to improve the steering performance and stability of the vehicle [27]. The controller designed using different control algorithms has different characteristics, among which the SMC algorithm has become the preferred control algorithm for active rear-wheel steering controllers due to its good fast response characteristics and robustness, as well as its strong immunity to changes in the system itself corresponding to parameters and external disturbances. However, in practice, the SMC algorithm can cause the control system to jitter, which can place great requirements on the hardware.

This paper proposed an active rear steering control strategy based on the sliding model predictive control algorithm to address the above issues. The main contribution of this paper is to correct and optimize the control rate of the traditional SMC algorithm by adopting the ideas of feedback correction and rolling optimization in the model predictive control algorithm, which suppresses the jitter phenomenon existing in the algorithm on the basis of maintaining the fast response and anti-interference ability of its algorithm, i.e., the controller output is smooth, thus improving the vehicle handling stability and active safety. The jitter handling method of the sliding mode control algorithm proposed in this paper has the characteristics of fast convergence error of the sliding mode control algorithm, strong anti-interference ability and insensitivity to model parameter changes, and the optimal control characteristics of the model prediction. Therefore, it has the characteristics of fast convergence, strong anti-interference ability, and low model accuracy requirement compared with the current jitter elimination methods. Firstly, this paper established a 2-DOF vehicle dynamics model for a four-wheel steering electric vehicle and obtained the ideal value of the vehicle's yaw rate and side slip angle based on the front-wheel steering angle, vehicle speed, and tire-ground friction coefficient. Secondly, a Sliding Mode Predictive Control (SMPC) active rear-wheel steering controller was designed based on the optimized sliding mode control of the model prediction algorithm to achieve a better anti-vibration effect. Finally, joint Carsim/Simulink simulation experiments and hardware-in-the-loop experiments were carried out under different operating conditions to verify the effectiveness and real-time performance of the designed control algorithm.

II. THE VEHICLE DYNAMICS MODEL AND THE IDEAL STEERING STATE OF THE VEHICLE

A. 2-DOF VEHICLE MODEL

At present, in the field of vehicle handling and stability control, a linear 2-DOF vehicle model can better describe the basic vehicle handling characteristics, and it can also simplify the calculation of the controller [28], [29], [30], [31]. Therefore, in order to facilitate the study of the handling and stability of four-wheel steering electric vehicles and the design of active rear-wheel steering controllers, a simplified

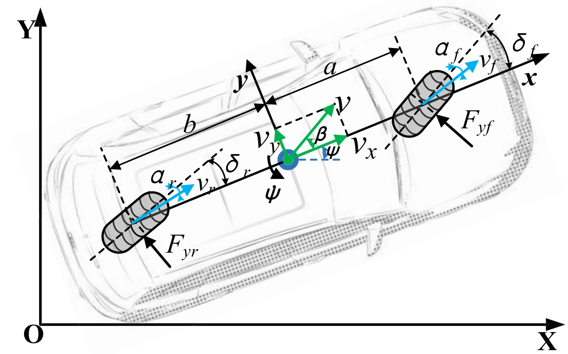

FIGURE 1. 2-DOF model of lateral vehicle dynamic.

2-DOF linear vehicle dynamics model was used in this paper, as shown in Fig. 1. The effects of roll, pitch, suspension, steering, drag and vertical dynamics were ignored in the analysis. The longitudinal velocity v_x of the vehicle was assumed to be constant, so only two degrees of freedom were considered, the lateral motion along the y-axis and the yaw motion of the body [14].

According to Newton's second law, the differential equation for the 2-DOF dynamics of an active rear-wheel steering vehicle is obtained as follows:

$$\begin{cases} ma_y = F_{yf}\cos\delta_f + F_{yr}\cos\delta_r \\ I_z\ddot{\varphi} = aF_{yf}\cos\delta_f - bF_{yr}\cos\delta_r \end{cases} \tag{1}$$

where m is the mass of the vehicle (kg), I_z is the rotational inertia of the vehicle about the vertical axis ($kg{\cdot}m^2$), a and b are the distances of the center of gravity of the vehicle from the front and rear axles, respectively (m), $\dot{\varphi}$ is the yaw rate of the vehicle (rad/s), F_{yf}f and F_{yr} rare the lateral forces of the tires corresponding to the front and rear wheels, respectively (N), $\boldsymbol{\delta}_f$ and $\boldsymbol{\delta}_r$ are the steering angles corresponding to the front and rear wheels, respectively (rad).

The lateral acceleration a_y of a vehicle, consisting of the acceleration component $\dot{v}_y$ and centripetal acceleration $v_x\dot{\varphi}$ of the vehicle acceleration along the y-axis can be described as:

$$a_y = \dot{v}_y + v_x\dot{\varphi} \tag{2}$$

When the lateral acceleration of the vehicle is limited to less than 0.4g and the cornering characteristic of the tire operates in a linear range, the lateral force of the tire corresponding to the front and rear wheels of the vehicle can be expressed as follows:

$$\begin{cases} F_{yf} = 2C_{\alpha f}\alpha_f \\ F_{yr} = 2C_{\alpha r}\alpha_r \end{cases} \tag{3}$$

where $\boldsymbol{C_{\alpha f}}$ and $\boldsymbol{C_{\alpha r}}$ are the tire side slip stiffness (N/rad) corresponding to the front and rear wheels, respectively, $\boldsymbol{\alpha}_f$ and $\boldsymbol{\alpha}_r$ are the tire slip angle (rad) corresponding to the front and rear wheels respectively. Among them, 2 indicates that there are two tires in the actual situation.

In view of the fact that $\boldsymbol{\delta_f}$ and $\boldsymbol{\delta_r}$ are small under high speeds conditions, the approximation gives: $\cos \boldsymbol{\delta_f} \approx \mathbf{1}$, $\cos \boldsymbol{\delta_r} \approx \mathbf{1}$, $\tan \boldsymbol{\beta} = v_y/v_x \approx \boldsymbol{\beta}$.

From the provisions of the coordinate system, the expressions for the tire slip angle corresponding to the front and rear wheels can be obtained:

$$\begin{cases} \alpha_f = \dfrac{a\dot{\varphi} + v_y}{v_x} - \delta_f = \dfrac{a\dot{\varphi}}{v_x} + \beta - \delta_f \\ \alpha_r = \dfrac{v_y - b\dot{\varphi}}{v_x} - \delta_r = \beta - \dfrac{b\dot{\varphi}}{v_x} - \delta_r \end{cases} \tag{4}$$

Given the system state variable is $X = \left[\, \beta \ \dot{\varphi} \,\right]^T$, the output variable is $Y = \left[\, \beta \ \dot{\varphi} \,\right]^T$. To joint Eq. (1), (2), (3), and (4), it can be concluded that the linearized 2-DOF model of an active rear-wheel steering vehicle is:

$$\begin{aligned} \dot{X} &= AX + B_1 U + B_2 W \\ Y &= CX \end{aligned} \tag{5}$$

where $A = \begin{bmatrix} \frac{2C_{\alpha f}+2C_{\alpha r}}{mv_x} & \frac{2aC_{\alpha f}-2bC_{\alpha r}}{mv_x^2} - 1 \\ \frac{2aC_{\alpha f}-2bC_{\alpha r}}{I_z} & \frac{2a^2C_{\alpha f}+2b^2C_{\alpha r}}{I_z v_x} \end{bmatrix}$, $B_1 = \begin{bmatrix} \frac{-2C_{\alpha r}}{mv_x} & \frac{2bC_{\alpha r}}{I_z} \end{bmatrix}^T$, $B_2 = \begin{bmatrix} \frac{-2C_{\alpha f}}{mv_x} & \frac{-2aC_{\alpha f}}{I_z} \end{bmatrix}^T$, $U = [\delta_r]$, $W = [\delta_f]$, $C = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$, $X = \left[\, \beta \ \dot{\varphi} \,\right]^T$, $Y = \left[\, \beta \ \dot{\varphi} \,\right]^T$.

In practical control engineering applications, real-time sampling and calculation are required through the microprocessor, so before designing the controller, the continuous model of vehicle dynamics is discretized by the forward Euler method, and the equations of state space are:

$$\begin{aligned} x(k+1) &= \Theta x(k) + Gu(k) + Hw(k) \\ y(k) &= Jx(k) \end{aligned} \tag{6}$$

where $\Theta = I + T_s A$, $G = T_s B_1$, $H = T_s B_2$, T_s is the sampling period of the system, and k is the current time.

B. THE IDEAL STEERING STATE OF THE VEHICLE

Active rear-wheel steering vehicles must retain a similar steering handling driving experience to front-wheel steering vehicles, while the vehicle should have good maneuverability at low speeds, and the system can quickly reach a stable state at high speeds.

In this paper, the ideal value of the side slip angle is set to zero to better control the vehicle motion attitude, and the ideal value of the yaw rate is obtained from the front wheel steering angle, vehicle speed, and other vehicle parameters. At the same time, because the maximum lateral acceleration of the vehicle is limited by the friction coefficient μ between the tire and the ground, the ideal value of the yaw rate should also be satisfied: $\dot{\varphi}_{bound} \leq 0.85\mu g/v_x$ [32]. The ideal values

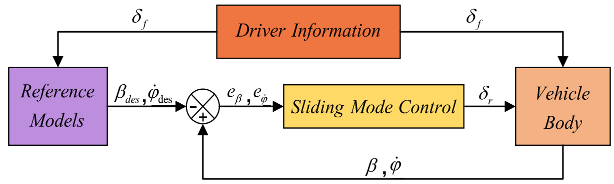

FIGURE 2. **Block diagram of the Sliding Mode Control system.**

of the yaw rate and the side slip angle are expressed as:

$$\begin{cases} \beta_{des} = 0 \\ \dot{\varphi}_{des} = \min\left(\left| \dfrac{v_x \delta_f}{L(1+Kv_x^2)} \right|, \left| \dfrac{0.85\mu g}{v_x} \right| \right) sgn(\delta_f) \end{cases} \tag{7}$$

where β_{des} is the ideal value of the side slip angle (rad), $\dot{\varphi}_{des}$ is the ideal value of the yaw rate (rad/s), $L = a+b$ is the vehicle axis distance (m), g is the acceleration of gravity ($\mathrm{m/s^2}$), sgn() is the sign function, and $K = \frac{m}{L^2}\left(\frac{a}{2C_{\alpha r}} - \frac{b}{2C_{\alpha f}}\right)$ is the stability factor ($\mathrm{s^2/m^2}$).

III. ACTIVE REAR-WHEEL STEERING CONTROL STRATEGY

A. OVERALL STRUCTURE OF THE ACTIVE REAR-WHEEL STEERING CONTROL STRATEGY

In this paper, considering the good transient response characteristics and robustness of the SMC algorithm, as well as the strong anti-interference ability of the system itself according to parameter changes and external disturbances, combined with the uncertainty of the vehicle system itself, this paper used the SMC algorithm to actively control the rear-wheel steering angle to achieve the tracking control of the yaw rate and the side slip angle, so as to achieve the purpose of improving vehicle stability and active safety, and the structural block diagram of its control strategy is shown in Fig. 2.

B. ACTIVE REAR-WHEEL STEERING CONTROLLER DESIGN

Using the discrete sliding mode variable structure control algorithm, it is firstly necessary to determine a sliding mode surface that satisfies the trajectory of the side slip angle and the trajectory of the yaw rate. Therefore, in this paper, the weighted combination of the deviation of the yaw rate and side slip angle is selected to define the sliding mode surface, and the sliding mode switching function at the kth moment is defined as:

$$s(k) = C_e[x(k) - R(k)] \tag{8}$$

where $C_e = [\,\xi \ 1\,]$, ξ is the weighting factor, $x(k) = [\,\beta(k) \ \dot{\varphi}(k)\,]^T$, $\beta(k)$ and $\dot{\varphi}(k)$ are the vehicle's the side slip angle and yaw rate at the kth moment, $R(k) = [\,\beta_{des}(k) \ \dot{\varphi}_{des}(k)\,]^T$, $\beta_{des}(k)$ and $\dot{\varphi}_{des}(k)$ are the vehicle's the ideal value of the side slip angle and the yaw rate at the kth moment, respectively.

The ideal value of the vehicle's side slip angle and the yaw rate at the $(k+1)$th moment is obtained by the linear extrapolation method as follows:

$$R(k+1) = 2R(k) - R(k-1) \tag{9}$$

where $R(k-1) = [\,\beta_{des}(k-1)\ \dot{\varphi}_{des}(k-1)\,]^T$, $\beta_{des}(k-1)$ and $\dot{\varphi}_{des}(k-1)$ are the vehicle's ideal value of the vehicle's the side slip angle and yaw rate of at the previous moment respectively.

According to the Eq. (6), (8), and (9), the sliding mode switching function at the $(k+1)$th moment can be calculated as follows:

$$s(k+1) = C_e[x(k+1) - R(k+1)] = C_e[\Theta x(k) + Gu(k) + Hw(k) - 2R(k) + R(k-1)] \tag{10}$$

In order to ensure that the side slip angle and yaw rate can move from any initial state to the sliding mode surface and have good motion performance and error range, this paper uses the discrete time exponential reaching law as the discrete sliding mode reaching law [33], that is:

$$s(k+1) = -\varepsilon T_s sgn[s(k)] - (qT_s - 1)s(k) \tag{11}$$

where ε is the constant velocity approach parameter and $\varepsilon > 0$, q is the approach velocity parameter and $q > 0$, $qT_s - 1 < 0$. T_s is the system sampling periods.

By making Eq. (10) and (11) equal, the sliding mode variable structure control rate $u(k)$ can be obtained, that is, the active rear-wheel steering angle can be described as follows:

$$u(k) = -(C_eG)^{-1}\{C_e\Theta x(k) + C_eHw(k) - 2C_eR(k) + C_eR(k-1) + (qT_s - 1)s(k) + \varepsilon T_s sgn[s(k)]\} \tag{12}$$

C. STABILITY ANALYSIS

Since the tracking error of the side slip angle and yaw rate tends to zero only when the vehicle is in ideal motion and the whole system is stable, it is necessary to analyze the stability of the active rear-wheel steering control system. According to the Lyapunov stability theorem [34], the chosen Lyapunov function can be described as follows:

$$V(k) = \frac{1}{2}s(k)^2 \tag{13}$$

Then the derivative of the *Lyapunov* function can be described as:

$$\dot{V} = \frac{V_{k+1} - V_k}{T_s} = \frac{s(k+1)^2 - s(k)^2}{2T_s} \tag{14}$$

Substituting Eq. (11) into Eq. (14) gives the following:

$$\begin{aligned}
&[s(k+1) - s(k)]sgn[s(k)] \\
&\quad = \{-\varepsilon T_s sgn[s(k)] - (qT_s - 1)s(k) - s(k)\}sgn[s(k)] \\
&\quad = \{-\varepsilon T_s sgn[s(k)] - qT_s s(k)\}sgn[s(k)] \\
&\quad = -\varepsilon T_s - qT_s\,|s(k)| < 0
\end{aligned} \tag{15}$$

$$\begin{aligned}
&[s(k+1) + s(k)]sgn[s(k)] \\
&\quad = \{-\varepsilon T_s sgn[s(k)] - (qT_s - 1)s(k) + s(k)\}sgn[s(k)] \\
&\quad = \{-\varepsilon T_s sgn[s(k)] - (qT_s - 2)s(k)\}sgn[s(k)] \\
&\quad = -\varepsilon T_s - (qT_s - 2)\,|s(k)|
\end{aligned} \tag{16}$$

When $|s(k)| > -\varepsilon T_s/(qT_s-2)$, $[s(k+1) + s(k)]\,\mathrm{sgn}[s(k)] > 0$ and $\dot{V} < 0$, it is easy to conclude that the active rear-wheel steering controller designed in this paper can ensure that the whole vehicle system is asymptotically stable and the system converges to the equilibrium point $s(k) = 0$. Combined with Eq. (8), it is easy to see that when $s(k)$ converges to 0, the tracking error of the side slip angle and yaw rate converges to zero, i.e., the side slip angle and yaw rate of the vehicle converges to the ideal value.

D. SLIDING MODE CONTROL RATE OPTIMIZATION

The SMC algorithm is characterized by fast convergence errors and high noise immunity. However, the discrete sliding mode convergence law used in the actual system is subject to jitter due to the existence of the symptom function, which affects the control effect of the controller. At present, the solution to the jitter problem of the sliding mode control algorithm mainly includes the continuous switching-based sliding mode control and super-spiral sliding mode control. The former is to reduce the jitter problem of sliding mode control by using a hyperbolic function instead of a discontinuous symbolic function, but it reduces the response speed while reducing the sliding mode control. The latter reduces the jitter problem of sliding mode control by using the integral to obtain the actual control quantity to eliminate the high-frequency switching quantity, but its introduction of integral calculation will greatly increase the computational effort and difficulty of controller design. Model predictive control as a special optimal control has the characteristics of prediction, optimization, and correction, so in this paper, in order to suppress the jitter phenomenon of the SMC algorithm, the feedback correction and rolling optimization ideas in the model predictive control algorithm are used to correct and optimize the control rate of the SMC algorithm, so as to suppress the jitter phenomenon of the SMC algorithm on the basis of maintaining its fast response and anti-interference capability. The structure schematic of the control algorithm is shown in Fig. 3.

From the state Eq. (6), (8), and (9), the sliding mode switching function prediction model can be calculated to predict at the current moment k to the predicted value $s(k+p|k)$ at the $(k+p)$th moment, which can be described as:

$$s(k+p|k) = C_eA^px(k) + \sum_{i=1}^{p} C_eA^{i-1}Bu(k+p-i|k) - C_e2^pR(k) + \sum_{i=1}^{p} C_e2^{i-1}R(k-p+i-1|k) \tag{17}$$

In the actual use of the active rear-wheel steering controller, the deviation between the actual value at the current moment and the predicted value in the past is used to correct the predicted value of the sliding mode switching function at the future moment in order to reduce the error

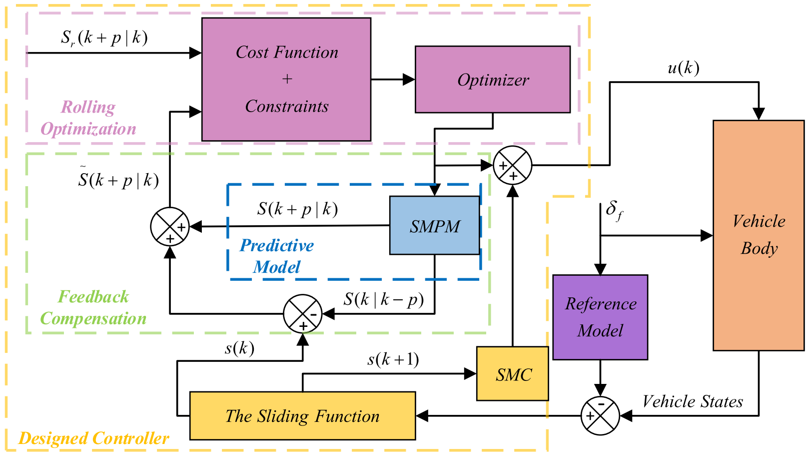

FIGURE 3. Schematic diagram of the structure of the SMP control algorithm.

in the prediction model of the sliding mode switching function due to possible disturbances outside the system. From Eq. (17), the predicted value $s(k|k-p)$ at the past $(k-p)$th moments for the kth moment can be calculated and described as:

$$\begin{aligned} s(k|k-p) = &\sum_{i=1}^{p} C_e A^{i-1} B u(k-i|k-p) \\ &+ C_e A^p x(k|k-p) - C_e 2^p R(k|k-p) \\ &+ \sum_{i=1}^{p} C_e 2^{i-1} R(k-2p+i-1|k-p) \end{aligned} \quad (18)$$

The predicted value of the sliding switching function after correcting the feedback is described as follows:

$$\hat{s}(k+p|k) = s(k+p|k) + h_p[s(k) - s(k|k-p)] \quad (19)$$

where h_p is the calibration parameter and $h_p \in \boldsymbol{R}$.

In this paper, the exponential rate of convergence, where the rate of convergence decreases from large, is chosen as the reference trajectory of the sliding mode expectation and described as follows:

$$\begin{aligned} s_r(k+p) = &-\varepsilon T_s sgn[s_r(k+p-1)] \\ &- (qT_s - 1)s_r(k+p-1) \end{aligned} \quad (20)$$

$$s_r(k) = s(k) \quad (21)$$

where $s_r(k+p)$ is the expected future reference value of the sliding mode switching function at the $(k+p)$th moment.

The optimization ideas in the model control algorithm are used to construct the performance objective function, which is described as follows:

$$\begin{aligned} &J = \sum_{i=1}^{p} q[\hat{s}(k+i|k) + s_r(k+i)]^2 + \sum_{j=1}^{c} r[u(k+j|k)^2] \\ &s.t.\ u_{\min} \le u(k+i) \le u_{\max}, \quad i = 0, 1, \cdots, c-1 \end{aligned} \quad (22)$$

where q and r are the error of the sliding mode switching function and the weight coefficient of the rear-wheel steering angle in the objective function, respectively. p and c are the prediction time domain and the control time domain, respectively.

In this paper, the performance objective function is transformed into a quadratic standard form and combined with constraints to derive the optimal control sequence by solving a standard quadratic programming problem, which is described as follows:

$$U(k) = [u^*(k), u^*(k+1), \cdots, u^*(k+c-1)]^T \quad (23)$$

The first element of the control sequence is selected as the control output value for the actual control, based on the principles of model predictive control.

IV. EXPERIMENTS AND DISCUSSION

After the design of the active rear-steering steering controller was completed, a simulation model of the active rear-wheel steering controller was built using MATLAB/Simulink to verify the controller's effectiveness and a joint simulation with the vehicle dynamics model in the commercial software Carsim was performed for experimental verification. Since the object of this paper is a four-wheel independent drive electric vehicle, the powertrain system of the vehicle model in Carsim needs to be modified and set to external input through the Powertrain Systems module firstly. Secondly, the steering system is set to four-wheel steering, and the rear-wheel steering angle is input externally. Finally, the Carsim interface is set up. The input and output of the Carsim vehicle model are one-to-one with the output input of the control strategy model in MATLAB/Simulink.

The feasibility of the active rear-wheel steering controller was verified by steering wheel corner step input and sinusoidal input simulation experiments, respectively. And the experimental results of the ARS vehicle using the SMC algorithm, the ARS vehicle using the LQR algorithm, the ARS vehicle with the SMPC algorithm, and the uncontrolled front wheel steering (FWS) vehicles were compared and analyzed.

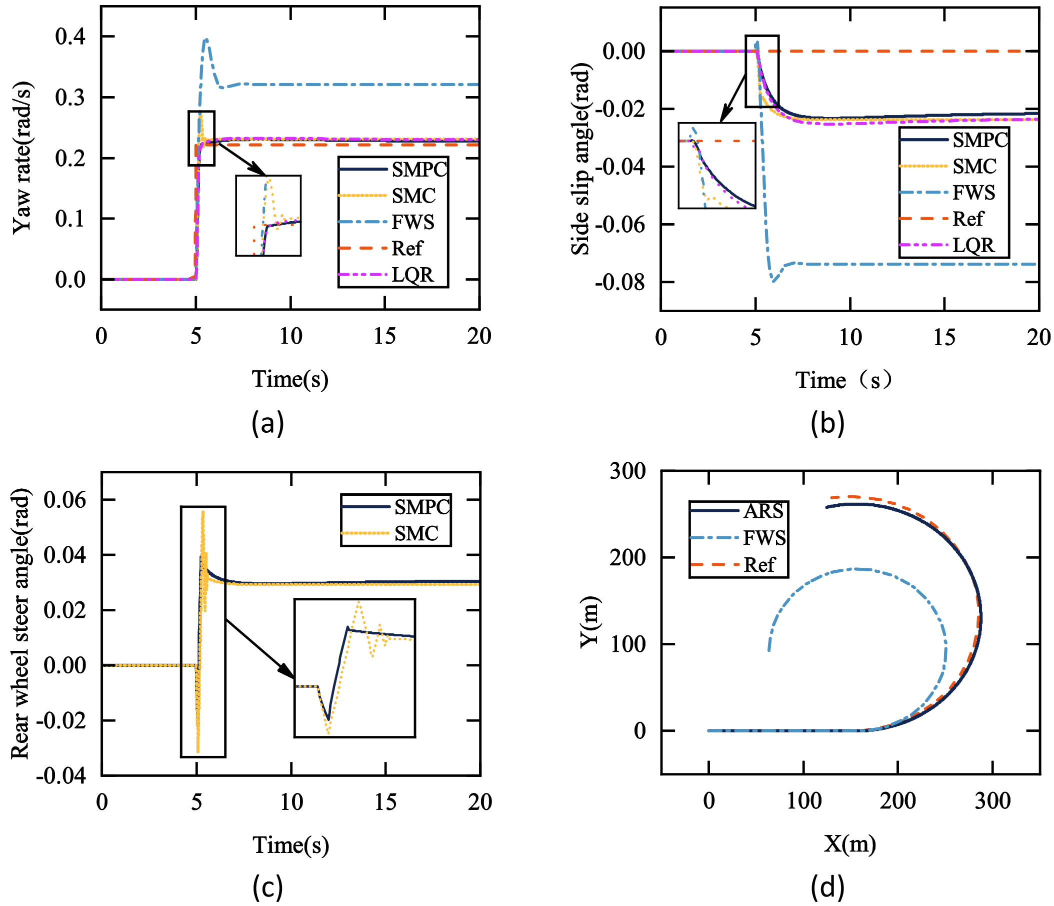

FIGURE 4. Comparison of simulation results under steering wheel step input conditions. Among them, (a) is the yaw rate comparison between ARS vehicles with different control algorithms and FWS vehicles; (b) is the side slip angle comparison between ARS vehicles with different control algorithms and FWS vehicles; (c) is the ARS controller output comparison between different control algorithms; (d) is the trajectory comparison between ARS vehicles and FWS.

TABLE 1. Parameters of the vehicle.

Symbol/Units	Value
m/kg	3018
I_z/kg•m^2	10437
a/m	1.84
b/m	1.88
L/m	3.72
$C_{\alpha f}$	-46328
$C_{\alpha r}$	-76690
g/(m/s^2)	9.8
q	10
r	200
p	10
c	2
T_s	0.01

All parameters of the vehicle and controller are described in TABLE 1.

In the simulation, the steering wheel steering input and sinusoidal input were simulated on a road surface with a coefficient of adhesion $\mu = 0.8$ at a speed of 30m/s to compare and analyze the control effect of the active rear-wheel steering controller using two different control algorithms.

A. STEERING WHEEL CORNER STEP INPUT SIMULATION EXPERIMENT

In order to analyze the transient response characteristics of the proposed active rear-wheel steering controller, simulation experiments were carried out to verify the steering wheel angle step input conditions. The steering system was ignored in this article, i.e., the front-wheel steering angle was equal to the angle at which the steering wheel would be turned. Given an initial value of zero for the steering wheel angle, the front-wheel steering angle reached $\boldsymbol{\pi/30\ rad}$ in 0.1s at the 5ths and then remained constant. The results of the simulation experiments are shown in Fig. 4. X and Y in Fig. 4(b) are the longitudinal displacement and transverse displacement of the vehicle respectively.

According to Fig. 4(a), 4(b), and 4(d), it can be seen that in the steering wheel corner step input simulation test, the vehicle with the ARS control system has 28.324% and 68.517% lower deviation values of the yaw rate and side slip angle at steady state compared to the conventional FWS vehicle, and the motion trajectory can track the ideal values very well. This indicates that the ARS vehicle is effective in reducing the risk of vehicle drift at high speeds, which

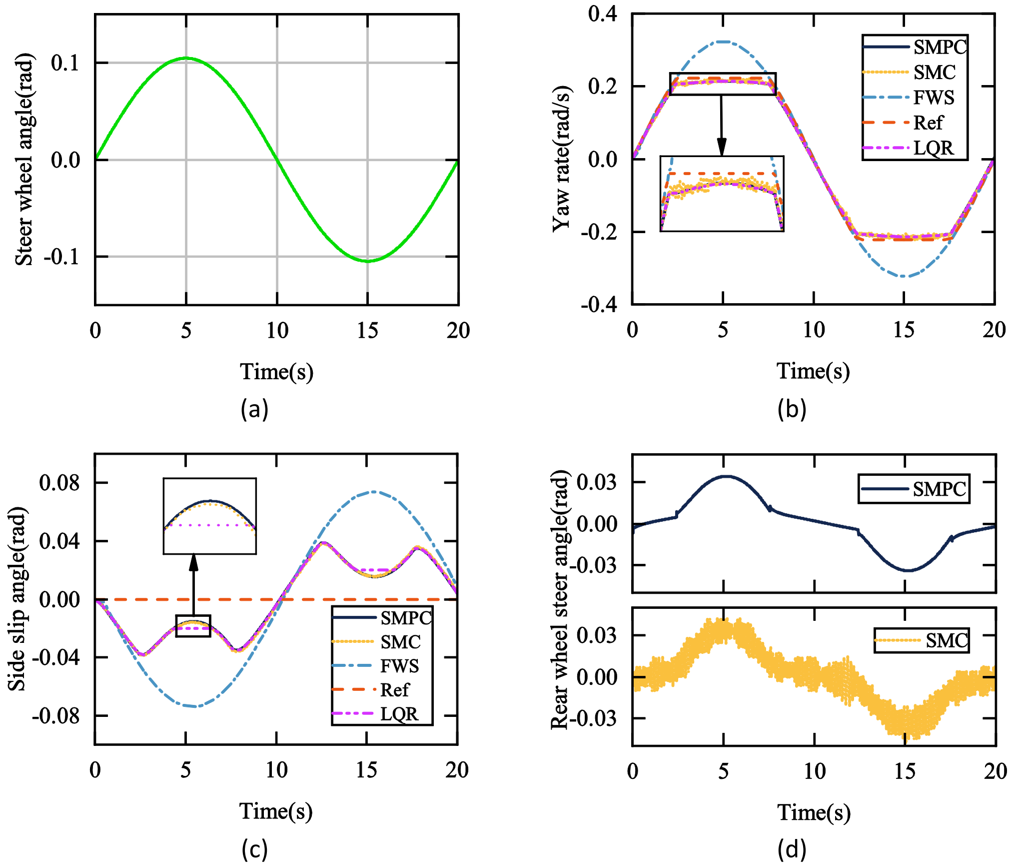

FIGURE 5. **Comparison of simulation results for steering wheel sinusoidal input conditions. Among them, (a) is the steering wheel angle signal map; (b) is the yaw rate comparison between ARS vehicles with different control algorithms and FWS vehicles; (c) is the side slip angle comparison between ARS vehicles with different control algorithms and FWS vehicles; (d) is the ARS controller output comparison between different control algorithms.**

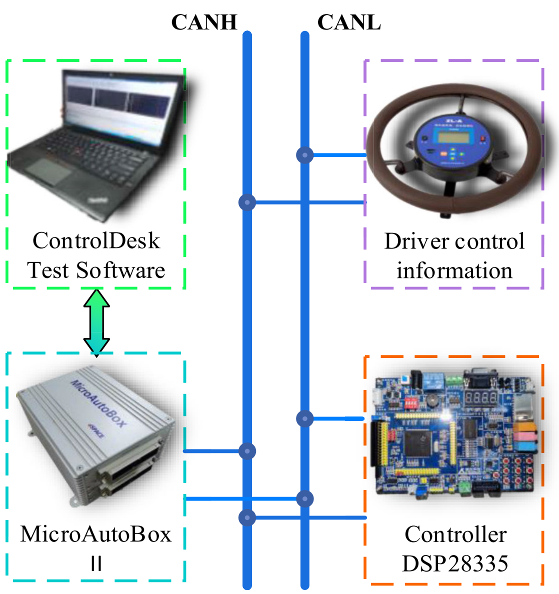

FIGURE 6. **Schematic diagram of the working process of the hardware-in-the-loop platform.**

in turn improves the handling stability of the vehicle at high speeds. According to Fig. 4(a), the active rear-wheel steering controller with the SMC algorithm has an overshoot of 18.103%. In contrast, the active rear-wheel steering controller with the SMPC algorithm has no overshoot and a short steady state arrival time, and it is able to control the maximum deviation to within 0.007 rad/s, a reduction of 30.012% compared to the SMC algorithm. At the same time, it is obvious that the active rear-wheel steering controller with the SMPC algorithm has a faster response time and better control effect than the active rear-wheel steering controller with the LQR algorithm. According to Fig. 4(b), the active rear-wheel steering controller with the SMPC algorithm has a smaller steady-state error than the active rear-wheel steering controller with the SMC algorithm and the active rear-wheel steering controller with the LQR algorithm in controlling the side slip angle. According to Fig. 4(c), it can be seen that the active rear-wheel steering controller using the SMC algorithm has a larger amount of overshoot compared to the SMPC algorithm, which seriously affects the performance of the controller in practical applications. On the contrary, the SMC algorithm is modified by using the feedback correction and rolling optimization ideas of the model predictive control algorithm to make the overshoot phenomenon well suppressed. In summary, the active rear-wheel steering controller using the SMPC algorithm was characterized by good

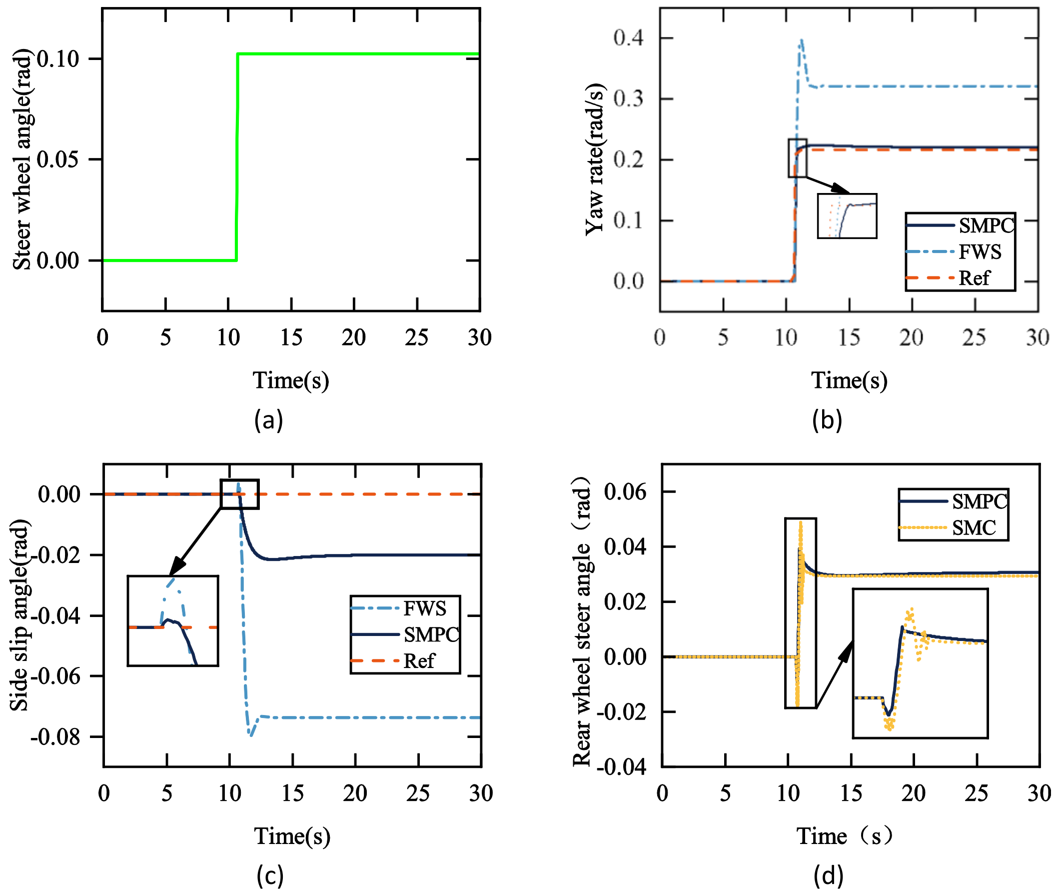

FIGURE 7. Comparison of hardware-in-the-loop simulation results for steering wheel sinusoidal input conditions. Among them, (a) is the steering wheel angle signal map; (b) is the yaw rate comparison between ARS vehicles with SMPC algorithms and FWS vehicles; (c) is the side slip angle comparison between ARS vehicles with SMPC algorithms and FWS vehicles; (d) is the ARS controller output comparison between different control algorithms.

transient response and significant suppression of the overshoot phenomenon. The control effect of the SMPC algorithm is better than that of the SMC algorithm and LQR algorithm, which ensures the handling stability and active safety of the vehicle.

B. STEERING WHEEL CORNER SINE INPUT SIMULATION EXPERIMENT

In this paper, in order to further verify the effectiveness of the active rear-wheel steering controller proposed by the analysis, simulation experiments were carried out in this section under sinusoidal input conditions of the steering wheel angle. In the simulation, the maximum input amplitude of the steering wheel turning angle is $\pi/30\ rad$ and the frequency is 0.05Hz, as shown in Fig. 5(a), and the simulation test results are shown in Fig. 5.

According to Fig. 5(b) and 5(c), the vehicle with the ARS control system can effectively reduce the yaw rate and side slip angle and accurately track their ideal values compared to the FWS vehicle, indicating that the proposed active rear-wheel steering control strategy can effectively reduce the vehicle tail risk at high speed, thus improve the handling stability of the vehicle at high speed. Compared with the SMC algorithm, the deviations of the vehicle's yaw rate and side slip angle are reduced by 0.0003 rad/s and 0.0004 rad, respectively, and there is no jitter. The actual side slip angle of the vehicle is able to track the ideal value of the slip angle with smaller deviations than the LQR algorithm. In Fig. 5(d), it is obvious that the output rear-wheel steering angle of the controller with the SMC algorithm has significant jitter, and the average ripple of the output rear-wheel steering angle is more than 5%. The output rear-wheel steering angle of the controller with the SMPC algorithm is smooth, indicating that the controller of the SMPC algorithm optimized by SMC based on the model prediction algorithm has the good anti-jitter capability. As shown above, the control effect of the SMPC algorithm designed in this paper based on the sliding mode prediction algorithm for optimization of SMC is better than the control effect of the SMC algorithm and the LQR algorithm.

C. HARDWARE-IN-THE-LOOP EXPERIMENT

In this paper, a hardware-in-the-loop simulation platform composed of a DSP28335 controller, a MicroAutoBox II from dSPACE, and a steering wheel angle sensor was built to verify the effectiveness and real-time performance of the SMPC algorithm based on the model prediction algorithm for optimizing SMC in practical engineering applications. In addition, the experimental platform used CAN communication like the real car, with the baud rate set at 500Kbit/s. Firstly, the vehicle model and the active rear-wheel steering control algorithm were downloaded to the MicroAutoBox II and the controller DSP28335 respectively through the automatic code generation function. Secondly, the driver's maneuver command through the steering wheel angle sensor was input to the controller DSP28335. Then the controller DSP28335 based on CAN communication obtained the current vehicle state information to calculate the rear-wheel steering angle, and made the rear-wheel angle signal input to the MicroAutoBox II in real-time vehicle model. Finally, the upper computer comprehensive test software ControlDesk completed the vehicle state information observation and recorded the observation data. The experimental verification of the active rear-wheel steering control algorithm was completed. The workflow diagram is shown in Fig. 6.

As this section focused on verifying the control effectiveness of the control algorithm in practical applications, the steering wheel angle step input condition introduced in section IV-A was used as the verification condition for this section. The setup was the same as in section IV-A, with a stepped steering angle signal of size $\pi/30\,rad$ being input to the front wheel at a certain point and then held constant, as shown in Fig. 7(a). Given a prediction step of p = 10, the control step was taken to be c = 2. The results of the test are shown in Fig. 7.

Fig. 7(b) and 7(c) show that at the beginning of the steering wheel rotation and beyond $\pi/30\,rad$, the actual deviation of the yaw rate and side slip angle have small fluctuations of 10.7 s to 11.1 s due to the presence of human influence. According to Fig. 7(b) and 7(c), it can be seen that the actual deviation of the yaw rate and side slip angle of the ARS vehicle are reduced by 0.1004 rad/s and 0.0536 rad, respectively, at high speed compared to the FWS vehicle, which is significantly lower than that of the FWS vehicle. According to Fig. 7(d), the SMPC algorithm based on the model prediction algorithm is designed to optimize the SMC so that the actual yaw rate can track the ideal value well, with excellent dynamic response and no jitter phenomenon. In summary, the active rear-wheel steering control strategy effectively reduces the risk of high-speed drift and effectively inhibits the jitter phenomenon in practice, and improves the control stability at high speed. The same control effect is obtained as in section IV-A, further verifying the effectiveness and real-time performance of the SMPC algorithm.

V. CONCLUSION

In this paper, an active rear-wheel steering control strategy based on sliding model predictive control algorithm was proposed for vehicle stability control and active safety issues. In terms of the control algorithm, the control rate of the traditional SMC algorithm was corrected and optimized by adopting the ideas of feedback correction and roll optimization in the model predictive control algorithm, thus suppressing the original jitter phenomenon of the SMC algorithm while maintaining its fast response and anti-disturbance capability. The effectiveness and real-time performance of the proposed SMPC active rear-wheel steering controller in practical applications in different operating conditions were verified through a joint simulation platform and a hardware-in-the-loop experimental platform, and the results were compared with those of FWS vehicle with the same parameters and ARS vehicle with sliding mode control algorithm. The results showed that the proposed SMPC algorithm, optimizing SMC based on the model prediction algorithm, reduced the actual steady-state error of the vehicle yaw rate and the overshoot of the yaw rate of the vehicle by 30.012% and 18.103%, respectively, compared with the SMC algorithm, and effectively suppressed the vibration phenomenon. This result indicates that the proposed SMPC algorithm has higher control accuracy, better transient response characteristics, and smoother control output. The ARS vehicle was able to accurately track the ideal value of the yaw rate and trajectory compared to the FWS vehicle, and the actual deviation of the yaw rate and side slip angle were reduced by 28.324% and 68.517%, respectively. This demonstrates that the proposed active rear-wheel steering control strategy can provide better handling stability and active safety at high speeds. In addition, in future research, we will consider the control method of combining active rear-wheel steering and driving force to improve further the handling stability of four-wheel independent drive electric vehicles when the lateral force of tires is saturated.

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