

Article

Torque Coordination Control for Dual-Drive-Axle Electric Commercial Vehicles Under Wheel-Speed Signal Loss

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Abstract

For 6×4 dual-drive-axle battery electric tractors subject to intermediate-axle wheel-speed signal loss, the drive-wheel slip state is difficult to determine under adhesion-limited driving conditions, which can readily lead to drive torque distribution mismatch between the intermediate and rear axles and excessive slip of the intermediate-axle drive wheels. To address the above problems, this paper proposes a torque coordination control method for dual-drive-axle battery electric commercial vehicles. First, a drive-axle slip ratio is constructed based on the motor speed of the drive axle to enable axle-level characterization of the drive-axle slip state under wheel-speed signal loss. Second, the drive-axle slip state is identified using the local variation relationship between the ground driving force and the drive-axle slip ratio, and the reference slip ratio of the drive axle and the wheel-end transmissible torque limit are determined. Then, a hierarchical torque coordination control method is established. In the upper layer, with convergence of the drive-axle slip ratios of the intermediate and rear axles as the objective, the wheel-end transmissible torque limit is introduced for inter-axle drive torque distribution and constraint handling. In the lower layer, super-twisting sliding mode control is employed to perform anti-slip correction on the target torque of the drive axle. Finally, the proposed method is validated in a TruckSim/Simulink co-simulation environment. The results show that, under drive-wheel speed signal loss, the proposed method can achieve reasonable drive torque distribution between the intermediate and rear axles under adhesion-limited driving conditions, suppress excessive slip of the drive axles, and improve the smoothness of the vehicle longitudinal response.

Keywords: electric commercial vehicle; drive-wheel speed signal loss; torque coordination control; drive-axle slip ratio; slip-state identification; dynamic torque distribution



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1. Introduction

With the accelerated electrification of commercial vehicles, 6×4 electric tractors are playing an increasingly important role in road transportation owing to their zero-emission operation, low operating costs, and strong traction capability [1,2]. In this vehicle configuration, the intermediate and rear axles adopt an independent electric-drive architecture, while the intermediate axle is usually not equipped with wheel-speed sensors, making it difficult to directly determine its slip state. Under adhesion-limited driving conditions, the drive wheels of the intermediate axle are prone to excessive slip, which reduces the effective ground driving force and exacerbates tire wear. Meanwhile, excessive slip of the

intermediate axle makes it difficult for the original inter-axle drive torque distribution to adapt to the current adhesion conditions. Therefore, investigating torque coordination control methods for adhesion-limited driving conditions with wheel-speed signal loss is of considerable significance for achieving reasonable drive torque distribution and constraint handling between the intermediate and rear axles and suppressing excessive slip of the drive axles.

Numerous studies have been conducted in the field of acceleration slip regulation. From the perspectives of control strategy and feedback information, existing methods can mainly be classified into drive-torque-regulation-based control methods and slip-ratio-tracking-based control methods [3,4]. Among control methods based on drive torque regulation, Hori et al. [5] referred the translational inertia of the vehicle body to the drive-wheel side, established a reference wheel-speed model under the no-slip condition, and corrected the drive torque input according to the deviation between the actual wheel speed and the reference wheel speed to suppress the abnormal increase in drive-wheel speed on low-adhesion roads. Building on this work, Akiba et al. [6] introduced back electromotive force to correct the reference wheel speed and improved the motor dynamic response and alleviated control saturation through dynamic component separation and control input limiting, thereby enhancing the stability of the torque regulation process. Yin et al. [7,8] proposed a traction control method based on maximum effective torque estimation. In this method, the maximum effective torque transmissible under the current road-surface condition was estimated from the motor torque and the angular acceleration of the drive wheel, and the drive torque was then limited accordingly. On this basis, an acceleration compensation mechanism was introduced to improve torque-tracking performance during the initial stage of acceleration. Hu et al. [9] addressed the sensitivity of the maximum effective torque estimation method to wheel rotational inertia disturbances and inverse-model accuracy by introducing a closed-loop compensator with a friction detection unit. The disturbance torque was estimated online using wheel rotational information and motor torque, thereby improving system robustness and slip control performance. Geamanu et al. [10] estimated the instantaneous adhesion coefficient and maximum adhesion coefficient using motor torque, wheel angular acceleration, and vehicle longitudinal acceleration. In the linear region of the longitudinal tire adhesion characteristics, flatness-based drive torque control was adopted to achieve vehicle-speed tracking, whereas in the nonlinear region, wheel slip was limited through drive torque saturation. On this basis, Geamanu et al. [11] further introduced sliding mode control to drive the instantaneous adhesion coefficient toward the maximum adhesion coefficient, thereby improving acceleration slip regulation performance on roads with varying adhesion. This class of methods mainly achieves drive torque correction by constructing a reference wheel speed, estimating the maximum effective torque, or estimating the road adhesion state, and has demonstrated good slip suppression performance in acceleration slip regulation on low-adhesion roads.

Among control methods based on slip-ratio tracking, threshold logic control has been widely applied in acceleration slip regulation owing to its simple structure and ease of implementation [12]. Wang et al. [13] proposed a target-slip-ratio-based threshold control method, in which the drive torque was adjusted according to the relationship between the actual slip ratio and the threshold, thereby achieving acceleration slip regulation. PID control and its improved variants are also commonly used in acceleration slip regulation owing to their clear control structure and ease of engineering implementation. Nguyen et al. [14] proposed a fuzzy-PID-based acceleration slip regulation method. In this method, the real-time drive-wheel slip-ratio error was used as the input, and the PID parameters were adjusted online through fuzzy logic to suppress excessive slip of the drive wheels. To address the degradation of slip control accuracy caused by axle-load transfer during

acceleration, Chen et al. [15] used the drive-wheel slip-ratio error and trained a BP neural network using an improved particle swarm optimization algorithm to adaptively adjust the PID parameters, thereby improving the accuracy of acceleration slip regulation. Compared with PID control, sliding mode control offers a fast response and strong robustness. For acceleration slip regulation in four-wheel-drive hybrid electric vehicles, related studies have adopted sliding mode control, in which the sliding surface is constructed using the deviation between the real-time slip ratio and the target slip ratio, and the target drive torque is adjusted to improve slip-ratio tracking performance [16]. Li et al. [17] proposed an anti-windup sliding mode control method, in which the deviation between the actual and optimal speeds of the drive wheel was used as the tracking error, and an anti-windup mechanism was introduced to improve control smoothness under low-speed conditions. Considering the characteristics of heavy commercial vehicles, such as substantial load variations, Subramaniyam et al. [18] developed an integral sliding mode control method based on a power-rate exponential reaching law. In this method, an integral sliding surface was constructed using the slip-ratio tracking error, and chattering was suppressed through reaching-law design, thereby improving slip-ratio tracking accuracy and the smoothness of the dynamic response. Savitski et al. [19] proposed a hybrid sliding mode–PID slip control method, in which feedback control was performed based on the actual speed of the drive wheel or the slip-ratio error to achieve accurate slip-ratio tracking on icy and snowy roads. In addition, model predictive control has also been applied to acceleration slip regulation. Shi et al. [20] proposed a method based on nonlinear model predictive control, in which the slip ratio was selected as a key state variable and constraints on the rate of change in the control input were introduced to achieve smooth drive-torque regulation and accurate slip-ratio control. These methods generally use the slip-ratio error or wheel-speed error as the core feedback variable and suppress excessive slip of the drive wheels through feedback correction of the drive torque, thereby improving the traction performance of vehicles under adhesion-limited driving conditions.

The aforementioned studies mainly focus on suppressing drive-wheel slip. For vehicles with multiple drive units, multi-motor drive systems offer high power and torque output capabilities and have advantages in driving flexibility and system redundancy; however, their performance depends on coordinated control among the drive units [21]. When the drive wheels are under adhesion-limited conditions, the original torque distribution relationship cannot sufficiently adapt to the current road adhesion conditions. Therefore, acceleration slip regulation constraints need to be incorporated into the torque distribution process. For front-axle and rear-axle independently driven vehicles, Tong et al. [22] adopted a sliding mode control method, in which the wheel-speed difference between the front and rear wheels was used as the feedback variable to dynamically adjust the front–rear axle torque distribution ratio, thereby suppressing drive-wheel slip under low-adhesion conditions. Zhang et al. [23] and Guo et al. [24] considered the drive-wheel slip state in front–rear axle torque distribution. By applying torque limitation to the drive axle with a higher slip level and providing torque compensation through the other drive axle, these methods maintained vehicle traction performance while suppressing drive-wheel slip. Bao et al. [25] incorporated the slip control results into the constraints of a quadratic-programming-based torque distribution problem, imposed constraints on the driving force of the slipping wheels, and coordinated the torque distribution among the remaining wheels, thereby improving slip control accuracy and torque distribution stability. These studies indicate that, in vehicles with multiple drive units, incorporating drive-wheel slip information into the torque distribution process can suppress excessive slip in individual drive units while coordinating the use of the remaining adhesion capacity available at the other drive units, thereby improving vehicle traction performance.

In summary, existing methods have achieved substantial progress in acceleration slip regulation and torque distribution under acceleration slip regulation constraints. Existing methods generally require drive-wheel speed information and further calculate state variables such as wheel-speed error and slip ratio, which are used as the basis for slip determination, acceleration slip regulation, and torque distribution. For the dual-drive-axle battery electric commercial vehicle studied in this paper, under conditions of intermediate-axle wheel-speed signal loss and difficulty in accurately obtaining drive-axle vertical load information, existing methods are difficult to apply directly to drive torque coordination control between the intermediate and rear axles. Therefore, this paper proposes a torque coordination control method for dual-drive-axle battery electric commercial vehicles. Without relying on drive-wheel speed or drive-axle vertical load information, the proposed method achieves reasonable drive torque distribution between the intermediate and rear axles and suppression of excessive slip. The main contributions of this paper are as follows:

- (1) An axle-level slip-state characterization method under wheel-speed signal loss is constructed. To address the difficulty in characterizing the drive-wheel slip state caused by intermediate-axle wheel-speed signal loss in dual-drive-axle battery electric commercial vehicles, the drive-axle slip ratio is defined based on the drive-axle motor speed. Accordingly, the object of slip-state characterization is extended from an individual drive wheel to the drive axle.
- (2) A drive-axle slip-state identification and parameter determination method is proposed. Without relying on drive-axle vertical load information, the local variation relationship between the ground driving force and the drive-axle slip ratio is used to achieve drive-axle slip-state identification, reference slip ratio determination, and wheel-end transmissible torque limit estimation, thereby providing a basis for subsequent torque coordination control.
- (3) A hierarchical torque coordination control method for the dual-drive-axle system is established. In the upper layer, with convergence of the drive-axle slip ratios of the intermediate and rear axles as the objective, the wheel-end transmissible torque limit is introduced for inter-axle drive torque distribution and constraint handling. In the lower layer, super-twisting sliding mode control is employed to perform anti-slip correction on the target torque of the drive axle. Without relying on drive-wheel speed information, the proposed method achieves reasonable drive torque distribution between the intermediate and rear axles and suppression of excessive slip.

2. Overall Architecture of Dual-Drive-Axle Torque Coordination Control

The overall architecture of torque coordination control for dual-drive-axle electric commercial vehicles is shown in Figure 1. This architecture uses the driver-requested drive torque as input and outputs the target motor torques for the intermediate and rear axles. It consists of three modules: slip-state identification and estimation of the wheel-end transmissible torque limit, inter-axle drive torque distribution and constraint handling, and drive-axle anti-slip torque correction.

The slip-state identification and wheel-end transmissible torque limit estimation module calculates the drive-axle slip ratio based on vehicle feedback signals, including vehicle speed, motor speed, and motor torque. It then identifies the slip state based on the ground driving force of the drive axle and its variation characteristics and determines the wheel-end transmissible torque limit and the reference slip ratio of the drive axle. The drive-axle slip state, drive-axle slip ratio, and wheel-end transmissible torque limit are used as state inputs and constraint information for the inter-axle drive torque distribution and constraint-handling module. The drive-axle slip state, drive-axle slip ratio, and reference slip ratio of

the drive axle are used as state feedback and reference inputs for the drive-axle anti-slip torque-correction module.

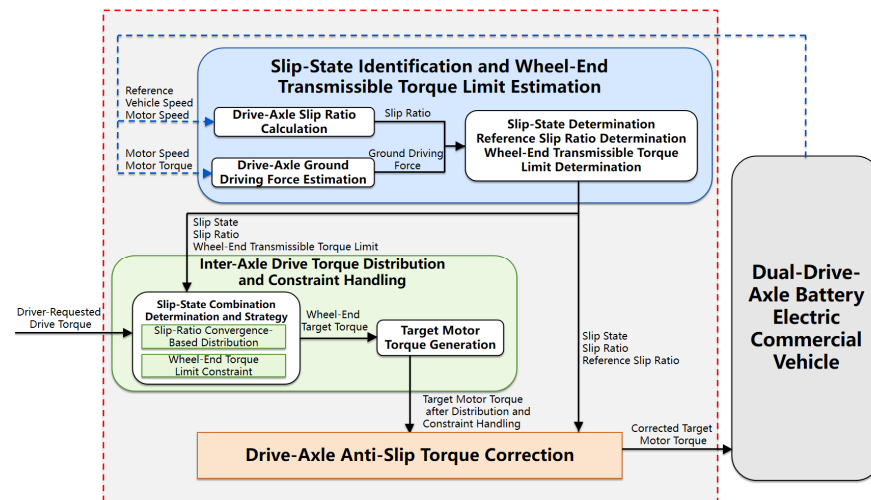


Figure 1. Overall architecture of dual-drive-axle torque coordination control.

The inter-axle drive torque distribution and constraint-handling module distributes and constrains the target motor torques for the intermediate- and rear-axle motors according to the driver-requested drive torque and the state information of each drive axle. This module functions as a feedforward distribution stage based on the slip state and the wheel-end transmissible torque limit. According to the differences in slip state and torque transfer capability between the intermediate and rear axles, the target torque of the drive axle with a higher slip level is reduced, while part of the demanded drive torque is reallocated to the drive axle with a lower slip level within its allowable range.

The drive-axle anti-slip torque-correction module is located downstream of the inter-axle drive torque distribution and constraint-handling module. Based on the error between the drive-axle slip ratio and the reference slip ratio of the drive axle, this module further corrects the target motor torques of the intermediate- and rear-axle motors after distribution and constraint handling and applies the corrected target motor torques to the vehicle. This process is updated online using vehicle feedback signals, forming a closed-loop torque coordination control process.

3. Drive-Axle Slip Ratio Construction Based on Motor Speed and Constraint Characteristic Analysis

3.1. Drive-Axle Slip Ratio Construction Method

In cases of drive-wheel speed signal loss, the rotational speed of an individual drive wheel is difficult to obtain directly; therefore, conventional slip-state characterization methods that rely on drive-wheel speed cannot be applied directly. For dual-drive-axle electric commercial vehicles, the drive-axle motor speed can be acquired in real time and converted through the transmission system to obtain the equivalent wheel-end speed of the drive axle. Accordingly, this paper constructs a drive-axle slip ratio based on the drive-axle motor speed to describe the axle-level slip state in cases of wheel-speed signal loss.

The equivalent wheel-end speed of the drive axle is expressed as:

$$\omega_{eq} = \frac{\omega_m}{i_g i_0 i_w} \quad (1)$$

where ω_m is the drive-axis motor speed, i_g, i_0, i_w are the transmission gear ratio, final drive ratio, and wheel-end reducer ratio, respectively.

Based on the relationship between the equivalent wheel-end speed of the drive axle and the reference vehicle speed under driving conditions, the drive-axle slip ratio is defined as:

$$s_a = \frac{r\omega_{eq} - v}{r\omega_{eq}} \quad (2)$$

where r is the wheel rolling radius, and v is the reference vehicle speed, which is obtained from the left and right wheel speeds of the non-driven front wheels after conversion using the rolling radius and filtering.

In this paper, the drive-axle slip ratio is constructed based on the forward kinematic relationship, while the effects of factors such as the transmission system, backlash, and variations in tire rolling radius on the slip ratio are neglected. To illustrate the rationality of the drive-axle slip ratio, a simulation comparison between the drive-axle slip ratio and the actual wheel-end slip ratio of the drive wheels is conducted, as shown in Figure 2. It can be observed that the deviation between the drive-axle slip ratio and the actual slip ratio of the drive wheels is small, and their overall variation trends are generally consistent. This indicates that, under wheel-speed signal loss, the drive-axle slip ratio can effectively characterize the slip state, thereby providing a basis for subsequent state identification, parameter determination, and torque coordination control.

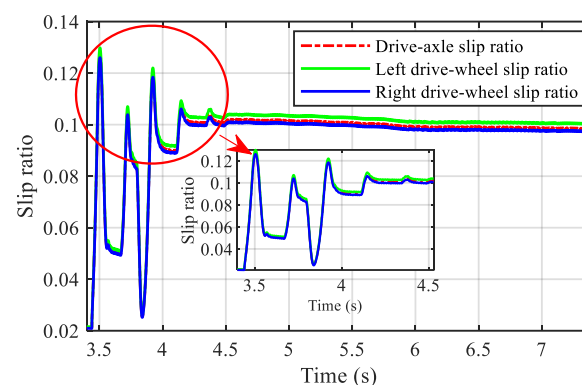


Figure 2. Comparison between the drive-axle slip ratio and the drive-wheel slip ratio.

3.2. Constraint Characteristics of the Drive-Axle Slip Ratio with Respect to Excessive Slip of the Drive Wheels

A well-defined kinematic relationship exists between the drive-axle motor speed and the rotational speeds of the left and right drive wheels. After being referred through the transmission system, the equivalent wheel-end speed of the drive axle is equal to the average speed of the left and right drive wheels. Therefore, the drive-axle slip ratio and the slip ratios of the left and right drive wheels satisfy the following constraint relationship:

$$s_a = \frac{s_L + s_R - 2s_L s_R}{2 - s_L - s_R} \quad (3)$$

where s_L and s_R are the slip ratios of the left and right drive wheels, respectively.

Equation (3) indicates that, for a given drive-axle slip ratio, the slip ratios of the left and right drive wheels cannot vary independently but are constrained by the axle-level slip state.

On a uniform low-adhesion road surface, the adhesion conditions of the left and right drive wheels are essentially identical, and the difference in tire force conditions is relatively small. Therefore, the slip states of the left and right drive wheels are usually close, i.e.,

$s_L \approx s_R$. According to Equation (3), $s_a \approx s_L \approx s_R$ can be obtained. This indicates that, when the adhesion conditions of the left and right drive wheels are similar, the drive-axle slip ratio can approximately characterize the slip states of the drive wheels on both sides of the drive axle. Controlling the drive-axle slip ratio can approximately impose a simultaneous constraint on the slip ratios of the drive wheels, thereby suppressing excessive slip.

Under split- μ road conditions, the adhesion conditions of the left and right drive wheels differ significantly, and s_L and s_R may exhibit substantial divergence. In this case, the drive-axle slip ratio is no longer equal to the slip ratio of either drive wheel. To analyze the boundary of the one-sided drive-wheel slip ratio under the constraint of the drive-axle slip ratio, the most unfavorable split-adhesion condition is considered. The slip ratio on the side with higher slip is denoted as $s_h = \max(s_L, s_R)$, and the drive wheel on the other side is approximately assumed to be in a pure rolling state, i.e., $\min(s_L, s_R) \approx 0$. Substituting these conditions into Equation (3) and combining them with the drive-axle slip ratio control target $s_a \leq s_{th}$ yields:

$$s_h \leq \frac{2s_{th}}{1 + s_{th}} \quad (4)$$

Equation (4) gives the upper bound on the one-sided drive-wheel slip ratio under the constraint of the drive-axle slip ratio. The slip-ratio boundaries of the left and right drive wheels under the drive-axle slip-ratio threshold constraint are shown in Figure 3. The different curves in the figure represent the boundary relationships between the slip ratios of the left and right drive wheels under different drive-axle slip-ratio thresholds. Taking $s_{th} = 0.20$ as an example, the light-blue region represents the feasible combination region of the left and right drive-wheel slip ratios that satisfies this threshold constraint. When the slip ratio of one drive wheel approaches 0, the boundary value of the slip ratio of the other drive wheel is approximately 0.33. Therefore, although the drive-axle slip ratio cannot uniquely determine the one-sided drive-wheel slip ratio, limiting the drive-axle slip ratio can still impose a boundary constraint on excessive slip of one drive wheel.

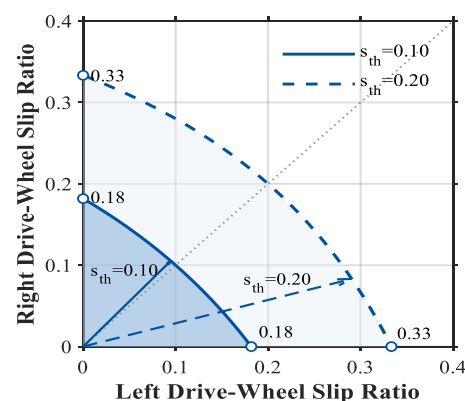


Figure 3. Slip-ratio boundaries of the left and right drive wheels under the drive-axle slip-ratio threshold constraint.

4. Drive-Axle Ground Driving Force Estimation and Slip-State Identification

Existing methods for slip-state identification and reference slip ratio determination mostly rely on information such as wheel speed and vertical load. For dual-drive-axle electric commercial vehicles, such methods are difficult to apply directly when drive-wheel speed signals are unavailable and accurate drive-axle vertical-load information cannot be obtained. Therefore, this paper identifies the drive-axle slip state based on the local characteristics of the drive-axle ground driving force with respect to the drive-axle slip ratio

and determines the reference slip ratio of the drive axle and the wheel-end transmissible torque limit.

4.1. Drive-Axle Ground Driving Force Estimation

The drive-axle ground driving force can characterize the level of longitudinal force currently transmitted between the tire and the road surface, and it serves as an important basis for subsequent slip-state identification. According to the relationship between motor drive torque and wheel-end rotational dynamics, a drive-axle ground driving force estimation model is established [7,9]. To reduce the influence of fluctuations in the instantaneous estimated value of the ground driving force on slip-state identification, a recursive least squares method with a forgetting factor is adopted to update the parameters of the ground driving force estimation model online. The observation used for model updating at time instant k is defined as y_k :

$$y_k = \phi_k^T \theta + \zeta_k \quad (5)$$

where $\phi_k = [T_w(k) \ \dot{\omega}_{eq}(k)]^T$ is the regression vector at k ; $T_w(k)$ is the wheel-end drive torque of the drive axle at k ; $\dot{\omega}_{eq}(k)$ is the equivalent wheel-end angular acceleration of the drive axle at k ; $\theta = [\theta_1 \ \theta_2]^T$ is the parameter vector to be estimated; and ζ_k is the noise at k . Expanding Equation (5) yields:

$$y_k = \theta_1 T_w(k) + \theta_2 \dot{\omega}_{eq}(k) + \zeta_k \quad (6)$$

The parameter vector is updated online using the recursive least squares method with a forgetting factor, and the estimation error is defined as:

$$e_k = y_k - \phi_k^T \hat{\theta}_{k-1} \quad (7)$$

where $\hat{\theta}_{k-1}$ is the estimate of the parameter vector at time instant $k - 1$.

The recursive equations are expressed as:

$$\begin{cases} K_k = \frac{P_{k-1} \phi_k}{\lambda + \phi_k^T P_{k-1} \phi_k} \\ \hat{\theta}_k = \hat{\theta}_{k-1} + K_k e_k \\ P_k = \frac{1}{\lambda} (I - K_k \phi_k^T) P_{k-1} \end{cases} \quad (8)$$

where K_k is the gain vector at time instant k ; P_{k-1} and P_k are the covariance matrices at time instants $k - 1$ and k , respectively; $\hat{\theta}_k$ is the estimate of the parameter vector at time instant k ; and λ is the forgetting factor. This factor is mainly used to regulate the influence of historical data on the current estimation result. Considering the estimation stability, noise robustness, and dynamic tracking capability under adhesion abrupt-change conditions, this paper sets this factor to $\lambda = 0.98$ [26]. Meanwhile, to reduce the fluctuation level during the initial stage of estimation, the initial covariance matrix is set to $P_0 = I$.

Thus, the estimated drive-axle ground driving force is obtained as:

$$\hat{F}_d(k) = \phi_k^T \hat{\theta}_k \quad (9)$$

To verify the accuracy of the ground driving force estimation results under measurement noise, validation was conducted with a motor torque measurement noise standard deviation of 5 N·m and a motor angular acceleration measurement noise standard deviation of 3 rad/s². The results are shown in Figure 4. Under the given measurement noise disturbance, the estimated ground driving force can closely track the variation in the actual ground driving force, with a relative error of less than 2.5% and a small estimation delay, thereby satisfying the control requirements.

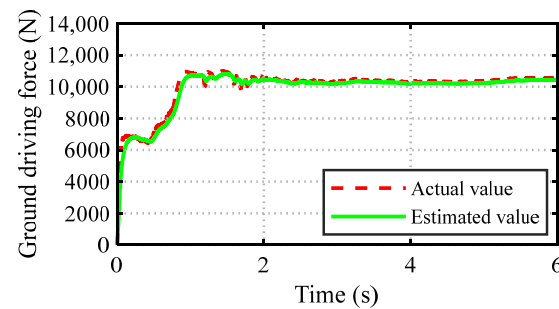


Figure 4. Comparison between the estimated and actual ground driving forces.

4.2. Drive-Axle Slip-State Identification Based on the Variation Characteristics of the Ground Driving Force

As indicated by the longitudinal tire adhesion characteristics, as the drive-axle slip ratio increases, the ground driving force generally increases initially, then approaches a peak, and eventually decreases. Based on this characteristic, this paper classifies the drive-axle slip state into the states of insufficient slip, ideal slip, and excessive slip, and uses the local slope of the ground driving force with respect to the drive-axle slip ratio as the criterion for state identification.

To obtain the local variation trend between the ground driving force and the drive-axle slip ratio under the current operating condition, a rolling-window method is used to acquire multiple sets of synchronized data pairs. The data within the window are then fitted linearly, and the slope of the fitted line is taken as the local slope K_d for the current window. The selection of the rolling-window length needs to comprehensively consider noise robustness and identification timeliness. If the window is too short, the local slope becomes more sensitive to data fluctuations, which may easily lead to misidentification of the slip state. If the window is too long, changes in the local slope will be delayed, which is not conducive to timely identification of the drive-axle slip state [27]. In this paper, the rolling-window length is set to 200.

The slope thresholds $K_{thr,L}$ and $K_{thr,R}$ are set to satisfy $K_{thr,L} > K_{thr,R}$. The slope thresholds are used to classify the states of insufficient slip, ideal slip, and excessive slip, and the range bounded by $K_{thr,L}$ and $K_{thr,R}$ corresponds to the interval width of the state of ideal slip. If this interval is too narrow, the drive axle is more likely to be identified as being in a state of insufficient slip or excessive slip, which may lead to frequent switching of the control strategy. If this interval is too wide, the state of excessive slip may not be identified in a timely manner, which may result in a relatively high drive-axle slip ratio. To reduce the difficulty of parameter calibration, the interval corresponding to the state of ideal slip is set as a small symmetric interval in this paper, and the slope thresholds $K_{thr,L}$ and $K_{thr,R}$ are set to 40,000 and $-40,000$, respectively. When $K_d > K_{thr,L}$, the ground driving force increases significantly with an increase in the drive-axle slip ratio, indicating that the drive axle is in the state of insufficient slip. When $K_{thr,R} \leq K_d \leq K_{thr,L}$, the variation in the ground driving force tends to level off and approach the peak region, indicating that the drive axle is in the state of ideal slip. When $K_d < K_{thr,R}$, the ground driving force no longer increases or even decreases, indicating that the drive axle is in the state of excessive slip. The classification of the drive-axle slip state is shown in Figure 5.

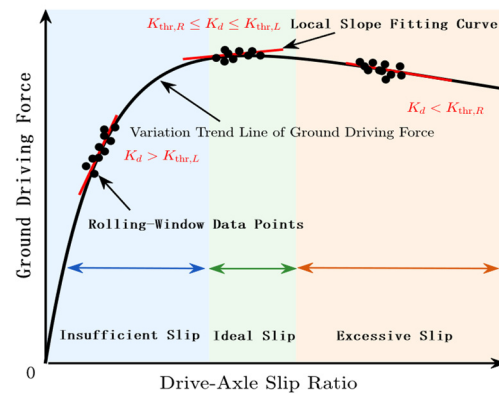


Figure 5. Drive-axle slip-state classification based on the variation characteristics of the ground driving force.

4.3. Determination of the Drive-Axle Wheel-End Transmissible Torque Limit and Reference Slip Ratio

When the drive axle is in the state of insufficient slip, the ground driving force increases significantly with an increase in the drive-axle slip ratio, indicating that the current road adhesion capability has not yet been fully utilized. Therefore, the window data in this state are not used to update either the reference slip ratio of the drive axle or the wheel-end transmissible torque limit.

When the drive axle is in the state of ideal slip, the ground driving force is within the peak region under the current road conditions. The window data in this state can therefore be used to update the reference slip ratio of the drive axle and the wheel-end transmissible torque limit. In this case, the maximum value of the ground driving force within the rolling window is extracted as the current peak transmissible ground driving force of the drive axle, namely:

$$\hat{F}_{d,\max} = \max_{j=1,2,\dots,N} \{ \hat{F}_{d,j} \} \quad (10)$$

The drive-axle slip ratio corresponding to this maximum value is then selected as the reference slip ratio of the drive axle:

$$s_{ref} = s_{a,j^*}, \quad j^* = \arg \max_{j=1,2,\dots,N} \{ \hat{F}_{d,j} \} \quad (11)$$

where s_{a,j^*} denotes the drive-axle slip ratio corresponding to the maximum ground driving force within the rolling window, and j^* denotes the sample index corresponding to the maximum ground driving force within the rolling window.

The wheel-end transmissible torque limit is further obtained as:

$$T_{w,able} = r \hat{F}_{d,\max} \quad (12)$$

When the drive axle is in the state of excessive slip, the drive-axle slip ratio continues to increase, whereas the ground driving force no longer increases and may even decrease. To prevent data from the descending branch under excessive slip from causing the reference slip ratio and the wheel-end transmissible torque limit to deviate from the peak region, the data in this segment are no longer used for parameter updating. Instead, the most recent valid parameter results obtained in the state of ideal slip are retained. The wheel-end transmissible torque limit is used for inter-axle drive torque distribution constraints, whereas the reference slip ratio of the drive axle is used for drive-axle anti-slip torque correction.

5. Inter-Axle Drive Torque Distribution Strategy

The intermediate and rear axles of a 6×4 electric tractor use an independent drive architecture. Each drive axle can be classified into the state of insufficient slip, state of ideal slip, or state of excessive slip, resulting in nine slip-state combinations for the intermediate and rear axles, as shown in Table 1.

Table 1. Intermediate- and rear-axle slip-state combinations.

Slip-State Combination		Rear-Axle Slip State		
		Insufficient Slip (A)	Ideal Slip (B)	Excessive Slip (C)
Intermediate-Axle Slip State	insufficient slip (A)	AA	AB	AC
	ideal slip (B)	BA	BB	BC
	excessive slip (C)	CA	CB	CC

The inter-axle drive torque distribution strategy proposed in this paper consists of two components: drive-axle slip-ratio convergence and the wheel-end transmissible torque limit constraint. The drive-axle slip-ratio convergence component reduces the slip-ratio deviation between the intermediate and rear axles and drives the slip states of the two axles toward consistency, thereby coordinating the drive torque distribution between the intermediate and rear axles. The wheel-end transmissible torque limit is used to constrain the wheel-end target torque of the drive axle and, accordingly, to limit the target motor torques of the intermediate- and rear-axle motors.

The drive-axle slip-ratio convergence component determines the inter-axle dynamic torque distribution coefficient according to the slip-ratio deviation between the intermediate and rear axles and obtains the wheel-end target torques of the intermediate and rear axles, so that the slip ratios of the two axles are driven toward consistency. Its control block diagram is shown in Figure 6.

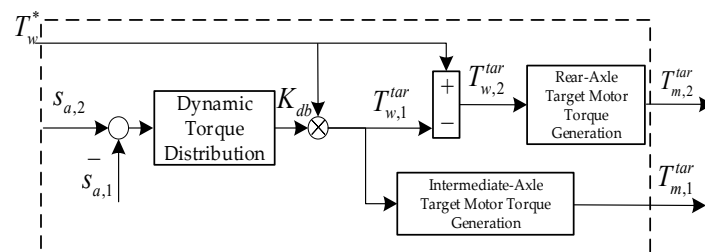


Figure 6. Inter-axle dynamic torque distribution based on drive-axle slip-ratio convergence.

In Figure 6, T_w^* denotes the driver-requested wheel-end torque, and $s_{a,1}$, $T_{w,1}^{tar}$, and $T_{m,1}^{tar}$ denote the slip ratio, wheel-end target torque, and target motor torque of the intermediate axle, respectively. Similarly, $s_{a,2}$, $T_{w,2}^{tar}$ and $T_{m,2}^{tar}$ denote the corresponding parameters of the rear axle.

5.1. Inter-Axle Drive Torque Distribution Rules Based on Slip-State Combinations

For different slip-state combinations, the inter-axle drive torque distribution rules are as follows:

- (1) For the AA combination, both axles are in the state of insufficient slip. Through inter-axle dynamic torque distribution, the allocated drive torque of the drive axle with the higher slip ratio is reduced, while the allocated drive torque of the drive axle with the lower slip ratio is appropriately increased. This drives the slip ratios of the intermediate and rear axles toward consistency, thereby coordinating the drive

torque distribution between the intermediate and rear axles and reducing the tire-wear difference caused by inter-axle slip differences. The slip-ratio convergence process is shown in Figure 7.

- (2) For the AB and AC combinations, the intermediate axle is in the state of insufficient slip, whereas the rear axle is in the state of ideal slip or excessive slip. Through inter-axle dynamic torque distribution, the drive torque of the intermediate axle is appropriately increased while that of the rear axle is reduced, thereby decreasing the slip level of the rear axle. To prevent the intermediate axle from entering the state of excessive slip due to the increase in drive torque, when the slip level of the intermediate axle approaches the state of ideal slip, any further increase in the allocated drive torque of the intermediate axle is stopped, and the torque distribution coefficient from the previous time instant is retained.

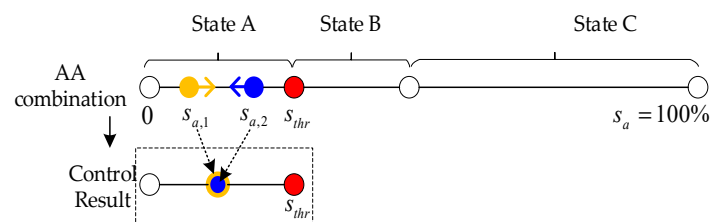


Figure 7. Schematic diagram of drive-axle slip-ratio convergence under the AA combination.

As shown in Figure 8, taking the AB combination as an example, after inter-axle dynamic torque distribution, the slip ratios of the intermediate and rear axles converge, and the slip ratio of the rear axle decreases. The control result may transition from the AB combination to the AA combination, or it may remain in the AB combination after the slip level of the rear axle decreases. For the AC combination, because the rear axle is already in the state of excessive slip, inter-axle dynamic torque distribution is mainly used to reduce the target drive torque of the rear axle and decrease its degree of excessive slip. If the rear axle remains in the state of excessive slip after distribution, rear-axle anti-slip control continues to correct the drive torque to suppress its excessive slip.

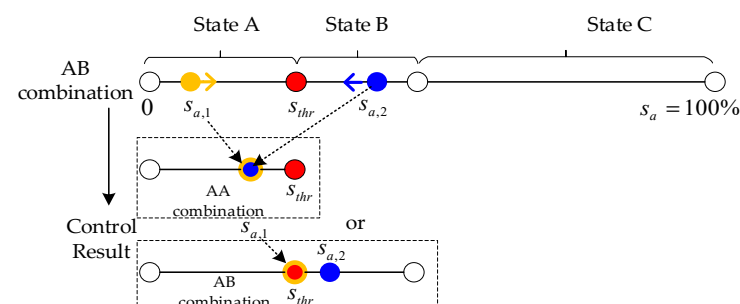


Figure 8. Schematic diagram of drive-axle slip-ratio convergence under the AB combination.

Similarly, for the BA and CA combinations, the same distribution rules as those for the AB and AC combinations can be adopted.

- (3) For the BB, BC, CB, and CC combinations, the wheel-end target torques are constrained according to the wheel-end transmissible torque limits of the intermediate and rear axles, and the target motor torques of the intermediate- and rear-axle motors are determined accordingly. When both axles are in the state of ideal slip, only drive torque distribution is performed. When the combination includes a state of excessive slip, anti-slip torque correction is applied to the drive axle in the state of excessive

slip on the basis of drive torque distribution, thereby suppressing excessive slip of that axle.

5.2. Determination of the Dynamic Torque Distribution Coefficient Based on Drive-Axle Slip-Ratio Convergence

To determine the inter-axle dynamic torque distribution coefficient K_{db} , a sliding mode control framework is introduced, in which the sliding surface is constructed using the slip-ratio deviation between the intermediate and rear axles:

$$\sigma_a = s_{a,2} - s_{a,1} \rightarrow 0 \quad (13)$$

where σ_a is the slip-ratio deviation between the intermediate and rear axles, and the control objective is to drive σ_a toward zero.

Based on the definition of the drive-axle slip ratio given in Equation (2), the rotational dynamics of the drive axle, and the inter-axle drive torque distribution relationship shown in Figure 6, the following expression can be obtained:

$$\dot{\sigma}_a = \frac{v}{r} \left[\frac{(1 - K_{db})T_w^* - \hat{F}_{d,2}r}{J_{eq,2}\omega_{eq,2}^2} - \frac{K_{db}T_w^* - \hat{F}_{d,1}r}{J_{eq,1}\omega_{eq,1}^2} \right] + \frac{\dot{v}(\omega_{eq,2} - \omega_{eq,1})}{r\omega_{eq,1}\omega_{eq,2}} \quad (14)$$

where $\hat{F}_{d,1}$, $J_{eq,1}$ and $\omega_{eq,1}$ denote the estimated ground driving force of the intermediate axle, the transmission-system rotational inertia referred to the wheel end, and the equivalent wheel-end speed of the intermediate axle, respectively; and $\hat{F}_{d,2}$, $J_{eq,2}$ and $\omega_{eq,2}$ denote the corresponding parameters of the rear axle.

Let $\dot{\sigma}_a = 0$. The vehicle-speed derivative term in Equation (14) that is independent of K_{db} is treated as a disturbance term, and a sliding mode switching term is introduced. The dynamic torque distribution coefficient can then be obtained as:

$$K_{db} = \frac{T_w^* J_{eq,1} \omega_{eq,1}^2 - \hat{F}_{d,2} r J_{eq,1} \omega_{eq,1}^2 + \hat{F}_{d,1} r J_{eq,2} \omega_{eq,2}^2}{T_w^* (J_{eq,1} \omega_{eq,1}^2 + J_{eq,2} \omega_{eq,2}^2)} + \lambda_a \text{sat} \left(\frac{\sigma_a}{\phi_a} \right) \quad (15)$$

where λ_a is the switching gain, and ϕ_a is the boundary-layer thickness. After saturation limiting, K_{db} satisfies $0 \leq K_{db} \leq 1$.

6. Drive-Axle Anti-Slip Torque Correction Method

When a drive axle is in the state of excessive slip, inter-axle drive torque distribution alone is insufficient to fully suppress slip of that axle. Therefore, anti-slip correction needs to be applied to the target motor torque of that axle, so that the drive-axle slip ratio converges to the reference slip ratio. In this paper, a super-twisting sliding mode control method (ST-SMC) is adopted to calculate the anti-slip torque correction term, thereby reducing the chattering caused by high-frequency switching in conventional sliding mode control.

The drive-axle slip-ratio tracking control problem is expressed as a second-order system:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = u + d \end{cases} \quad (16)$$

where $x_1 = s_a - s_{ref}$, $x_2 = \dot{s}_a - \dot{s}_{ref}$ and u denotes the drive-axle anti-slip torque correction term, and d denotes the disturbance.

The sliding surface is selected as:

$$\sigma = c_1 x_1 + x_2 = c_1 (s_a - s_{ref}) + (\dot{s}_a - \dot{s}_{ref}) \quad (17)$$

where $c_1 > 0$. When the system state converges along the sliding surface, x_1 and x_2 gradually approach zero, and the drive-axle slip ratio approaches the reference slip ratio.

Considering that the slip ratio and its derivative are susceptible to noise in practical implementation, a super-twisting observer [28] is adopted in this paper to estimate the system states. The observer equations are given as:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + k_1 |e_1|^{1/2} \text{sgn}(e_1) \\ \dot{\hat{x}}_2 = u + k_2 \text{sgn}(e_1) \end{cases} \quad (18)$$

where $\hat{x}_1$ is the estimate of x_1 ; $\hat{x}_2$ is the estimate of x_2 ; $e_1 = x_1 - \hat{x}_1$ is the estimation error; and k_1 and k_2 are the observer gains.

Furthermore, by introducing the super-twisting sliding mode control law [29], the following expression is obtained:

$$\begin{cases} u = -c_1 \hat{x}_2 - \lambda_1 |\hat{\sigma}|^{1/2} \text{sgn}(\hat{\sigma}) + \zeta \\ \dot{\zeta} = -\lambda_2 \text{sgn}(\hat{\sigma}) \end{cases} \quad (19)$$

where $\hat{\sigma} = c_1 \hat{x}_1 + \hat{x}_2$ is the estimate of the sliding surface; λ_1 and λ_2 are the super-twisting control gains; and ζ is the integral compensation term. Thus, the drive-axle anti-slip torque correction term is obtained and used to correct the target motor torque of the drive axle in the state of excessive slip.

7. Simulation Validation

In this paper, a vehicle combination consisting of a dual-drive-axle battery electric 6×4 tractor and a three-axle trailer is used as the study object. The total mass of the vehicle combination is 19,000 kg, the tire rolling radius is 0.527 m, and the wheel rotational inertia is $4.51 \text{ kg} \cdot \text{m}^2$. To verify the effectiveness of the proposed method, simulations were conducted in a TruckSim 2024.1/Simulink (MATLAB R2023a) co-simulation environment. Considering that the proposed method does not use the slip ratio of an individual drive wheel as the feedback variable, the drive-axle slip ratio was adopted as the control feedback variable for all four control methods to ensure feedback-variable consistency across different control methods. The settings of the four control methods are shown in Table 2.

Table 2. Settings of different control methods.

Method	Inter-Axle Drive Torque Distribution	Drive-Axle Anti-Slip Control
Method 1	1:1 fixed distribution	PID
Method 2	1:1 fixed distribution	SMC
Method 3	Dynamic torque distribution based on slip-ratio convergence	SMC
Proposed Method	Dynamic torque distribution based on slip-ratio convergence	ST-SMC

7.1. Evaluation Metrics

To comprehensively evaluate the effectiveness of the proposed method, the standard deviation of longitudinal acceleration fluctuations, the cumulative drive-axle slip, and the motor torque fluctuation index are selected for quantitative evaluation of the control performance.

- (1) The standard deviation of longitudinal acceleration fluctuations σ_{a_x} , is used to reflect the fluctuation level of longitudinal acceleration during acceleration. A smaller value indicates a smoother acceleration process. It is calculated as:

$$\begin{aligned}\sigma_{a_x} &= \sqrt{\frac{1}{T} \int_0^T [a_x(t) - \bar{a}_x]^2 dt} \\ \bar{a}_x &= \frac{1}{T} \int_0^T a_x(t) dt\end{aligned}\quad (20)$$

where $a_x(t)$ is the longitudinal acceleration at time t , $\bar{a}_x$ is the average longitudinal acceleration within the evaluation time window, and T is the length of the evaluation time window.

- (2) The cumulative drive-axle slip I_s is used to quantify the cumulative slip level of the intermediate and rear axles and indirectly reflects the potential risk of tire wear. A larger value indicates more pronounced cumulative slip. It is calculated as:

$$I_s = \int_0^T [|s_{a,1}(t)| + |s_{a,2}(t)|] dt \quad (21)$$

where $s_{a,1}(t)$ and $s_{a,2}(t)$ denote the drive-axle slip ratios of the intermediate and rear axles at time t , respectively.

- (3) The motor torque fluctuation index σ_{T_m} , is defined as the sum of the standard deviations of the corrected target motor torques of the intermediate- and rear-axle motors. This metric is used to evaluate the smoothness of the torque control process. A smaller value indicates smaller motor torque fluctuations and a smoother control process. It is calculated as:

$$\begin{aligned}\sigma_{T_m} &= \sum_{b=1}^2 \sqrt{\frac{1}{T} \int_0^T [T_{m,b}^{cmd}(t) - \bar{T}_{m,b}^{cmd}]^2 dt} \\ \bar{T}_{m,b}^{cmd} &= \frac{1}{T} \int_0^T T_{m,b}^{cmd}(t) dt\end{aligned}\quad (22)$$

where $T_{m,b}^{cmd}(t)$ is the corrected target motor torque of the motor corresponding to the $b - th$ drive axle at time t , $\bar{T}_{m,b}^{cmd}$ is its average value within the evaluation time window, and $b = 1, 2$ denotes the intermediate and rear axles, respectively.

7.2. Analysis of Simulation Results

Three typical adhesion-limited driving cases are designed in this paper. Case 1 is a uniform low-adhesion road condition, with a road adhesion coefficient of 0.2. Case 2 is a split- μ road condition, where the road adhesion coefficient is 0.2 on one side and 0.5 on the other side. Case 3 is an instantaneous adhesion abrupt-change road condition, in which a 1 m long low-adhesion section with an adhesion coefficient of 0.2 is set on a base road surface with an adhesion coefficient of 0.5 to simulate an abrupt change in the road adhesion coefficient.

Figure 9 presents the simulation results of the four control methods under the uniform low-adhesion road condition. As shown in the figure, Methods 1 and 2 both exhibit excessive slip of the drive axles during the launch stage, with the drive-axle slip ratios remaining generally high. Compared with Method 2, Method 3 adopts the dynamic torque distribution strategy based on slip-ratio convergence in the upper layer, which substantially reduces the drive-axle slip ratio. Combined with the data in Table 3, the cumulative drive-axle slip decreases from 0.54 to 0.30, indicating a significant improvement in acceleration slip regulation performance. Compared with Method 3, the proposed method further employs ST-SMC in the lower-layer anti-slip control. The standard deviation of longitudinal acceleration fluctuations decreases from 0.10 m/s² to 0.09 m/s², the cumulative drive-axle slip decreases from 0.30 to 0.29, and the motor torque fluctuation index decreases from 105.27 N·m to 95.62 N·m. These results indicate that the proposed method can effectively reduce the degree of drive-axle slip under the uniform low-adhesion road condition,

decrease motor torque and longitudinal acceleration fluctuations, and provide good slip suppression performance and longitudinal response smoothness.

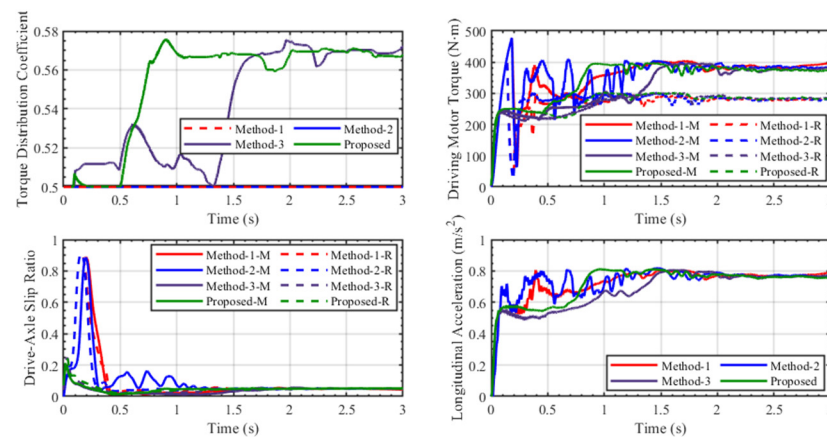


Figure 9. Simulation results under the uniform low-adhesion road condition.

Table 3. Comparison of evaluation metrics for different control methods.

Case	Control Method	σ_{a_x} (m/s ²)	I_s	σ_{T_m} (N·m)
Case 1	Method 1	0.11	0.53	110.08
	Method 2	0.10	0.54	112.81
	Method 3	0.10	0.30	105.27
	Proposed Method	0.09	0.29	95.62
Case 2	Method 1	0.18	1.75	168.54
	Method 2	0.19	1.04	167.64
	Method 3	0.17	0.93	155.39
	Proposed Method	0.17	0.83	149.97
Case 3	Method 1	0.06	0.08	54.40
	Method 2	0.05	0.08	47.53
	Method 3	0.02	0.07	61.96
	Proposed Method	0.02	0.07	60.29

Figure 10 presents the simulation results of the four control methods under the split- μ road condition. As shown in the figure and Table 3, the drive-axle slip ratios of Methods 1 and 2 remain generally high. Compared with Method 2, Method 3 reduces the cumulative drive-axle slip from 1.04 to 0.93, indicating that the upper-layer dynamic torque distribution strategy based on slip-ratio convergence is also applicable to the split- μ road condition. Compared with Method 3, the proposed method maintains the standard deviation of longitudinal acceleration fluctuations at 0.17 m/s², reduces the cumulative drive-axle slip from 0.93 to 0.83, and reduces the motor torque fluctuation index from 155.39 N·m to 149.97 N·m. The results show that, under the split- μ road condition, the proposed method still provides good slip suppression performance and longitudinal response smoothness.

Figure 11 presents the simulation results of the four control methods under the instantaneous adhesion abrupt-change road condition. As shown in the figure, when the vehicle passes through the low-adhesion section, the torque distribution coefficients of Method 3 and the proposed method change markedly, and the corresponding peak values and overall levels of the drive-axle slip ratios are significantly lower than those of Methods 1 and 2. This indicates that the upper-layer dynamic torque distribution strategy based on slip-ratio convergence can respond to instantaneous adhesion changes and helps suppress short-duration slip caused by abrupt adhesion changes.

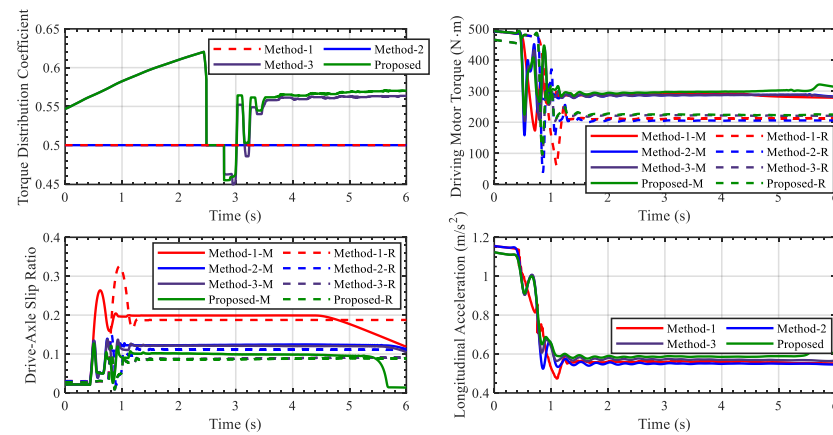


Figure 10. Simulation results under the split- μ road condition.

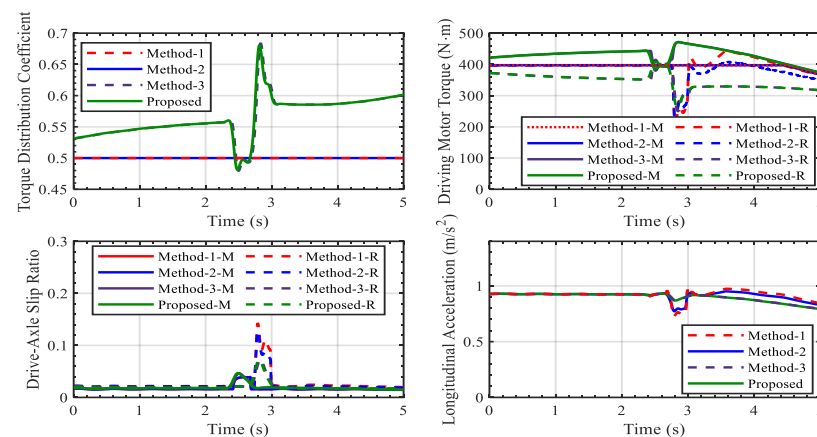


Figure 11. Simulation results under the instantaneous adhesion abrupt-change road condition.

The simulation results under the three cases show that the upper-layer dynamic torque distribution strategy based on slip-ratio convergence can adjust the drive torque distribution between the intermediate and rear axles according to the drive-axle slip state, thereby providing more reasonable target torques for the lower-layer anti-slip torque correction. On this basis, the lower-layer ST-SMC further improves the effectiveness of anti-slip torque correction. The proposed method exhibits good control performance under the uniform low-adhesion, split- μ , and instantaneous adhesion abrupt-change road conditions.

8. Conclusions

To address the problem that, under intermediate-axle wheel-speed signal loss, dual-drive-axle battery electric commercial vehicles are prone to inter-axle drive torque distribution mismatch and excessive slip of the drive axles under adhesion-limited driving conditions, this paper proposed a hierarchical torque coordination control method. The main conclusions are as follows:

- (1) A drive-axle slip ratio was constructed based on the drive-axle motor speed, and the object of slip-state characterization was extended from an individual drive wheel to the drive axle, thereby enabling axle-level characterization of the drive-axle slip state under wheel-speed signal loss.
- (2) Using the local variation relationship between the ground driving force and the drive-axle slip ratio, drive-axle slip-state identification, reference slip ratio determination, and wheel-end transmissible torque limit estimation were achieved. This provided a

constraint basis for torque coordination control without relying on drive-axle vertical load information.

- (3) A hierarchical torque coordination control strategy was established. In the upper layer, inter-axle drive torque distribution and constraint handling were performed to achieve dynamic coordination of the drive torque between the intermediate and rear axles. In the lower layer, super-twisting sliding mode control was used to perform anti-slip correction on the target torque of the drive axle, thereby suppressing excessive slip of the drive axles.
- (4) The simulation results show that, under typical adhesion-limited driving cases, the proposed method can make the drive torque distribution between the intermediate and rear axles more rational, effectively suppress excessive slip of the drive axles, and improve the smoothness of the vehicle longitudinal response.

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