



Article

Research on the Multi-Mode Composite Braking Control Strategy of Electric Wheel-Drive Multi-Axle Heavy-Duty Vehicles

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Abstract: Electric wheel-drive multi-axle heavy-duty vehicles have the characteristics of strong maneuverability and good passability, thereby they are widely used in heavy equipment transportation. However, current research on the composite braking of multi-axle heavy-duty vehicles is rare, which is not conducive to improving braking performance and braking energy utilization efficiency. This work proposes a multi-mode composite braking control strategy for the five-axle distributed electric wheel-drive heavy-duty vehicle. Firstly, given the differences in braking dynamics between two-axle vehicles and multi-axle vehicles, the brake dynamics characteristics of multi-axle vehicles are analyzed, and the vehicle dynamics model of multi-axle vehicles is constructed. Next, a multi-mode composite braking control strategy including a fully electric braking state and hybrid electro-hydraulic braking state is proposed in order to improve the braking energy recovery and braking stability. Finally, a hardware-in-the-loop simulation system is established, and the single-braking conditions and China heavy-duty commercial vehicle test cycle-heavy truck (abbreviated as CHTC-HT) are conducted to verify the performance of the braking control strategy. The results indicate that the recaptured braking energy and braking stability are significantly increased by applying the control strategy proposed in this work.

Keywords: multi-axle heavy-duty vehicle; composite braking; electric wheel-drive; control strategy; regenerative braking; braking force distribution



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1. Introduction

With the rapid development of technology and the economy in recent years, the transportation task of heavy equipment with high tonnage and large volume is also growing [1]; however, due to the limitation on axle load of road vehicles by highway regulations, the transportation of heavy equipment must be undertaken by multi-axle vehicles, so the demand for multi-axle heavy-duty vehicles has also increased gradually in recent years. Nevertheless, the mechanical drive systems usually used in traditional heavy-duty vehicles are complex, have low reliability, and incur high costs, which limits the development of multi-axle drive technology for heavy-duty vehicles [2] and indirectly hinders the development and application of heavy equipment. The electric wheel integrates the drive motor, reducer, and brake into the rim space, greatly reducing the complexity of the drive system and providing a convenient vehicle layout, as well as allowing for driving of the vehicle more efficiently by using the distributed drive mode. Hence, use of the electric wheel as the driving device of multi-axle heavy-duty vehicles has gradually become a hotspot topic in recent years. In addition, owing to the fact the electric wheel can recover part of the braking energy by the motor regenerative braking function, the electric wheel can not only improve vehicle energy utilization but also reduce the friction brake frequency, thus improving the

lifespan of the friction brake. Therefore, composite braking control has become a research focus in the field of electric wheel-driven multi-axle heavy-duty vehicles.

Different from the common two-axle vehicles, the braking system of multi-axle vehicles consists of multiple brakes, and its braking control is more complex than that of two-axle vehicles [3]; therefore, current research on multi-axle vehicle braking is mainly divided into three aspects: modeling and analysis of braking performance of multi-axle vehicles; braking response characteristics of multi-axle vehicles; and anti-lock control of multi-axle vehicle braking, etc., which are as follows:

(a) Research on modeling and analysis of braking performance of multi-axle vehicles.

Unlike the mature theories on the braking performance of two-axle vehicles, current research on the modeling and analysis of the braking performance of multi-axle vehicles remains limited. In the existing research literature, Liu Zhiqiang et al. [4] have conducted an experimental study on a multi-axle all-terrain truck crane, elucidating the influencing factors in the braking process and the special characteristics of the braking performance of all-terrain vehicles. Wang Zhiwang et al. [5] have established a braking dynamics model for a triaxial mid-sized truck, designed the main parameters of the braking system, and tested the vehicle's braking performance by employing a roller counterforce braking test rig under the relevant braking regulations, thus verifying the correctness of the model.

(b) Braking response characteristics of multi-axle vehicles.

Hou Xianxiao [6] has analyzed the performance of air-hydraulic hybrid braking systems by simulating the air-hydraulic hybrid braking system and valves of a four-axle heavy-duty vehicle. Wang Zhe [7] has constructed a test rig for the pneumatic braking system of an eight-axle vehicle and analyzed the influence of brake pedal stroke and speed on the hysteresis characteristics of the dual-braking circuit. Bartlett W D [8] investigated the braking response characteristics of a pneumatic braking system on a four-axle heavy truck during the road test. Chen Shanshan [9] has built a combined braking system model for a semi-trailer heavy equipment transporter, including a service brake and a hydraulic retarder, analyzed the downhill braking characteristics and the braking directional stability, and finally proposed a scheme that can improve the braking delay for multi-axle heavy trucks.

(c) Anti-lock control of brakes on multi-axle vehicles.

He Xinglei [10] has built a single-wheel brake ABS control model for multi-axle heavy-duty vehicles, compared various brake ABS control strategies such as logic threshold control, PID control, and sliding-mode variable-structure control by adopting the model, and finally found that nonlinear self-immobilization control is more suitable for the brake control of heavy-duty multi-axle special vehicles. Zhang Wei [11] has proposed an improved feedback control algorithm for the ABS switching valves of multi-axle vehicles by considering the axle load transfer of braking, and also formulated an optimal slip-rate identification method based on fuzzy logic, which improved the performance of ABS for multi-axle vehicles. Wang Dengfeng [12] has established a simulation model of a pneumatic ABS system for multi-axle heavy-duty vehicles by utilizing Matlab/Stateflow, and also developed an ABS control strategy under various road conditions by combining this with predictive control technology.

For electric vehicle composite braking, some research focuses on the braking energy recovery rate. Andrew Pennycott et al. [13] have designed an offline regenerative braking control algorithm for four-wheel independent-drive electric vehicles by using the European braking regulations as constraints, thereby maximizing the braking energy recovery. Yan Yunbing et al. [14] have designed a regenerative braking control strategy for dual-motor-drive vehicles by using the fuzzy algorithm, which effectively improved the braking energy recovery rate. Tonglie Wu [15] has proposed a regenerative braking control strategy based on a genetic algorithm for dual-motor all-wheel-drive electric vehicles, which effectively improved the braking energy recovery rate and braking stability. Meanwhile, other research focuses on brake stability control. Yu Deliang et al. [16] have designed an anti-lock composite braking control strategy by using an improved adaptive sliding-mode controller,

thereby solving the braking stability problems caused by the tremor in the process of composite anti-lock braking. Yu Yang et al. [17] have designed an advanced emergency braking system controller for hub-motor four-wheel independently driven vehicles by utilizing the regenerative braking regulation of the hub motor to ensure a vehicle's braking stability under high and low coefficients of adhesion.

However, the composite braking systems of electric wheel-drive multi-axle heavy-duty vehicles consist of multiple hub motors with regenerative braking function and multiple friction brakes, and their control strategy needs to consider the energy recovery of the motor regenerative braking as well as the braking force coordination distribution among different axles and different wheels. Consequently, the composite braking control strategy of electric wheel-drive multi-axle heavy-duty vehicles has characteristics such as multiple modes and complexity. In the study of composite braking for electric wheel-drive heavy-duty vehicles, Xingjing Song [18] has studied the parallel braking control strategy of a five-axle electric-wheel mining vehicle by using the braking intensity as the basis for the distribution of regenerative braking force and friction braking force. However, the load transfer caused by the different suspension stiffnesses of each axle was not considered in the braking process. Zhang Wei [19] has studied the regenerative braking control strategy of an electric-wheel mining vehicle under ramp braking conditions by comparing the four braking control strategies based on vehicle velocity, I-curve, P-line, and the maximum braking force of the front axle, but lacked an analysis of braking stability under the four braking control strategies. Hu Xingzhi [20] has investigated the electro-mechanical combined braking control system of an electric-wheel mining dump truck; nevertheless, the feedback braking power flow of asynchronous motors was mainly focused on, and specific characteristic analysis and control research on mechanical braking were lacking. Shen Yanhua [21] has proposed a rule-based electromechanical serial composite braking control strategy by considering the problem of axle load transfer for a multi-axle vehicle during braking, which improved the utilization rate of road adhesion and the braking energy recovery rate. Notwithstanding, the method used in this braking control strategy to reduce rear three-axis air pressure braking to improve the braking energy recovery rate will have a certain impact on braking stability and needs further improvement. Xuebo Li [22] has proposed a composite braking control strategy including segmental torque distribution and a dynamic coordination of electro-mechanical braking torque for a four-axle battery heavy-duty truck equipped with an EMB system, which effectively improved the braking energy recovery efficiency and braking stability. Regardless, this article simplifies the braking force analysis of the four-axle suspension into a three-axle suspension model; although this increases the calculation speed, it also affects the improvement of the braking energy recovery efficiency.

The above research on composite braking for electric wheel-drive multi-axle heavy-duty vehicles has achieved some results. However, it has not been able to fully exert the advantages of electric wheel multi-axle vehicles in terms of braking energy recovery rate and braking efficiency by combining the characteristics of multi-axle-distributed electric-drive vehicles. Therefore, this paper formulates a multi-mode composite braking control strategy for electric wheel-drive multi-axle heavy-duty vehicles by fully analyzing the characteristics of multi-motor braking and multi-axle braking force distribution. The structure of this paper is arranged as follows: in Section 2, the brake dynamics characteristics of multi-axle vehicles are analyzed, and the vehicle dynamics model of multi-axle vehicles is constructed; in Section 3, according to the brake dynamics characteristics of multi-axle vehicles, the multi-mode composite braking control strategy applicable to electric wheel-drive heavy-duty vehicles is proposed; in Section 4, the composite braking control strategy under the single braking condition and CHTC-HT cycle condition is carried out on the HIL simulation system to verify the performance of composite braking; Section 5 details the conclusions.

2. Analysis and Modeling of Brake Dynamics for Electric Wheel-Drive Multi-Axle Heavy-Duty Vehicles

2.1. Structure and Characterization of Composite Braking System for Electric Wheel-Drive Multi-Axle Heavy-Duty Vehicles

Different from the common two-axle-drive vehicles, as shown in Figure 1, the electric wheel-drive multi-axle heavy-duty vehicle studied in this paper is a five-axle vehicle in the 10×10 form, which is driven by 10 hub motors with a single-step gear. The composite braking system of this vehicle is composed of the hydraulic braking system and the motor braking system for each axle. The composite braking system can switch and distribute electric braking and hydraulic braking through the booster motor, push rod mechanism, and other components of the serial braking system. Therefore, the hydraulic braking torque of each axle is adjusted by the pressure regulator, and regenerative braking of the hub motor for each axle is adjusted by the motor controller, which receives the command of the composite braking control in VCU. The detailed parameters of the vehicle are shown in Table 1.

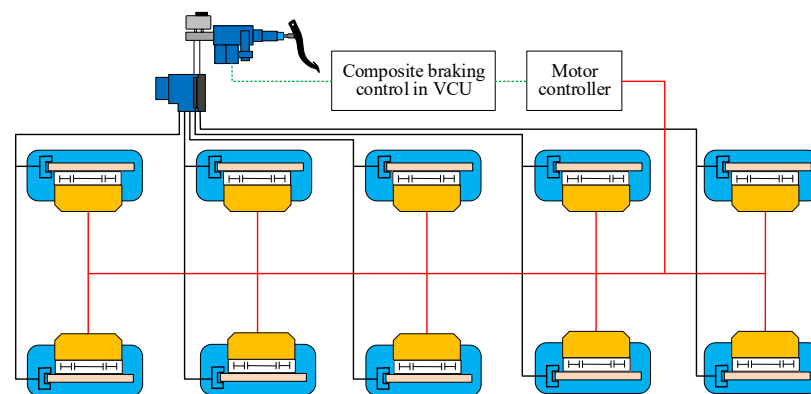


Figure 1. Structure of composite braking systems for electric wheel-driven multi-axle heavy-duty vehicles.

Table 1. Vehicle parameters and performance index.

	Parameter	Value	Unit
Vehicle parameters	Curb weight/ m_0	19,000	kg
	Max. weight/ M	47,000	kg
	Wind resistance coefficient/ C_d	0.29	-
	Windward area/ A	7.58	m^2
	Tire radius/ r	0.59	m
	Rolling resistance coefficient/ f	0.02	-
	Distance of each axle to 1st axis	0/3.3/6.7/9.1/11.5	m
	Max. velocity	90	km/h
Hub motor	Reduction ratio/ i_0	10.81	-
	Peak power	110	kW
	Rated power	80	N·m
	Max. torque	1100	N·m
	Peak speed	5000	$r \cdot min^{-1}$

2.2. Braking Force Analysis of Multi-Axle Vehicles

Compared with two-axle vehicles, the braking dynamics of multi-axle vehicles have changed, so it is necessary to analyze the braking dynamics in conjunction with the structural characteristics of multi-axle vehicles. For the super-static problem brought about by the multi-axle vehicle dynamics model, the braking force condition on the horizontal road surface with the adhesion coefficient φ is shown in Figure 2, under the conditions of

ignoring the rolling drag moment, air resistance, and inertial drag moment generated by the deceleration of the vehicle's rotating mass. In this paper, the equivalent stiffnesses of the suspension and tires are simplified to the vehicle spring stiffness, and a linear braking dynamics model of a multi-axle vehicle coordinated with suspension deformation is established [23], which can accurately analyze the braking dynamics of multi-axle vehicles.

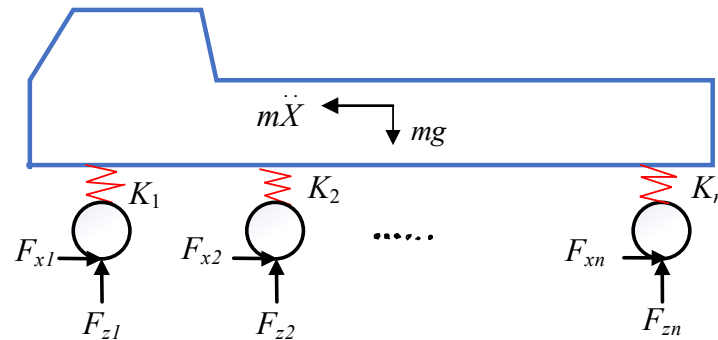


Figure 2. Dynamic model considering the coordination of suspension deformation.

The equilibrium equation for the normal reaction force on each axle of an n -axle vehicle ($n \geq 3$) braking on a horizontal road surface is:

$$\sum_{i=1}^n F_{zi} = mg \quad (1)$$

where F_{zi} is the ground normal reaction force in each axis in N·m; m is the mass of the vehicle in kg; and g is the gravitational acceleration in m/s^2 .

Taking moments on the centerline of the 1-axis yields an independent moment balance equation:

$$\begin{cases} \sum_{i=1}^n F_{zi} l_i = mgl_c - m\ddot{X}h_g \\ F_{zi} = k_i \Delta z_i, i = 1, 2, \dots, n \end{cases} \quad (2)$$

where l_i is the distance from the centerline of i -axis to the centerline of 1-axis in m ; l_c is the horizontal distance from the vehicle mass center to the centerline of 1-axis in m ; h_g is the height from the vehicle mass center to the ground in m ; $\ddot{X}$ is the braking deceleration in m/s^2 ; k_i is the equivalent stiffness of each axle suspension in N/m ; Δz_i is the deformation of each axle suspension in m .

The deformation coordination equations for $n - 2$ independent suspensions are obtained from the linear constraints in Figure 3:

$$\begin{cases} (l_2 - l_1) (\Delta z_j - \Delta z_1) = (l_j - l_1) (\Delta z_2 - \Delta z_1) \\ \Delta z_j = \Delta z_i \\ l_j = l_i \quad j = 3, 4, \dots, n \end{cases} \quad (3)$$

By coupling the above equations, the normal reaction force at any braking deceleration can be solved as follows:

$$F_{zi} = D_i \ddot{X} + C_i \quad (4)$$

where the parameters D_i and C_i are as follows:

$$D_i = \frac{mh_g k_i \sum_{j=1}^n k_j (l_j - l_i)}{\sum_{j=1}^n \sum_{m=1}^n k_j k_m (l_2 - l_j) (l_m - l_j)}$$

$$C_i = \frac{mgk_i \sum_{j=1}^n k_j(l_j - l_i)(l_j - l_c)}{\sum_{j=1}^n \sum_{m=1}^n k_j k_m (l_2 - l_j)(l_m - l_j)}$$

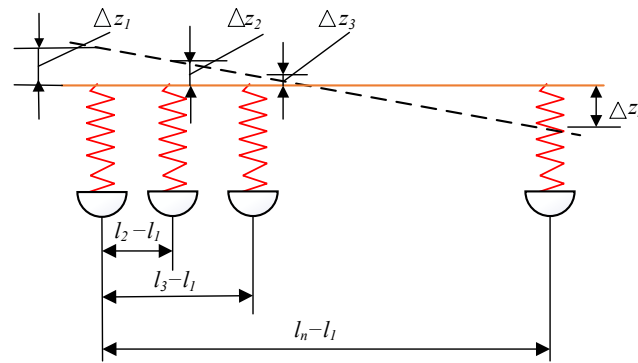


Figure 3. Linear constraint analysis of multi-axle vehicle suspension.

On this basis, the maximum ground braking force for each axis determined by the ground adhesion capacity is derived as follows:

$$F_{xbmax_i} = \varphi \cdot F_{zi} \quad (5)$$

where F_{xbmax_i} is the maximum ground braking force for each axle in N, and φ is the road adhesion coefficient.

It is known from automobile theory that, for multi-axle vehicles, when the braking force assigned by each axle brake is less than its corresponding maximum ground braking force, the ground braking force of each axle is equal to the braking force assigned by each axle brake; furthermore, when the braking force assigned by each axle brake is greater than or equal to its corresponding maximum ground braking force, the ground braking force of each axle will no longer be increased after the maximum ground braking force is reached due to the limitation of the ground's maximum adhesion, which is shown as follows:

$$F_{xbi} = \begin{cases} F_{bdi}, & F_{bdi} < F_{xbmax_i} \\ F_{xbmax_i}, & F_{bdi} \geq F_{xbmax_i} \end{cases} \quad (6)$$

where F_{xbi} is the ground braking force for each axle in N; F_{bdi} is the braking force distributed by each axle brake in N.

2.3. Motor Braking Characteristics

For motor braking, the motor can be in a state of regenerative braking or energy consumption braking, so its motor braking torque cannot exceed the maximum torque limited by the generated power at the current rotational speed. When the motor speed is below the base speed, the external characteristics of the motor braking torque can maintain a constant torque, and a constant power will be maintained when the motor speed is above the base speed, which should satisfy the following relationship:

$$T_{reg0} = \begin{cases} T_{max} & n \leq n_b \\ 9550P_{max}/n & n > n_b \end{cases} \quad (7)$$

where T_{reg0} is the motor braking torque in N·m; T_{max} is the maximum braking torque of the motor in N·m; P_{max} is the power of the motor in kW; n_b is the base speed of the motor in r/min; and n is the speed of the motor in r/min.

From analysis of the motor drag theory, it can be understood that when the motor speed is too low for braking, the motor's reverse electromotive force will be sharply reduced,

and the motor needs to be in a state of energy-consuming braking to maintain the demand for regenerative braking torque. In the energy-consuming braking state, the motor will not only fail to recover energy, but also will consume electrical energy. Therefore, reasonable selection of a regenerative braking exit point for low-speed motors will be able to effectively reduce the consumption of electric energy in the regenerative braking process and improve the vehicle's economy [24]. Therefore, to improve the economy of motor regenerative braking in the low-speed state, a specific low speed is often set as the motor braking speed threshold, and regenerative braking is withdrawn when the motor speed is lower than this threshold. In the specific control of the vehicle, the motor braking speed threshold is usually converted into the corresponding vehicle velocity, so it is necessary to make the following corrections to the motor regenerative braking torque:

$$T_{\text{reg}} = \lambda(v)T_{\text{reg}0} \quad (8)$$

where T_{reg} is the corrected motor regenerative braking torque in N·m; $\lambda(v)$ is the correction factor related to the vehicle velocity (a linear relationship with the motor speed); for $0 \leq \lambda(v) < 1$, $\lambda(v) = 0$ represents the exit from the motor regenerative braking, $0 < \lambda(v) < 1$ is the regenerative braking coefficient when motor gradually exiting to prevent switching fluctuations, and $\lambda(v) = 1$ represents only motor regenerative braking. In this paper, we set the following:

$$\lambda(v) = \begin{cases} 0, & v < v_{01} \\ 0.2(v - 5), & v_{01} \leq v < v_{02} \\ 1, & v \geq v_{02} \end{cases} \quad (9)$$

where v_{01} and v_{02} are the set speed thresholds in km/h, respectively. $v_{01} = 5$ and $v_{02} = 10$ are used in this paper.

2.4. Battery Characteristics

The electric energy generated by motor regenerative braking is finally stored in the battery. The maximum charging current, maximum charging power, and the state of charge (SOC) of the battery will affect the efficiency of the regenerative braking energy; therefore, in order to ensure the performance of regenerative braking, the charging current and charging power of regenerative braking should be limited to the maximum permissible charging current and charging power of the battery, and the threshold value of the battery SOC should be set to avoid overcharging, which would adversely affect the life of the battery [25]. Therefore, the battery should comply with the following conditions during regenerative braking:

$$\begin{cases} P_{\text{bat}} \leq P_{\text{batmax}} = (U_{oc} + I_0 e^{-\sigma t}) I_0 e^{-\sigma t} \\ I_{\text{reg}} \leq I_{\text{regmax}} \\ \text{SOC} \leq 90\% \end{cases} \quad (10)$$

where P_{bat} and P_{batmax} are the regenerative braking charging power and the maximum battery charging power in kW, respectively; U_{oc} is the open-circuit voltage that is a function of the battery SOC in V; R is the internal resistance of the battery in Ω ; I_0 is the maximum initial charging current at the moment of $t = 0$ in A; σ is the attenuation coefficient; and I_{reg} and I_{regmax} are the regenerative braking charging current and the maximum charging current of the battery in A, respectively.

3. Multi-Mode Composite Braking Control Strategy Based on Braking Dynamics of Electric Wheel-Drive Multi-Axle Heavy-Duty Vehicles

Compared with two-axle vehicles, the braking force adhesion state of each axis caused by load transfer during braking also changes. In addition, the hub motor braking force of the electric wheel can act on each axle independently. Hence, the braking dynamics of multi-

axle vehicles have changed a lot, thereby making the brake control of multi-axle heavy-duty vehicles more complex than of two-axle vehicles. Therefore, this section formulates a multi-mode composite braking control strategy for electric wheel-drive multi-axle heavy-duty vehicles.

3.1. Composite Braking Control Strategy Architecture and Process

Demand braking command comes from a driver's manipulation of the brake pedal. In earlier years, the total required braking torque was proportional to the brake pedal opening. However, with the increase in vehicle comfort requirements in recent years, the total required braking torque is generally made into be a function that varies jointly with the velocity and the pedal opening, as follows:

$$T_{breq} = f(v, \alpha_{bp}) \quad (11)$$

where T_{breq} is the total demand braking torque in N·m; α_{bp} is the brake pedal opening in %; and

$$T_{breq} = \sum_{i=1}^5 T_{bdi} \quad (12)$$

where T_{bdi} is the braking torque assigned to the i -th axle in N·m.

For an electric wheel composite braking system, the braking torque for each axle is borne by both the regenerative braking torque of the hub motor and the friction braking torque of the hydraulic brake, that is:

$$T_{bdi} = 2(T_{m_i} + T_{f_i}) \quad (13)$$

where T_{m_i} and T_{f_i} are the regenerative braking torque of a single hub motor and the friction braking torque of a single brake on the i -th axle in N·m, respectively;

According to the braking dynamics characteristics of multi-axle heavy-duty vehicles and braking characteristics of hub motors, this paper divides the braking control of electric wheel-drive multi-axle heavy-duty vehicles into a full electric braking state and a hybrid electro-hydraulic braking state. And the control strategies for each mode are formulated, which constitute the multi-mode composite braking control for electric wheel-drive multi-axle heavy-duty vehicles. As shown in Figure 4, the specific control strategy process is as follows.

- (1) Calculate the total required braking torque T_{breq} according to the vehicle velocity and brake pedal opening. Then, judge electric braking based on battery SOC. That is, when $SOC \geq 90\%$, the hub motor regenerative braking can not be activated because the battery's power is too high, and when $SOC < 90\%$, the electric braking can be activated, and enters the next step of judgment.
- (2) According to the current velocity, the maximum regenerative braking torque $T_{mreg_max_i}(v)$ of individual motors on each axle is calculated. Then, the total maximum braking torque of the motors is compared with the total required braking torque. When $T_{breq} \leq \sum_{i=1}^5 2 \cdot T_{mreg_max_i}(v)$, regenerative braking of the hub motors can satisfy the total required braking torque, and the vehicle enters into the electric braking state; correspondingly, when $T_{breq} > \sum_{i=1}^5 2 \cdot T_{mreg_max_i}(v)$, regenerative braking of the hub motors can not satisfy the total required braking torque that the deficient part needs to be supplemented by hydraulic braking, and the vehicle enters into the hybrid electro-hydraulic braking state.
- (3) Under the fully electric braking state, according to the instantaneous braking energy recovery power of motor braking, the regenerative braking force distribution ratio of each axle motor is optimized with the goal of optimal braking energy recovery, and this optimized ratio is used as the braking force distribution ratio of each axle to improve the braking energy recovery.

- (4) Under the hybrid electro-hydraulic braking state, the motor brake and hydraulic brake work together. In order to make full use of the ground adhesion, the braking torque distribution strategy based on the proportion of axle loads is adopted. The braking torque of each axle is preferentially distributed to the motor braking. The insufficient portion is supplemented by the hydraulic braking to achieve the purpose of improving the recovery rate of braking energy.
- (5) When the velocity decreases to the low-speed threshold of motor braking, in order to avoid the motor braking from regenerative braking to energy-consumption braking, the torque correction factor $\lambda(v)$ is introduced with reference to Formulas (8) and (9), and the motor braking is gradually withdrawn after reaching the velocity threshold, thereby improving the braking energy recovery efficiency at low velocities and enhancing the braking stability.

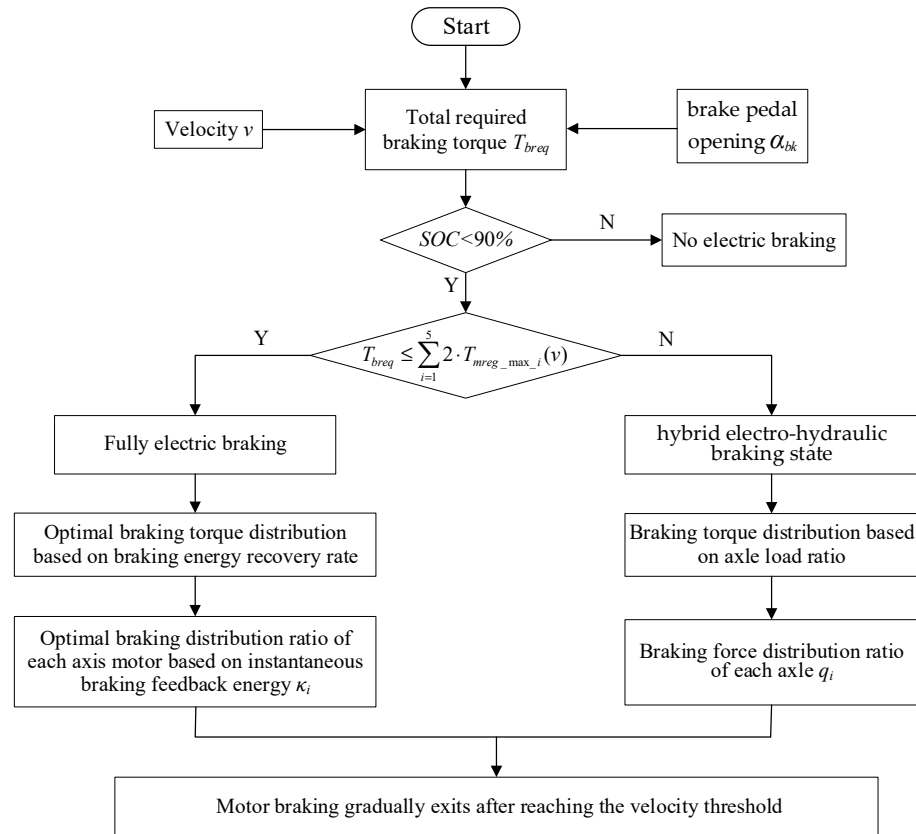


Figure 4. Flow of multi-mode composite braking control strategy for electric wheel-driven multi-axle heavy-duty vehicles.

3.2. Brake Force Distribution Strategy for Fully Electric Braking State Based on Optimal Instantaneous Brake Energy Recovery

Under fully electric braking, the total required braking torque is fully provided by regenerative braking of the hub motor. However, at the same rotational speed, the different torque loads assigned to the hub motor can cause differences in the motor efficiency, thus affecting the motor braking energy recovery efficiency [26]. Since the 10 in-wheel motors of this vehicle have the same specification, the overall motor regenerative braking efficiency can be changed by the different motor braking torque distribution on different axles, thereby affecting the overall braking energy recovery rate. Therefore, in the fully electric braking state, this paper uses the ratio of the motor braking torque of each axle to the total braking torque as the optimization variable, and the regenerative braking power as the objective function. Under the constraints of load, ground adhesion, and battery SOC, a real-time optimization algorithm for the motor braking torque distribution is applied to obtain maximum braking energy recovery.

3.2.1. Construction of Regenerative Braking Optimization Function Based on Instantaneous Braking Energy Recovery

Generally speaking, the high-efficiency zone of a vehicle's electric-drive system is mostly concentrated in the high-speed or high-torque zone. So, it can make the motor work point move to the high-speed and high-torque zone to improve the motor efficiency by optimizing motor torque distribution, ultimately improving the vehicle's economy. For multi-axle electric-drive vehicles, the efficiency of the electric-drive system can be improved to increase the motor braking load through the use of single-axle drive, two-axle drive, three-axle drive, four-axle drive, or five-axle drive modes, which ultimately saves energy.

In the fully electric braking state, the total required braking power is entirely provided by the hub motor braking, so the total braking power is as follows:

$$P_{reg} = 2 \cdot \sum_{i=1}^5 P_{m_i} \quad (14)$$

where P_{reg} is the total motor braking power in kW. P_{m_i} is the individual motor regenerative braking power in kW on the i -th axle.

$$P_{m_i} = T_{m_i} \omega_{m_i} \quad (15)$$

where ω_{m_i} is the motor rotational speed of the i -th axle after conversion to international standards in rad/s, namely

$$\omega_{m_i} = \frac{2\pi n_{m_i}}{60} \quad (16)$$

n_{m_i} is the rotational speed of the motor on the i -th axis in r/min , where

$$n_{m_i} = \frac{v_{i0}}{0.377r} \quad (17)$$

When using regenerative braking, the motor can be regarded to work as a generator, and the above regenerative braking power P_{m_i} is equivalent to the input power of the generator, which is equal to the sum of the various loss powers such as copper loss, iron loss, mechanical loss, and the power recovered to the battery. Therefore, the power ultimately recovered to the battery is as follows:

$$P_{breg_i} = P_{m_i} - P_{mloss_i} \quad (18)$$

where P_{breg_i} is the braking power recovered to the battery from the individual motor on the i -th axle in kW, and P_{mloss_i} is the power loss from the individual motor on the i -th axle in kW. For the convenience of control, this article uses a motor regenerative braking efficiency map to calculate the motor braking power. Currently, in most cases, the motor power at different speeds and torques is tested to make an efficiency map by using a motor dynamometer. Therefore, the braking power of the battery is as follows:

$$P_{breg_i} = T_{m_i} \omega_{m_i} \eta(T_{m_i}, \omega_{m_i}) \quad (19)$$

In this equation, $\eta(T_{m_i}, \omega_{m_i})$ is the individual motor efficiency of the i -th axle at the specific speed ω_{m_i} and the specific torque T_{m_i} , and is generally obtained by the motor calibration test. In this equation, the braking torque of each motor is allocated by the fully electric braking control strategy. If the ratio of the braking torque generated by braking of the i -th axle motor to the total required braking torque is set as κ_i , then the motor braking torque of the i -th axle is as follows:

$$T_{m_i} = \frac{\kappa_i T_{breq}}{2i_0} \quad (20)$$

where κ_i is the motor braking torque distribution coefficient of the i -th axle and, in essence, represents the ratio of the motor braking torque of the i -th axle to the total braking torque under the fully electric braking state.

The total braking power recovered by the battery is as follows:

$$P_{reg_bat} = 2 \cdot \sum_{i=1}^5 P_{breg_i} \quad (21)$$

Putting Equations (21)–(24) into Equation (25) and sorting, we can obtain the following:

$$P_{reg_bat}(\kappa) = 2 \cdot \sum_{i=1}^5 \left[\frac{\kappa_i \pi T_{breq} v}{22.62r} \cdot \eta \left(\frac{\kappa_i T_{breq}}{2i_0}, \frac{\pi i_0 v}{11.31r} \right) \right] \quad (22)$$

$P_{reg_bat}(\kappa)$ is the total braking power of all motors under κ_i in kW. In this equation, the required torque T_{breq} can be analyzed with the brake pedal opening, the velocity v can be measured, and the other parameters are constants; so, the braking energy recovered to the battery can be finally expressed as a function of the variant κ_i . Hence, by optimizing the torque distribution ratios, the motor working points will work in the high-efficiency area as much as possible, thereby improving the braking energy recovery rate. Therefore, in this paper, κ_i is taken as the optimization variable, and the braking power recovered to the battery shown in Equation (22) is taken as the optimization function.

3.2.2. Optimization Constraints

(a) Motor torque constraints

When optimizing the braking torque distribution, it should be ensured that the braking torque distributed to each motor does not exceed its maximum external characteristic curve limit of braking torque, otherwise, the motor will seriously heat up and even be damaged; so, the following constraints of the motor must be met:

$$\kappa_i T_{breq} / 2 \leq T_{maxm_i}(\omega_{m_i}) \quad (23)$$

where $T_{maxm_i}(\omega_{m_i})$ is the maximum external characteristic braking torque of the motor on the i -th axle at the speed ω_{m_i} in N·m.

(b) ECE regulations

In fully electric braking, the adhesion utilization coefficient of each axle wheel can be obtained from Equation (6):

$$\varphi_{bi} = \frac{F_{bi}}{F_{zi}} = \frac{2 \cdot i_0 \cdot T_{m_i}}{r F_{zi}} \quad (24)$$

Then, the power distribution of the motor braking on each axle should also be subject to the ECE regulations; that is, the following equation needs to be satisfied:

$$\begin{cases} \varphi_{bi} = \frac{2 \cdot i_0 \cdot T_{m_i}}{r F_{zi}} \leq \frac{z_{req} + 0.07}{0.085} \\ \varphi_{b1} \geq \varphi_{b2} \geq \varphi_{b3} \geq \varphi_{b4} \geq \varphi_{b5} \end{cases} \quad (25)$$

3.2.3. Optimization Models

Therefore, the optimization model can be finally expressed as follows:

$$\begin{cases} \max : & P_{reg_bat}(\kappa) = 2 \cdot \sum_{i=1}^5 \left[\frac{\kappa_i \pi T_{breq} v}{22.62r} \cdot \eta \left(\frac{\kappa_i T_{breq}}{2i_0}, \frac{\pi i_0 v}{11.31r} \right) \right] \\ \text{S.t. :} & \kappa_i T_{breq} / 2 \leq T_{maxm_i}(\omega_{m_i}) \\ & \frac{2 \cdot i_0 \cdot T_{m_i}}{r F_{zi}} \leq \frac{z_{req} + 0.07}{0.085} \\ & \varphi_{b1} \geq \varphi_{b2} \geq \varphi_{b3} \geq \varphi_{b4} \geq \varphi_{b5} \\ & SOC < 90\% \end{cases} \quad (26)$$

Since the above optimization model is a convex optimization problem with inequality constraints, the Lagrangian slack variable method is used to solve the optimization problem in this paper. Through the optimization above, the motor braking torque distribution coefficient of the i -th axle corresponding to the optimal instantaneous braking energy recovery power under different velocities and different required braking torques can be quickly obtained. The κ_i after optimization can maximize the braking energy recovery, thereby improving the vehicle's economy.

3.3. Braking Torque Distribution Strategy for Hybrid Electro-Hydraulic Braking State Based on Axle-Load Ratio

It can be seen from the external characteristic curve of the motor brake that when the velocity or required braking intensity increases to a certain value, the motor braking torque can not meet the total required braking torque. At this time, in order to ensure braking efficiency, the vehicle is in the hybrid electro-hydraulic braking state, and the braking torque is provided by the motor braking and hydraulic braking together. However, from the braking dynamics analysis for multi-axle vehicles in Section 2.2, it can be seen that the adhesion capacity of each axle changes with the change in vertical loads for each axle. In order to avoid the braking stability deteriorating as a result of the braking force being greater than the adhesion force on the same axle, the braking torque distribution strategy based on the proportion of the axle load was applied to the hybrid electro-hydraulic braking state in this paper. From Equation (4), the axle-load ratio of each axle when braking can be expressed as follows:

$$q_i = \frac{F_{zi}}{mg} = \frac{h_g k_i \sum_{j=1}^n k_j (l_j - l_i)}{g \sum_{j=1}^n \sum_{m=1}^n k_j k_m (l_2 - l_j)(l_m - l_j)} + \frac{k_i \sum_{j=1}^n k_j (l_j - l_i)(l_j - l_c)}{\sum_{j=1}^n \sum_{m=1}^n k_j k_m (l_2 - l_j)(l_m - l_j)} \quad (27)$$

where q_i is the axle load ratio of the i -th axle, that is, the ratio of normal reaction force on the i -th axle to the vehicle gravity. Therefore, the braking torque distribution of each axle is as follows:

$$T_{bi} = q_i \cdot T_{breq} \quad (28)$$

T_{bi} is the braking torque assigned to the i -th axle, in N·m. At this time, the braking force of each axle is supplied by the motor braking and the hydraulic braking:

$$T_{bi} = 2(T_{m_i} \cdot i_0 + T_{f_i}) \quad (29)$$

In the process of hybrid electro-hydraulic braking, the required braking torque on each axle is preferentially allocated to the motor braking to ensure braking energy recovery. Hence, the motor is firstly allowed to brake with the maximum regenerative braking torque at the current speed, and then the insufficient part is supplemented by the hydraulic braking. Therefore, the motor braking torque of each axle is allocated as follows:

$$\begin{cases} T_{m_i} = T_{mreq_max_i}(v), & q_i \cdot T_{breq} \geq 2 \cdot T_{mreq_max_i}(v) \\ T_{m_i} = q_i \cdot T_{breq} \cdot \lambda(v)/2, & q_i \cdot T_{breq} < 2 \cdot T_{mreq_max_i}(v) \end{cases} \quad (30)$$

Correspondingly, the hydraulic braking torque of each axle is as follows:

$$\begin{cases} T_{f_i} = q_i \cdot T_{breq}/2 - T_{m_i} \cdot i_0, & q_i \cdot T_{breq} \geq 2 \cdot T_{mreq_max_i}(v) \\ T_{f_i} = 0, & q_i \cdot T_{breq} < 2 \cdot T_{mreq_max_i}(v) \end{cases} \quad (31)$$

In addition, under hybrid electro-hydraulic braking state, the distribution of braking torque to each axle needs to comply with constraints such as the ECE regulations, battery SOC, etc., and if these constraints are not satisfied, the braking torques of each axle need to be adjusted accordingly.

4. Hardware-in-the-Loop Simulation and Analysis of Test Results

4.1. Hardware-in-the-Loop Simulation Platform

In order to verify the effectiveness of the proposed multi-mode composite braking control strategy for electric wheel-drive multi-axle heavy-duty vehicles, a hardware-in-the-loop simulation platform was established as shown in Figure 5. The dynamics model of 10×10 electric wheel-drive multi-axle heavy-duty vehicles was set up in the Host computer by using Matlab/Simulink, and the compiled real-time vehicle model was deployed to the Target computer real-time simulator through the ethernet to simulate various movements during vehicle operation. Meanwhile, a multi-mode composite braking control strategy was developed by using Simulink and integrated into the vehicle control strategy, which was deployed to the VCU (vehicle control unit) through rapid code generation technology. The VCU and real-time simulation machine transmit the control signals and vehicle status signals through the CAN bus. On this basis, a single-stop braking condition and cycle condition were selected to verify the performance of the multi-mode composite braking control strategy.

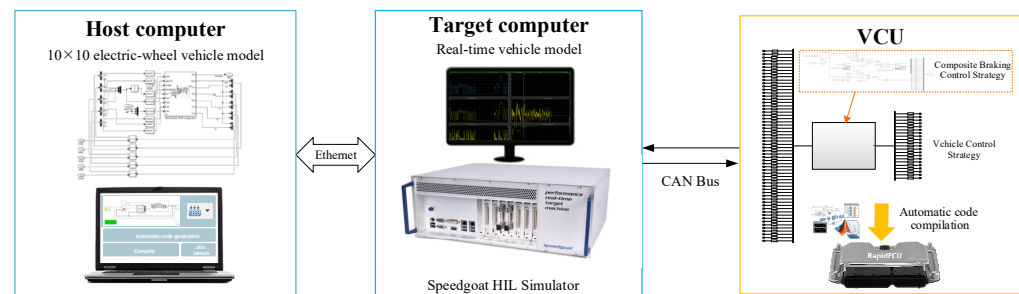


Figure 5. HIL simulation system.

After completing the hardware-in-the-loop system model construction and all configurations, the hardware among the Host computer, Target computer, and VCU are firstly connected by following the standard operating procedures, and the Simulink Realtime Explorer in the Host computer should be opened. Secondly, after opening the vehicle model in the Host computer, select the corresponding braking simulation conditions in conjunction with Sections 4.2 and 4.3 below, and then compile the model. Then, load the compiled model with the corresponding braking condition into the real-time simulation machine, and then run the program to simulate the braking condition while monitoring the braking performance parameters during the simulation operation. Finally, export the simulation parameters and analyze the braking performance results under this braking condition. After that, switch to other braking conditions in the sequence, and repeat the above steps to complete the hardware-in-the-loop simulation.

4.2. Simulation Results Analysis of the Single-Stop Braking Condition

The single-stop braking condition is a working condition that vehicles often encounter, and it can effectively characterize the vehicle's braking performance. To verify the effect of the proposed multi-mode composite braking control strategy in a single-stop braking condition, this paper selects two single braking stop conditions for HIL simulation, one for the fully electric braking state and one for the electric-hydraulic mixed braking state. Then, the data including the braking time, electric braking torque of each axle, hydraulic braking torque of each axle, and braking energy recovery under two single braking stop conditions were compared and analyzed, respectively. The initial vehicle velocity was set to 50 km/h for both conditions.

As shown in Figure 6, both braking conditions can decelerate smoothly from the initial velocity of 50 km/h until finally stopping at the velocity of 0 km/h, which proves that the multi-mode composite braking control strategy proposed in this paper can effectively and stably decelerate and brake. The braking time under the fully electric braking condition

is 17.2 s, and the braking time under the hybrid electro-hydraulic braking condition is 3.1 s. Comparing the changes in the braking torque and hydraulic braking torque of each axle motor under the fully electric braking condition in Figure 7a,b, it can be found that only the motor braking and hydraulic braking of the third axle are effective under the fully electric braking condition. And before the velocity decelerates to the motor braking threshold, only the motor braking of the third axle is effective and provides braking torque, which proves that only the third axle is relied on in the fully electric braking condition. That is because the motor braking of the third axle can meet the braking needs of the entire vehicle, and the motor efficiency is improved by increasing the motor braking load. After the velocity decelerates to the motor braking threshold, the motor braking of the third axle gradually decreases and finally exits; meanwhile, the hydraulic braking of the third axle gradually increases until the vehicle stops, which can avoid the motor braking entering the energy-consuming braking state at low speeds and increasing the energy consumption. Figure 8 shows the torque distribution of the hybrid electro-hydraulic braking condition. It can be seen from Figure 8a that regenerative braking is performed in motors on each axle. Between the time from 44.9 s to 46.8 s, the braking torque of every motor gradually increases to the maximum braking torque; correspondingly, the hydraulic braking torque in Figure 8b gradually decreases between 44.9 s and 46.8 s. This is due to the total required braking torque in this braking condition being too large, so the individual motor braking can no longer meet the total required braking torque. At this time, the motor brake works at the maximum braking torque, and the insufficient part of the required braking torque is provided by the hydraulic braking. After 46.8 s, the motor brake maintains the maximum braking torque within the range of base speed, with the hydraulic braking also maintaining a stable value. Then, the motor braking gradually decreases and exits after the vehicle velocity drops to the motor braking threshold, and the hydraulic braking continues to provide braking torque until the vehicle stops. During the entire hybrid electro-hydraulic braking process, the motors of each axle provide maximum braking torque within the range of regenerative braking capabilities, while the hydraulic braking of each axle supplements the corresponding insufficient part of required braking torque, ensuring appropriate braking performance in multi-axle heavy-duty vehicles.

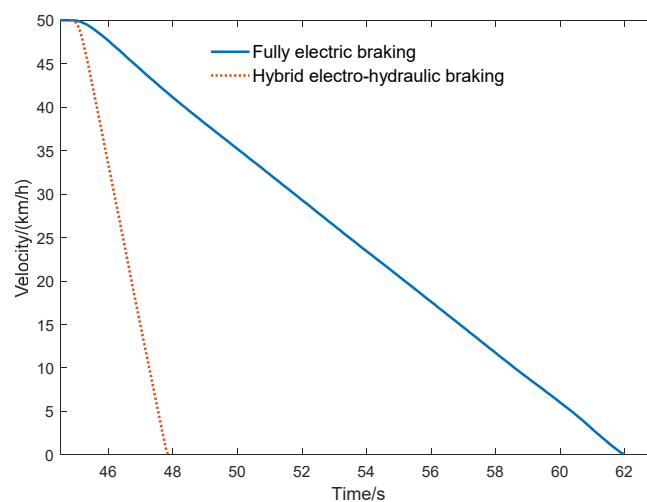


Figure 6. Velocities in single-stop braking condition.

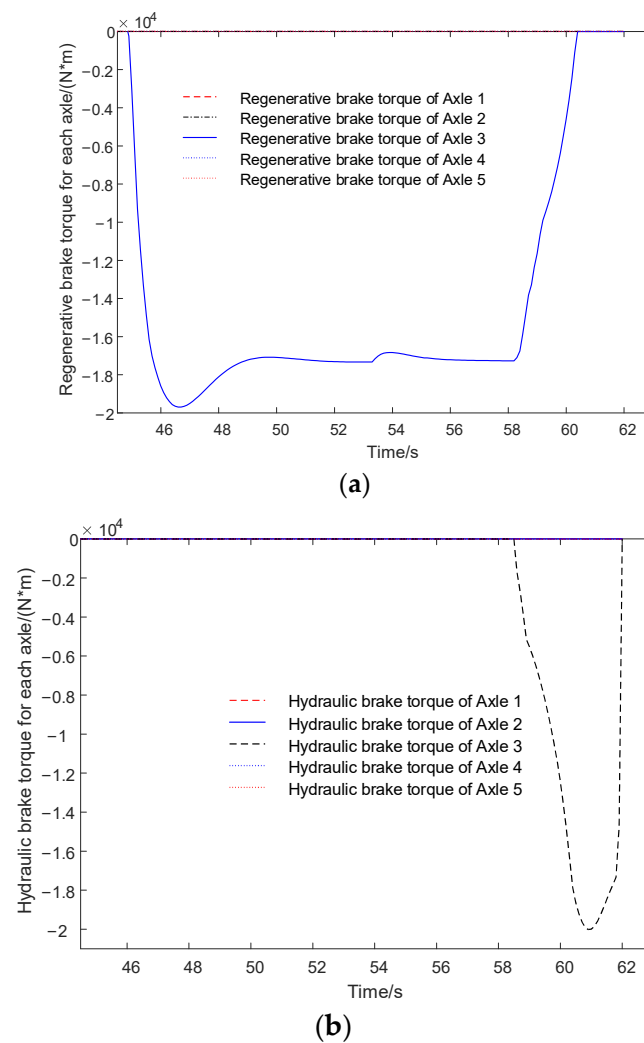


Figure 7. Torque distribution of fully electric braking condition. (a) Regenerative brake torque for each axle. (b) Hydraulic brake torque for each axle.

After analyzing the braking torque distribution under the two braking conditions, as well as Figure 9 and Table 2, it can be seen that part of the braking energy is recovered through the motor regenerative braking in the two braking conditions by using the composite braking control strategy in this article. The energy recovery rate reaches 39% in fully electric braking condition, which is higher than that in electro-hydraulic hybrid braking, which was an energy recovery rate of 27.5%. Therefore, the multi-mode composite braking control strategy proposed in this article works effectively in different types of single-stop braking conditions, and can improve the braking recovery energy and braking stability by optimizing the distribution of hub motor braking torque and hydraulic braking torque between each axle.

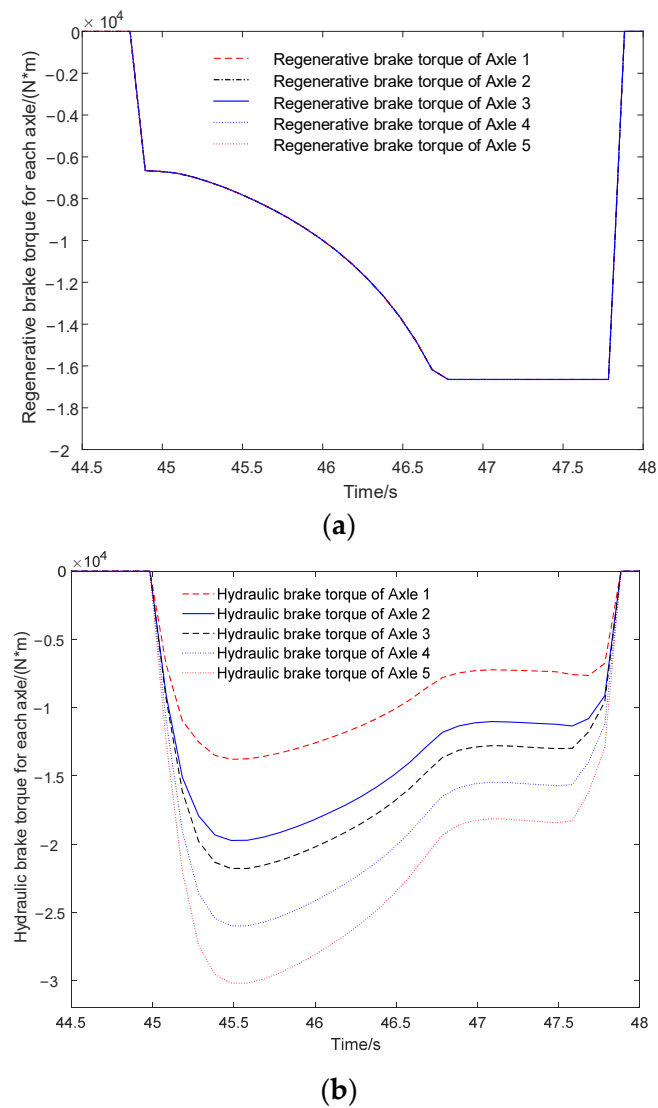


Figure 8. Torque distribution of hybrid electro-hydraulic braking condition. (a) Regenerative brake torque for each axle. (b) Hydraulic brake torque for each axle.

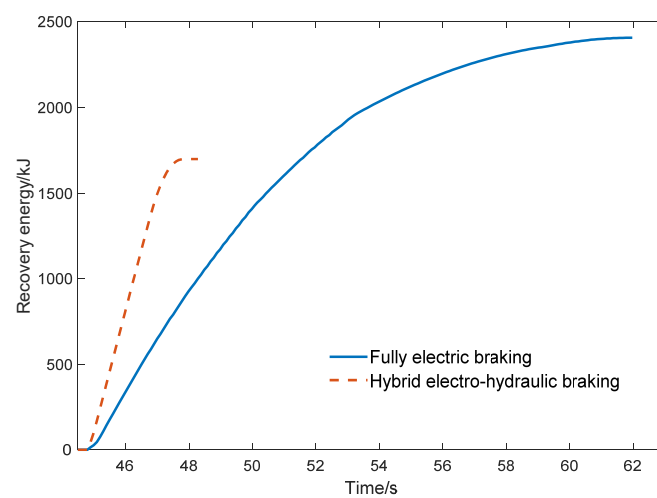


Figure 9. Recovery energy in single-stop braking condition.

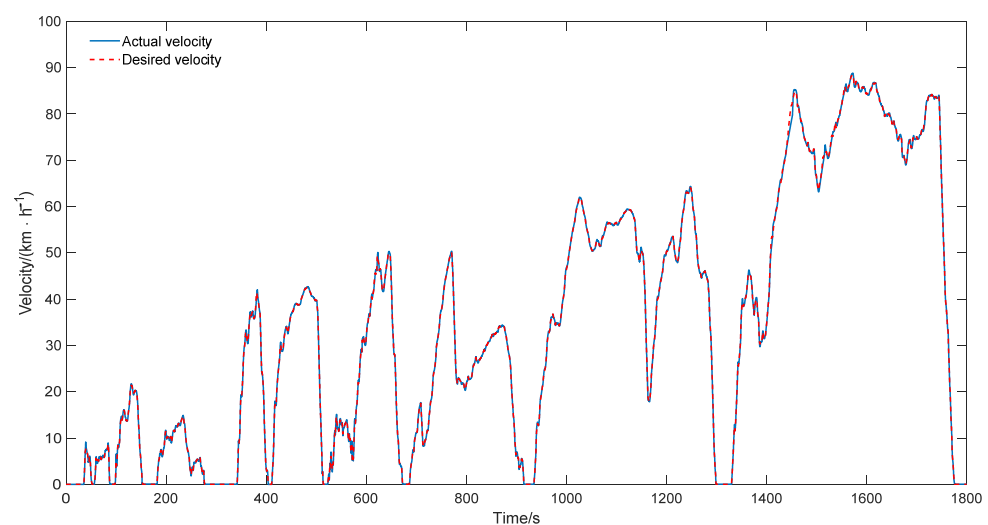
Table 2. Simulation results in single-stop braking conditions.

Braking Condition Performance	Electric Braking	Hybrid Electro-Hydraulic Braking
Braking time/s	17.2	3.1
Braking distance/m	121	23
Recovery energy/kJ	2407	1698
Braking energy consumption/kJ	6170	6170
Braking energy recovery rate/%	39	27.5

4.3. Simulation Results and Analysis of the Cycle Condition

The single-stop braking condition simulation can verify the braking performance of the multi-mode composite braking control strategy under the condition of braking to stop; however, in the actual driving process, there are quite a few scenarios where the vehicle only needs to decelerate, rather than braking to stop. Therefore, a single-stop brake condition alone cannot fully reflect the total braking performance of the vehicle. Therefore, this article selects the CHTC-HT as the cycle condition. The CHTC-HT includes low velocity, medium velocity, and high velocity, which can comprehensively and accurately reflect the performance of heavy-duty vehicles in China's road environment. In addition, in order to more intuitively illustrate the performance of the multi-mode composite braking control strategy proposed in this paper, a commonly used braking torque average distribution strategy is structured and applied as a comparison group in this part. The specific simulation results and analysis are as follows.

It can be seen from Figure 10 that after adopting the multi-mode composite braking control strategy in this article, the multi-axle heavy-duty vehicle can effectively brake according to the required vehicle speed in each braking segment of the CHTC-HT and has good braking-response ability. It can be seen from Figure 11a that in the different braking segments of the CHTC-HT, the motor braking torque of each axle is different. The main reason for this is that the motor braking torque distribution of each axle is optimized in the electric braking state to improve the motor braking efficiency, thereby improving the braking energy recovery rate during low braking demand. It can be seen from Figure 11b that the number of hydraulic braking actions is significantly less than that of motor braking, the duration of hydraulic braking actions is generally shorter, and most hydraulic braking actions occur at the end of the braking stop stage. This is mainly due to the fact that most braking decelerations in the CHTC-HT's operating conditions are not large, so most of the vehicle's braking is in the electric braking state, thus hydraulic braking is less.

**Figure 10.** Desired velocity and actual velocity in the CHTC-HT.

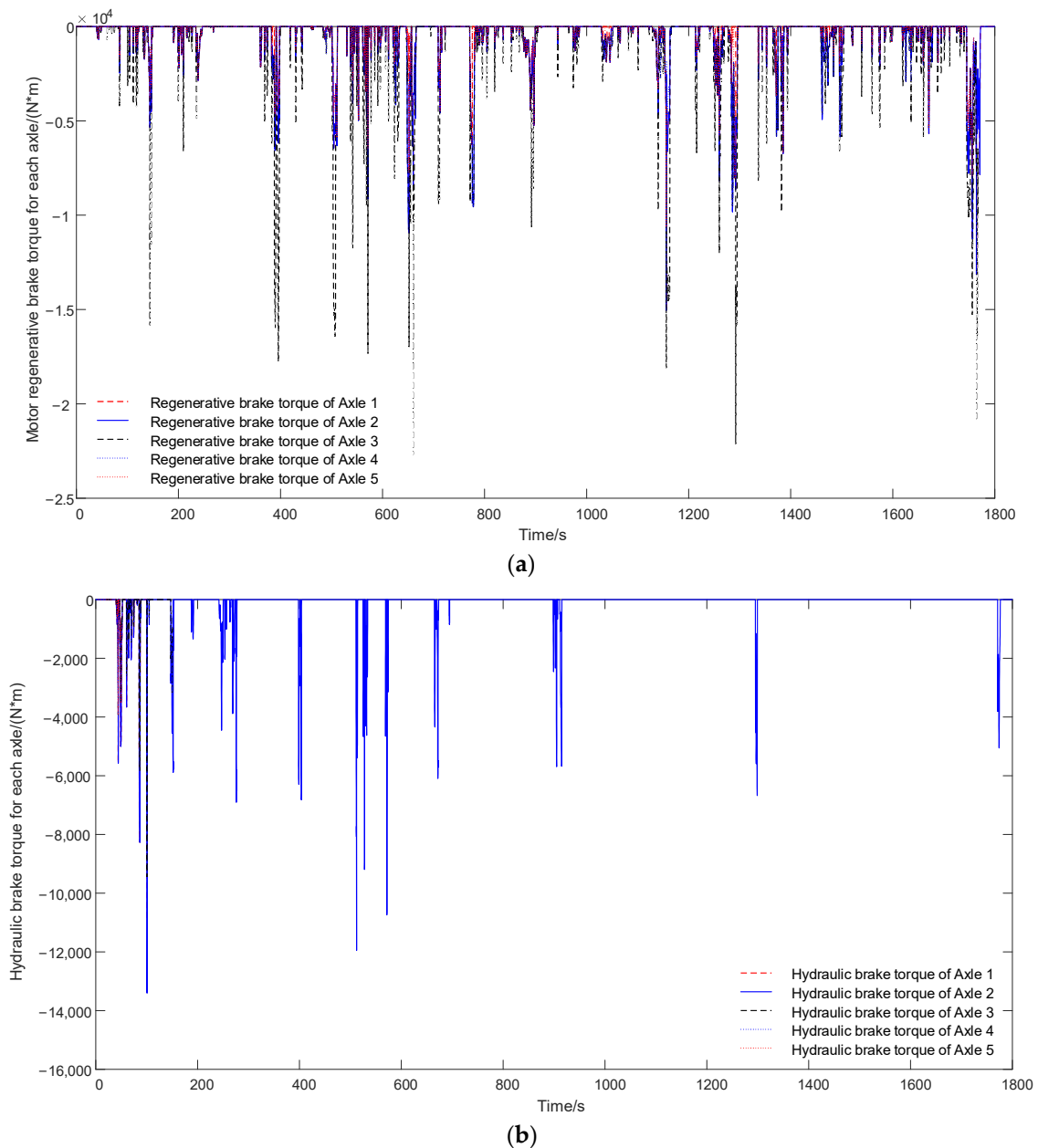


Figure 11. Braking torque of electric braking and hydraulic braking on each axle in the CHTC-HT. (a) Regenerative brake torque for each axle. (b) Hydraulic brake torque for each axle.

In order to verify the performance of braking energy recovery for the braking control strategy proposed in this paper, the SOC and recovery braking energy of this paper and a comparison group are compared in Figures 12 and 13, respectively. It can be seen from Figure 12 that the SOC of the two composite braking control strategies can both increase in certain braking segments, that is, braking energy is recovered at these segments. However, at the beginning stage of the CHTC-HT, the SOC realized by using the control strategy of this paper begins to exceed that of the comparison group's control strategy. Combined with the braking energy recovery at the corresponding time in Figure 13, it can be seen that under the same braking segments, the recovered braking energy resulting from using the control strategy in this article is higher than that of the comparison group. After the accumulation of multiple braking segments in the cycle condition, the SOC of the control strategy in this article was 35.36% at the end of the cycle condition, and the regenerative braking energy recovery rate was increased by 3.47% compared to the control strategy of

the comparison group form Table 3. Therefore, compared with the braking torque average distribution strategy, the energy recovered to the battery has increased by adopting the multi-mode composite braking control strategy proposed in this paper, which means that the braking energy recovery rate can improve in cycle conditions.

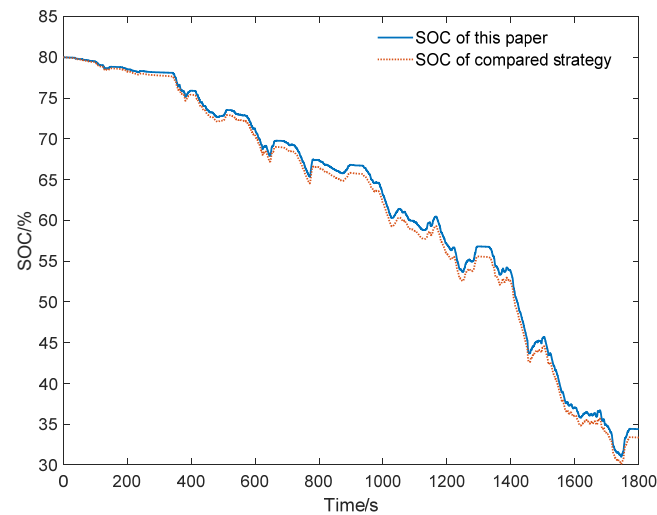


Figure 12. Comparison of SOC in CHTC-HT.

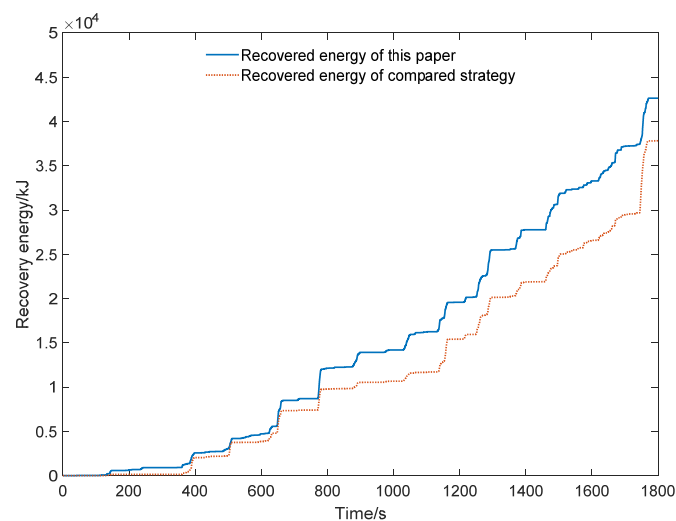


Figure 13. Comparison of recovery braking energy in CHTC-HT.

Table 3. Performance comparison of different braking control strategies in the CHTC-HT.

Braking Strategies		Control Strategy in This Paper	Comparison Strategy
Performance			
Driving distance/m		17,317	
Initial SOC/%		80	
Terminal SOC/%	35.36		34.39
Consumed energy/kJ		138,500	
Recovered braking energy/kJ	42,632		37,824
Braking energy recovery rate/%	30.78		27.31

5. Conclusions

The composite braking control of electric wheel-drive heavy-duty vehicles is of great significance for improving vehicle economy and braking stability. This paper focuses on the characteristic changes in the composite braking system of multi-axle heavy-duty

vehicles, and proposes a multi-mode composite braking control strategy by combining the battery changing characteristics and motor characteristics. This control strategy divides the composite braking into two states: a fully electric braking state and a hybrid electro-hydraulic braking state. Among them, the fully electric braking state focuses on improving vehicle economy, and so mainly optimizes the distribution of motor braking torque between each axle based on the instantaneous braking energy recovery, thereby improving the vehicle's braking energy recovery rate. The hybrid electro-hydraulic braking state focuses on ensuring vehicle braking stability, and mainly optimizes the braking torque distribution between each axle, as well as the torque coordination between the electric braking and hydraulic braking in the same axle based on the adhesion limit of each axle. Finally, through the hardware-in-the-loop simulation system, the multi-mode composite braking control strategy for electric wheel-drive multi-axle heavy-duty vehicles proposed in this paper was verified both in a single-stop braking condition and in the CHTC-HT, and a satisfactory braking energy recovery and a good braking performance were established. In addition, due to the large load mass, multi-axle heavy-duty vehicles mostly adopt a series hybrid configuration, and a vehicle can recover more braking energy by combining the multi-mode composite braking control strategy proposed in this article with the series hybrid control strategy by storing it in the power battery when the vehicle decelerates or brakes to stop, thereby increasing the vehicle's driving range. So, the multi-mode composite braking control strategy is of great significance in improving the economy of heavy-duty vehicles. However, the lateral dynamics of electric wheel-drive multi-axle heavy-duty vehicles during braking have a great impact on braking safety and stability, and will also become the focus of future research on the braking performance of distributed driven multi-axle vehicles.

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