Research on Anti-Lock Braking Control Strategy of Distributed-Driven Electric Vehicle

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This work was supported in part by the Heilongjiang Postdoctoral Foundation under Grant LBH-Q19107, and in part by the University Nursing Program for Young Scholars with Creative Talents in Heilongjiang Province under Grant UNPYSCT-2017097.

ABSTRACT Aiming at the problem of the stability of distributed-driven electric vehicles under braking conditions and the recovery of regenerative braking energy, a novel compound brake anti-lock braking control strategy based on wheel slip rate control is proposed. This article takes distributed-driven electric vehicles as the research object. From the perspective of the single-wheel braking dynamic model of the car, the method of designing a non-linear state observer is used to observe the changing longitudinal braking force of the wheels and then to achieve the estimation of the optimal slip rate. Using the improved adaptive sliding mode controller to achieve effective control of the optimal slip rate to improve the chattering problem of sliding mode control. According to the output of the controller, combined with the motor brake and hydraulic brake system of the car, a composite brake anti-lock control system is designed. The joint simulation is carried out in the environment of MATLAB and CARCISM. The simulation results show that the control method studied can effectively estimate and control the optimal slip rate under various complex road surfaces, and realize the safety and stability of electric vehicle brake control.

INDEX TERMS Distributed drive electric vehicles, optimal slip rate estimation, adaptive sliding mode controller, compound brake.

I. INTRODUCTION

With the increasingly serious problems of environmental pollution and energy crisis, electric vehicles have become a hot spot for research and development in countries around the world. Among them, the four-wheel-drive distributed electric vehicle simplifies the traditional mechanical gearbox components, and such electric vehicles usually use in-wheel motors or wheel-side motors to drive the car. It can be matched arbitrarily according to the demand, and the driving and braking torque can be controlled quickly, accurately and independently [1]–[5]. There are many advantages in vehicle chassis layout, energy saving optimization and dynamics control [6]. The electro-hydraulic composite braking system is composed of motor feedback braking and hydraulic braking. It is a key technology for electric vehicles to increase driving range and optimize braking safety performance. Due to the control advantages of distributed electric vehicles, it is an ideal vehicle for studying electro-hydraulic compound brake control.

The anti-lock braking system (ABS) in the automobile braking system is an important safety system in the case of emergency braking [7]. It can effectively prevent wheel skidding and reduce the braking distance. Especially on roads with low adhesion coefficients, such as icy and snowy roads, slipping caused by wheel locks will make the car lose its maneuverability and may cause serious safety accidents. The ABS system achieves the above functions by limiting the slip rate of the wheels [8], [9]. Electric vehicles can use hydraulic braking and motor regenerative braking to control the braking force during the working process of ABS. Among them, regenerative braking energy recovers braking energy for electric vehicles under the braking or deceleration conditions of electric vehicles. It is an important way to increase the mileage of electric vehicles. To ensure the safety and stability of the vehicle during braking is the prerequisite for
the work of the regenerative braking system. For this reason, it is necessary to study a safe and reasonable anti-lock braking system for automobile brakes. Since the motors of electric vehicles have the advantages of high control accuracy and fast response speed, the introduction of slip rate control in regenerative braking or compound braking control is a widely used method [10], [11]. For the optimal slip rate control of electric vehicles, the main problems are as follows: First, solve the problems about how accurately identify the pavement characteristics under various complex road conditions when the vehicle is decelerating. So as to realize the estimation of the optimal slip rate. Secondly, how to use the composite braking system of electric vehicles to realize the effective control of the constantly changing optimal slip rate.

Recent years, related scholars at home and abroad have conducted related researches on the above-mentioned problems. At present, the methods adopted in the recognition of pavement adhesion coefficient can be divided into two categories: environment-based and model-based [12]. Environment-based methods mainly use a variety of sensor fusion methods to estimate the road adhesion coefficient. This method is simple to control, but is limited by the recognition accuracy and cost of the sensor. The model-based method starts from the perspective of the electric vehicle dynamics model, and uses the existing vehicle state parameters to estimate the required road parameters. Due to the low cost, it is currently widely used.

Ren and Hai-Peng [13] and Ren and Mengqi [14] proposed an optimal slip rate control method based on a tire model. Based on the Burckhardt tire-road model, the adhesion coefficient curve of a typical road is used to estimate the adhesion coefficient curve of an unknown road surface, and then the peak road adhesion coefficient and the corresponding optimal slip rate are calculated, and PID control is used to control the motor torque during braking. Li et al. [15] analyzed the deterministic factors of road friction and used fuzzy control and signal fusion methods to realize road condition recognition. The method used does not require extra sensors. Erdogan et al. [16] proposed a method using tire piezoelectric sensors to estimate the friction between the wheel and the road, but there is a problem of higher cost. Regolin and Ferrara [17] analyzes the parameters of several different tire empirical formulas, then uses SVM technology to classify the road types, and finally uses the Kalman filtering method to improve the accuracy and realize the recognition of road conditions. Zhao et al. [18], considering the load change of front and rear wheels during braking, a two-wheeled vehicle dynamics model was established to estimate the change of adhesion coefficient and slip rate, and the sliding mode surface of the sliding mode controller was redesigned to eliminate the vibration problem of sliding mode controller.

There are also many studies on compound braking and braking force distribution. Pi et al. [19] proposed a brake pressure control algorithm for electric trucks that cooperates with the regenerative braking system to improve the stability and accuracy of braking. Xu et al. [20] aiming at the torque optimization control of electric vehicles with four in-wheel motors, use the MPC method to solve the multiple optimization problems of braking safety and energy recovery. Li et al. [21] designed an adaptive braking torque control scheme based on nonlinear tire friction estimation to adjust the wheel slip rate of the front and rear wheels at the same time. Jin et al. [22] used a sliding mode controller and an electro-hydraulic braking torque distribution system based on mode switching to realize the coordinated distribution of four-wheel hydraulic braking and feedback braking torque.

This article designs a new type of braking power distribution for road condition recognition and electro-hydraulic compound braking. The slip rate control compound brake anti-lock control system in the current distributed driving electric vehicle emergency braking process improves the vehicle’s performance. The performance of braking stability and maneuverability during emergency braking. Its main contributions include the following three aspects. (1) In order to obtain vehicle status information, a single-wheel dynamics model of electric vehicles is established, and relevant factors affecting vehicle braking are considered, and the road condition recognition method based on empirical formulas commonly used in the current research is analyzed, and a non-standard method is designed. The linear state observer has realized the observation of the braking force, and then estimated the optimal slip rate of the road, avoiding the use of additional sensors. (2) This design guarantees the accuracy of slip rate control and the anti-interference ability of the system. The design of an improved adaptive sliding mode controller controls the output torque and optimizes the control accuracy and vibration problems of the sliding mode controller. (3) By considering the external characteristic curve of the motor, to ensure fast and accurate torque control and recovery of regenerative braking energy. The composite brake control system based on regenerative braking and hydraulic braking is designed and used for MATLAB/CARSIM co-simulation on constantly changing complex roads. Finally verify the effectiveness of the above system.

The rest of this article is organized as follows. Section II analyzes the automobile single-wheel model and tire empirical formula, and designs the optimal slip rate recognition system. Section III introduces the improved sliding mode controller in detail, which has realized the stable control of braking torque. Section IV introduces the compound brake control system designed in this article. In Section V, based on CARSIM and MATLAB/SIMULINK, under a variety of complex road conditions, simulation verification and result analysis of the control system designed in this article are carried out. Finally, Section VI is the summary of this research and future prospects.

II. ROAD STATE ESTIMATION

A. VEHICLE DYNAMICS MODEL

First, analyze the dynamic model of the car when braking. Ignore the rolling resistance and air resistance of automobile wheels, ignore the influence of steering system...
and suspension system on vehicle driving, and analyze the single-wheel rotation model of the automobile under braking conditions. The schematic diagram is shown in Fig.1.

![FIGURE 1. The single wheel-rotating model of a car.](image)

It can be seen from the Fig.1 that two forces mainly affect the vehicle during braking. One is \( T_b \) that directly decelerates the wheels, which is applied to the tires through the motor regenerative braking and hydraulic braking systems, and the other is the reverse friction force \( F_z \) formed by the friction between the tire and the ground, which is applied to the tire by the road surface and slows down the entire car. \( \omega \) represents the wheel angular velocity, and \( V_x \) represents the vehicle speed. The following vehicle braking dynamics equation can express the relationship between them:

\[
\dot{\omega}J = F_xR - T_b \\
F_x = \mu F_z \\
T_b = T_m + \frac{k_cP_c}{R} \\
F_x = -m\ddot{V}
\]

In the formula, \( J \) is the moment of inertia of the wheel, \( F_x \) is the longitudinal adhesion between the tire and the ground, \( T_b \) is the wheel braking torque, \( T_m \) is the torque applied to the wheel end by the motor, and \( P_c \) is the wheel cylinder pressure of the brake system, \( k_c \) is the conversion coefficient between wheel cylinder pressure and braking torque, and \( m \) is the total mass of the car.

**B. THE RELATIONSHIP BETWEEN TIRE AND ROAD SURFACE**

It can be seen from the above relationship that if the total mass of the car does not change, in the braking process, if you want to obtain a greater deceleration, you need to apply a greater reverse force on the tire. At this time, the control input of the tire is the braking torque \( T_b \). With the increase of \( T_b \), the wheel of the car approaches the locked state from rolling. However, excessive input braking torque will eventually lock the wheels completely. At this time, not only the maximum ground braking force cannot be obtained, but the locking of the tires will also make the car lose its steering ability. Especially when braking on a low-adhesion road, if the rear wheel locks, the car will braking skid, which will seriously endanger the safety of the car.

During the acceleration or braking of a car, the relationship between wheel speed and vehicle speed is generally characterized in the form of slip rate [11], and its definition is as follows:

\[
\lambda = \begin{cases} 
\frac{V - \omega R}{\omega R}, & V \neq 0, \text{ braking condition} \\
\frac{V}{\omega R}, & \omega \neq 0, \text{ acceleration condition}
\end{cases}
\]

Fig.2 shows the \( \mu - \lambda \) relationship diagram under typical road conditions based on the Burckhardt model. It can be seen from the figure that as the vehicle wheel slip rate increases, the road adhesion coefficient \( \mu \) first increases and then decreases. When the slip rate is between 0.1 and 0.2, the peak road adhesion coefficient \( \mu_{max} \) can be obtained. The peak adhesion coefficient can obtain the maximum road adhesion.
C. ESTIMATION OF ROAD CONDITIONS

For the road surface identification problem, scholars at home and abroad have conducted extensive research. The focus of this chapter is how to determine the peak pavement adhesion coefficient $\mu_{\text{max}}$ and its corresponding optimal slip rate $\lambda_{\text{opt}}$ in the relationship curve of slip rate and adhesion coefficient. In response to this problem, most of the current researches are based on various tire-road empirical formulas to classify the current road conditions and find the peak point by looking up the table. This type of method is only suitable for road recognition under a single road condition or several typical road conditions.

Li et al. [23] and Tanelli et al. [24] proposed a method of identifying the peak road adhesion coefficient combining the Kiencke tire model and the recursive least square method. First, the functional formula of the tire model is given as follows:

$$\mu(\lambda) = \frac{30\lambda}{1 + p_1\lambda + p_2\lambda^2} \quad (7)$$

By deriving the above formula, it can be concluded that the point where the slope is zero is the peak point of the adhesion coefficient. When the car is driving, collect the real-time slip rate and the corresponding road adhesion coefficient, and use the recursive least square method to identify the parameters $p_1$ and $p_2$ in the relationship equation online to obtain the peak road adhesion coefficient $\mu_{\text{max}}$ under current road conditions. Fig.3 shows the recognition effect of this method on wet asphalt pavement. It can be seen that with the iteration of the data, the peak adhesion coefficient and corresponding slip rate of the pavement are successfully identified in 0.1 seconds.

![FIGURE 3. Recursive least square method road surface recognition results.](image)

The above-mentioned road condition recognition methods based on the empirical tire formula or the least square method on-line recognition have two problems. First, only a single road condition can be recognized, and the actual car often faces complex road conditions when driving. Second, the above methods all require the input of real-time slip rate and road adhesion coefficient. Generally, the road adhesion coefficient is derived from the relationship between the current tire braking friction $F_x$ and the vertical load $F_z$.

However, in actual working conditions, the vertical load $F_z$ is affected by braking strength and vehicle deceleration, and braking force $F_x$ is difficult to be directly measured. The actual vertical load of the front and rear wheels during the braking process of a car can be described by the following formula about longitudinal acceleration:

$$\begin{align*}
F_x &= \frac{M(L_2g - ha_x)}{L_f + L_f} \\
F_z &= \frac{M(L_2g + ha_x)}{L_f + L_f}
\end{align*} \quad (8)$$

where $L_f$ and $L_r$ are the longitudinal distance from the center of gravity to the front and rear tires, $g$ is the acceleration due to gravity, $h$ is the distance from the center of gravity to the vehicle ground, $a_x$ is the longitudinal acceleration of the vehicle, and $M$ is the total weight of the vehicle.

Therefore, in view of the actual problems in car driving under complex road conditions, without using additional sensors, a good method for parameter estimation is to design a nonlinear observer [25]–[27]. Taking the wheel-end braking force $F_x$ that cannot be directly measured as the research object, because of the changes in the road adhesion coefficient $\mu$ and the vertical load $F_z$, the braking force $F_x$ can be characterized by the following formula:

$$F_x = f(\mu, F_z) \quad (9)$$

According to (1) and (2), the following state equation can be established:

$$\begin{align*}
\dot{X}_1 &= AX_2 + BU \\
y &= X_1 \\
X_1 &= \omega \\
X_2 &= F_x = f(m, F_z) \\
U &= T_b
\end{align*} \quad (10)$$

Among them, the output $Y$ of the observer is the state to be observed $F_x$, which is regarded as a system disturbance, and the following nonlinear extended state observer is designed:

$$\begin{align*}
\dot{e} &= x_1 - y \\
\dot{x}_1 &= \dot{x}_2 - \alpha_1g_1(e) + bu \\
\dot{x}_2 &= -\alpha_2g_2(e)
\end{align*} \quad (11)$$

In the formula, $\alpha_1$ and $\alpha_2$ are constants greater than zero. In the design of the observer, since the state of the system state $x_2$ is unknown, we assume that it is a disturbance function. By artificially introducing the functions $g_1$ and $g_2$ of the error $e$, the system disturbance is suppressed and the system enters a steady state, thereby observing the unknown state quantity $x_2$.

In order to make the observer quickly approach stability and reduce the high-frequency chattering phenomenon of the estimator in the process of system state changes, the selected nonlinear feedback functions $g_1$ and $g_2$ are as follows:

$$\begin{align*}
g_1 &= e \\
g_2 &= |e|^{\frac{1}{2}} \cdot \zeta(e)
\end{align*} \quad (12)$$
\[ \xi(e) = \begin{cases} \text{sign}(e), & |e| > \delta \\ \frac{e}{\delta}, & |e| \leq \delta \end{cases} \]  

(13)

where sign is a sign function, and \( \delta \) is a very small number near zero.

This observer can observe the ground braking force \( F_x \) and wheel speed \( \omega \) received by the wheel end in real time during the deceleration process. During the deceleration process, the longitudinal load can be considered as a slowly changing variable. The actual wheel end braking force \( F_x \) has the same changing law as the road adhesion coefficient \( \mu \).

Therefore, according to the previous analysis, as the slip rate changes, \( F_x \) also has a stable region that increases from small to large and an unstable region that gradually decreases from the peak value. According to this rule, this article designs a simple slope method to estimate the optimal wheel slip rate in the braking process of the car.

The specific control logic is shown in the figure below. The vehicle model outputs the braking torque \( T_b \) and wheel speed value \( \omega \) to the braking force observer. When the observer outputs the current wheel end braking force observation value \( F^*_x \), the current vehicle speed information is used calculate the actual slip rate, and discretize the observed value \( F^*_x \) and the slip rate \( \dot{\lambda} \). Use the \( k \) value at the current time and the \( k-1 \) value at the previous time to calculate \( \Delta F_x \) and \( \Delta \lambda \) respectively. As the wheel is controlled when the braking force \( F_x \) reaches the peak and the rate of change of slip rate is greater than zero, the current slip rate \( \lambda(k) \) is the slip rate corresponding to the peak adhesion coefficient.

**III. SLIP RATE CONTROLLER DESIGN**

The function of the slip rate controller is to control the actual slip rate near the target slip rate by controlling the output braking torque \( T_b \) at the wheel end, which is similar to the ABS anti-lock braking system in traditional cars. In the current research, the widely used slip rate control method is sliding mode control, which has the advantages of fast response, simple structure and strong robustness [28], [29].

First, define the closed-loop input as the slip rate difference:

\[ e = \lambda_{bat} - \lambda \]  

(14)

In the formula, \( \lambda_{bat} \) is the target slip rate value, and \( \lambda \) is the actual slip rate value, which can be obtained by formula (5).

Flutter phenomenon is one of the bad effects of sliding mode control. In order to eliminate the chattering phenomenon in sliding mode control, this article improves the sliding mode surface and the reaching law of the sliding mode controller to make the control process more stable. Among them, the sliding surface of the controller is defined as follows:

\[ s = e + a \int e \text{d}t \]  

(15)

The reaching law of traditional sliding mode control can be expressed by the following formula:

\[ \dot{s} = -e \cdot \text{sat}(i) - k \cdot s \]  

(16)

In the design of the reaching law, the coefficient \( \varepsilon \) has a great effect, and the decrease of \( \varepsilon \) value can reduce the system chattering phenomenon, but it will lead to the decrease of the speed of the system approaching stability. Therefore, this article improves the traditional sliding mode controller, using \( k_1 \times s(k) \) instead of the original coefficients, and using the continuous function \( G(s) \) instead of the original saturation function. The improved adaptive sliding mode controller’s reaching law is as follows:

\[ \dot{s}(k) = -k_1 |s(k)| gG(s) - k_2 g s \]  

(17)

\[ G(s) = \frac{1}{|s| + d} \]  

(18)

Derivation of the sliding mode surface of the controller, combined with the wheel braking dynamics equation described above, can obtain the following relationship:

\[ \begin{cases} \dot{i} = \dot{e} + ae \\ \dot{e} = -\dot{\lambda} \\ \dot{\lambda} = \frac{(1 - \lambda)\dot{V} - \dot{\omega} R}{V} \\ \dot{\omega} = \frac{F_x R - T_b}{R} \end{cases} \]  

(19)

Combining the above formulas, the output of the improved adaptive sliding mode controller is:

\[ T_b = \frac{F_x R^2 - ((1 - \lambda_{bat} + e)\dot{V} + V\dot{e})J}{R} \]  

(20)

where \( F_x \) is the braking force applied to the wheels by the ground, which is given by the braking force observer in the third section, \( R \) is the wheel radius, \( \dot{V} \) is the vehicle acceleration, which can be measured by the vehicle acceleration sensor. The optimal slip rate estimation module designed in the previous section obtains the target slip rate \( \lambda_{bat} \). The control block diagram is as follows:
The adaptive sliding mode controller designed above is simulated and verified with a target slip rate of 0.2, as shown in Fig.6. When the slip rate control is turned on between 1 second and 3.8 seconds during the vehicle deceleration process, the sliding mode controller can quickly approach the target slip rate, and the control is stable without chattering. After 3.5 seconds, due to deceleration, the vehicle speed and wheel speed are relatively low, so the control effect of the sliding mode controller is slightly reduced, but at this moment the braking process of the car is about to end, and the slip rate can affect the safety and stability of the vehicle.

After 3.5 seconds, due to deceleration, the vehicle speed and wheel speed are relatively low, so the control effect of the sliding mode controller is slightly reduced, but at this moment the braking process of the car is about to end, and the slip rate can affect the safety and stability of the vehicle.

**IV. COMPOUND BRAKE CONTROL SYSTEM DESIGN**

From the perspective of safety and recovery of braking energy, in the braking process of a distributed pure vehicle, when the regenerative braking torque of the motor theoretically provides the required braking torque, the response speed is the fastest and the energy recovery effect is the best. However, in fact, the regenerative braking torque that the motor can provide is limited. For distributed-driven in-wheel motors or hub motors, it is mainly limited by the external characteristic curve of the motor and the reduction ratio of the reducer.

From this figure, it can be seen that the maximum torque that a single motor can provide is 150N·m, and as the speed increases. In order to maintain a high speed after 2000 rpm, the torque that the motor can provide is gradually reduced to 50N·m.

Generally, the reduction ratio of the reducer ranges from 3 to 6, so the maximum torque that a single motor can provide to the wheel end is about 1000N·m, which cannot actually meet the deceleration requirements of electric vehicles. Therefore, it is necessary to design a composite brake control system that combines motor braking and hydraulic braking to provide braking torque for electric vehicles.

**V. UNITED SIMULATION EXPERIMENT VERIFICATION**

**A. SIMULATION MODEL AND PARAMETER SETTINGS**

Since it is necessary to use the vehicle model, road model and driver model to provide the input and output variables.
corresponding to the control strategy of the compound brake anti-lock system designed in this article, the vehicle dynamics simulation software CARSIM and MATLAB/SIMULINK co-simulation method are selected. The previous control method is simulated and verified. The key parameter settings of the vehicle for simulation are shown in the table below.

**TABLE 2. Key parameter setting of the whole vehicle.**

| Variable                          | Symbol   | Value   |
|-----------------------------------|----------|---------|
| Vehicle mass                      | M        | 1650kg  |
| Maximum torque of single motor    | T<sub>max</sub> | 300N·m  |
| Maximum speed of single motor     | N<sub>max</sub> | 8000rpm |
| Distance from center of mass to front axle | L<sub>f</sub> | 1400mm  |
| Distance from center of mass to rear axle | L<sub>r</sub> | 1650mm  |
| Wheel radius                      | R        | 0.32m   |
| Moment of inertia of the wheel    | J        | 0.5kg·m<sup>2</sup> |
| Reducer reduction ratio           | g        | 5       |

Fig.9 shows the block diagram of the overall control strategy designed by the joint simulation in this article. The control strategy is built by SIMULINK to build a simulation calculation model, and input the control variables into the CARSIM vehicle model. The CARSIM model includes vehicle braking, transmission system models, and tires. Models, road models and driver models provide the required calculation parameters for the control strategy.

It can be seen from Fig.10 that the road surface identification results quickly stabilize after large fluctuations, and can quickly estimate the ideal slip rate of each wheel under the current road surface. The target slip rate of the front and rear wheels is affected by the vertical load. There is a slight difference, but they remain around 1.7. Under this working condition, the target braking strength is higher than the peak adhesion coefficient of the road, and the adhesion provided by the road does not meet the braking demand. Therefore, it can be seen from the Fig.10 (d) that the motor outputs the maximum regenerative braking torque that can be provided. The remaining part is provided by hydraulic braking torque.

It can be seen from Fig.10 (b) and (c) that the adaptive sliding mode controller controls the required braking torque and keeps the target slip rate at around 1.7. From the overall deceleration, process in Fig.10 (a), it can be seen that the vehicle speed and the four-wheel speed always maintain a constant slip rate during the deceleration process, and under this high adhesion coefficient road condition, it only takes 2.7 seconds to complete the deceleration process. At 3.5 seconds, because the vehicle speed is too low, the motor brake control system is set to exit the braking process, and the hydraulic braking force directly decelerates the car, so the slip rate quickly rises to one. After the deceleration is completed, the vehicle speed and wheel speed are both zero, and the slip rate is reduced to zero at this moment.

Braking at a moderate braking intensity on a road with an adhesion coefficient of 0.5, the initial vehicle speed is 80km/h, and the braking process is shown in Fig.11. It can be seen from Fig.10 (c) that the road surface identification results tend to be stable after large fluctuations and the optimal slip rate can be quickly estimated to be 0.15, and the estimation accuracy is high. In this road condition, the regenerative braking torque can meet the required braking requirements, so the hydraulic braking force does not participate in the braking process. It can be seen from the Fig.10 (a) that the deceleration process is 4.2s.

Braking with moderate braking intensity on the road with adhesion coefficient of 0.2, the initial vehicle speed is 80km/h, and the car starts to brake after driving at a constant speed for one second. The braking process is shown in Fig.12. It can be seen from the Fig.11 (a) that the deceleration process takes about 12 seconds. This is because when driving on an icy and snowy road with an adhesion coefficient of 0.2 compared to a road with a higher adhesion coefficient, the tire cannot obtain a sufficient utilization coefficient from the road. Therefore, the car can only obtain a relatively small braking force. It can be seen from Fig11 (b) that the sliding mode controller can still output a stable demand torque value.

B. SIMULATION ANALYSIS OF A SINGLE ROAD

Perform moderate-intensity braking on a road with a peak adhesion coefficient of 0.85. The initial speed is 80Km/h. The driver model is set to keep driving at a constant speed in the previous second. At one second, the driver depresses the brake pedal to transfer the brake. Dynamic information is given to the compound brake control system. The simulation result is shown in Fig.10.
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FIGURE 10. Anti-lock braking under a single road adhesion coefficient of 0.85. (a) Diagram of vehicle speed and four-wheel speed during car deceleration. (b) Four-wheel demand total torque diagram during deceleration. (c) Four-wheel target slip rate control diagram during vehicle deceleration. (d) Distribution diagram of left front wheel regenerative braking force and hydraulic braking force.

FIGURE 11. Anti-lock braking under a single road adhesion coefficient of 0.5. (a) Diagram of vehicle speed and four-wheel speed during car deceleration. (b) Four-wheel demand total torque diagram during deceleration. (c) Four-wheel target slip rate control diagram during vehicle deceleration. (d) Distribution diagram of left front wheel regenerative braking force and hydraulic braking force.

...a long time to decelerate. Fig.11 (d) shows that, in order to maintain the target slip rate on a road with a higher adhesion coefficient, electric power and hydraulic braking power are required to participate in the braking process. Under medium...
and low adhesion coefficient roads, using the control method described above, only electric power is needed to complete the braking process. In this way, the regenerative braking of the motor not only realizes the speed and accuracy of control, but also realizes the purpose of maximizing the energy recovery of regenerative braking.

2) SIMULATION ANALYSIS OF BISECTIONAL ROAD

Fig. 14 shows the simulation analysis of the compound brake anti-lock brake control system under the bisectional road conditions. It can be seen from the Fig. 14 (c) that for the control of the slip rate, the right wheel is controlled at 1.4 to 1.5, the front and rear wheels on the left are controlled near 1.8. This is because the adhesion coefficient on the left side of the road is higher, while the right side of the road is a low adhesion coefficient road. By controlling the difference in slip rate, when the car is decelerating, the left and right wheels are controlled near the maximum friction force, so the car can still be safely slowed down without sideslip.

As shown in Fig. 14 (a), driving on roads with different left and right adhesion coefficients, comparing the road with adhesion coefficient of 0.8, the braking time of the car is slightly longer, and the braking process is 3.6 seconds. It can be seen from Fig. 14 (d) that under this working condition, the braking process can be completed by using regenerative braking force without the participation of hydraulic braking force.

3) SIMULATION ANALYSIS OF JOINT ROAD

At the initial speed of 80km/h, the simulation analysis of the joint road is carried out, and the simulation results are
shown in Fig.14. In this working condition, first let the car decelerate on a road with a peak adhesion coefficient of 0.8. During the deceleration process, the car drives onto a road with a low adhesion coefficient of 0.3. It can be seen from
the Fig.14 (c) that for the estimation control of slip rate, the braking control system can still correctly identify the road condition and estimate the best slip rate even when the road conditions change.

On a road with a high adhesion coefficient, the optimal slip rate is about 0.2. When driving on a road with a lower adhesion coefficient, due to the relationship between the adhesion coefficient and the slip rate, the optimal slip rate becomes about 0.12. It can be seen from the Fig.14 (d) that on the road with high adhesion coefficient, the hydraulic braking force and the regenerative braking force of the motor complete the braking work. On the road with low adhesion coefficient, the hydraulic braking system exits work and only the motor regenerative braking system performs braking, to maximize the recovery of regenerative braking energy.

As shown in Fig.14 (a), on a road with a high adhesion coefficient, since the ground can exert a relatively large ground braking force on the wheels, the vehicle speed and wheel speed are rapidly reduced. When driving on a road with a low adhesion coefficient, it cannot provide much the braking force on the ground, the deceleration of car driving and wheel rotation are all reduced, the braking process slows down, and finally in 5.5 seconds, the braking is completed and the car finally stops.

VI. CONCLUSION
This article designs a new compound brake anti-lock control system that combines regenerative braking and hydraulic braking for four-wheel-drive distributed pure electric vehicles. Firstly, an estimation system based on a nonlinear state observer is used to observe the braking force and estimate the optimal slip rate. Secondly, aiming at the control problem of slip rate, an improved adaptive sliding mode controller output demand torque is designed to realize the control of the target slip rate, which enhances the stability and robustness of the system. The compound brake control system prioritizes the use of motor regenerative braking force, which makes braking control fast and accurate, and can recover regenerative braking energy to the greatest extent. Simulations of different road conditions are carried out in MATLAB and CARSIM simulation environment. The simulation results show that the control system designed in this article can ensure the stability of the vehicle under emergency braking.

In future work, we will start with the multi-motor loss model of electric vehicles and the torque control of permanent magnet synchronous motors, and further explore how to distribute the braking force in a multi-motor system to achieve the best overall efficiency and how to use advanced control algorithms to achieve stable control of motor torque.

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