A Fuzzy-Algorithm-Based Sliding Mode Control Approach for Acceleration Slip Regulation of Battery Electric Vehicle

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Abstract
Due to large torque and quick response of electric motor, the traction wheels of battery electric vehicle (BEV) are easy to slip during the initial phase of starting. In this paper, an acceleration slip regulation approach based on sliding mode control algorithm is proposed to prevent the wheel slip of BEV. The traction wheel slip ratio is used as the control parameter, and a sliding mode controller is deduced from it. A fuzzy algorithm is employed to revise the switch function of sliding mode controller. After stability and robustness analysis, the sliding mode controller is validated by dynamic simulation of BEV. The sliding mode control law is transferred into vehicle control unit and is verified by road tests, the results of which show that the sliding mode controller is a good candidate to prevent the wheel slip.

Keywords: Electric motor; wheel slip; slip ratio; fuzzy algorithm; robustness analysis

1 Introduction
With the public increasing attention to environmental and energy issues, reducing emissions from fuel vehicles and finding alternatives have become the first priority of current automotive industry. As a kind of zero-emission vehicles, battery electric vehicle is gradually attracting more and more attention and becomes a research focus [1]. The acceleration slip regulation (ASR) system has been widely used in internal combustion engine (ICE) vehicles. Compared with traditional ICE vehicles, permanent magnet synchronous motor (PMSM) has the working characteristics of quick-response and high-torque, which leads to be easier to cause wheel slip during acceleration process [2]. It can not only reduce the driving efficiency of vehicle, accelerate tire wear, increase powertrain load and energy consumption, but also damage the driving maneuverability, stability and comfort. Therefore, the employment of ASR system is benefit to improve driving safety and reduce the risk of accidents [3].

Anti-slip measures for ICE vehicles include: cutting down opening degree of throttles to reduce engine power, controlling the wheel-side braking torque, and adjusting hold-down force of clutch [4]. Compared with ICE, PMSM has several advantages, such as sensitive response and precise torque control [5]. Therefore, the wheel slip can be controlled by regulating the output torque of electric motor.

Considering the security during the acceleration process, relevant researchers take vehicle yaw moment into account. When the driving force control system enters the ASR controller, there will be a problem of generating undesired yaw moment and yaw angular velocity. For an electric vehicle with rear wheel drive alone, a dynamic torque distribution controller is designed to solve this problem [6]. On the basis of model predictive control (MPC), except the regulation of wheel slip, Yuan also optimizes vehicle safety, braking performance, driver comfort, and energy efficiency by adding additional cost functions [7].

A majority of research into anti-slip systems focus on two-wheel independent drive (2WID) or four-wheel independent drive (4WID) electric vehicles. In fact, due to the disadvantages of immature technology and high cost, wheel hub motor is rarely applied into vehicle architecture [8]. In addition, it is essential for 4WID electric vehicles to be equipped with additional speed sensors due to adopting the slip-ratio-based anti-slip system. Therefore, most of the relevant research is still in a simulation stage and does not carry out verification on road test. In this paper, the research object is a rear-drive logistics BEV, the speed of which can...
be obtained by front wheels so that it is easier to be tested and verified in practice.

In the past ten years, several algorithms have been applied into this field [9]. Owing to the advantages of PID control that it does not depend on the system model, as well as dealing with uncertain and complex problems, Guo takes it as the core control algorithm. At the same time, for the assumption of constant longitudinal force, a compensator is designed to offset the calculation error [10]. However, the individual PID control is not suitable for changing environment. Li applied fuzzy PID control to ASR system. The parameters of the controller are amended in real time to make it satisfy the adaptability of nonlinear and time-variant system [11]. Considering the advantages of good robustness, adaptability, and better fault tolerance [12], fuzzy control is another method to solve the problem. Yin put forward a kind of ASR control system, which is designed by controlling the difference of actual angular acceleration and the threshold of angular acceleration. Compensating an adjusting torque on the basis of the expected one, so that the slip ratio can be controlled in an ideal range [13]. However, fuzzy control will lead to low control accuracy and poor dynamic quality, but it is suitable for the use in conjunction with other control algorithms. A rolling optimization strategy instead of a global one-time optimization is used in MPC algorithm, which can compensate uncertainty timely and has better dynamic performance [14]. Sekour proposed a non-linear model prediction (NMP) direct torque control (DTC) algorithm [15]. The fuzzy controller is used to generate weight factors online, which can guarantee a high response speed under the flexible loads.

Sliding mode control (SMC) is a kind of non-linear control method and has many advantages like fast response, insensitive to parameter changes and disturbances, etc [16]. Therefore, SMC is applied into ASR system by many researchers [17–22]. However, it will lead to system oscillation without targeted measures [17, 18, 22]. In order to make the system have a good dynamic quality during the process, an approach with a positive control gain coefficient is adopted in [19]. However, due to the existence of the ground longitudinal force observer, the error of the observer may cause the system to be unstable. In order to reduce system oscillation, Zhou applied a special switching function, which makes the system have less speed when near the sliding surface [20]. Ricardo proposed a continuous SMC algorithm and the sign function is replaced by continuous approximation [21]. Experimental results show good slip regulation and robustness to disturbances.

The acceleration and slip processes is a short time, which requires the ASR system to have a small overshoot and quick response. In this paper, based on the fuzzy control algorithm, a sliding mode control law is proposed to prevent the drive wheels slip. The controller is proved to have Lyapunov stability and good robustness. Finally, the control law is put into the vehicle control unit (VCU) to do some vehicle road tests.

The rest of this paper is given as follows. Section 2 introduces vehicle architecture and control problem. In Section 3, the sliding mode controller and a factor correction algorithm based on fuzzy control algorithm is described in detail. The Lyapunov stability and robustness analysis of the control laws are in Section 4. A description of the control approach is presented in Section 5. In Section 6, the simulation results validate the effectiveness of the control law. A test platform of BEV is built and some experimental results are analyzed in Section 7. Finally, some conclusions are shown in Section 8.

2 Problem description
2.1 Vehicle architecture

An architecture of a 4.5t rear-drive logistics BEV in cooperation with a company is illustrated in Fig.1. A new type of motor direct drive axle is adopted in this vehicle. Compared to the conventional motor drive axle architecture, this drive train is more compact, which can reduce the vehicle’s unloaded mass by about 20% and increase the transmission efficiency by 10%. An open differential is used in the drive axle, which allows the two wheels to have different speed while outputting the same torque. The transmission ratio of the final drive is 13.52. The PMSM can output a peak torque in a short time and some parameters of the motor are shown in Tab.1. The accelerator pedal is an electronic pedal with a voltage sensor in a range of 0.5-3.2V to cover the entire pedal stroke. Anti-lock brake system (ABS) adopts wheel speed sensors to obtain angular speed in real-time. Controllers on board communicate with each other through controller area network (CAN).

| Parameter   | Value   | Parameter   | Value   |
|-------------|---------|-------------|---------|
| Peak power  | 150 (kW)| Rated power | 70 (kW) |
| Peak speed  | 8600 (rpm) | Rated speed | 3290 (rpm) |
| Peak torque | 550 (Nm) | Rated torque | 203 (Nm) |

The VCU obtains the maximum torque by estimating system capacity of the current state, which is mainly limited by the drive motor performance and the power. The torque $T_{mot}$ limited by the motor depends on the external characteristic curve. The torque
The relationship between the adhesion coefficient and the slip ratio can be expressed by the following formula [23, 24]:

$$\mu_{\lambda_i} = \frac{2\mu_{\max} \lambda_i \lambda_{\text{opt}}}{\lambda_i^2 + \lambda_{\text{opt}}^2}$$

(3)

where, $\mu_{\lambda_i}$ is the wheel adhesion coefficient and $\lambda_i$ is the wheel slip ratio.

2.2 Control problem

There is a problem encountered in the development process of automobile. Due to the low-speed and high-torque working characteristics of PMSM, it is easy to cause wheels slip on the low-adhesion condition, which can lead to vehicle instability. Based on the above-mentioned, the following control objectives should be considered:

1. it must be reliable to prevent wheel from slipping during traction process;
2. the vehicle can achieve good longitudinal acceleration performance on the premise of vehicle stability;
3. practical application of the controller should be guaranteed.

As shown in Fig.2, when the slip ratio is in the optimal control zone, the biggest adhesion coefficient can be obtained. However, with a slip ratio is over the optimal value $\lambda_{\text{opt}}$, it will be difficult to achieve precise control and easy to cause a risk of side-slip. Therefore, at the same time, limiting the wheels slip ratio within the optimal control zone can ensure high longitudinal adhesion and avoid possible out-of-control.

The selected formula (3) has high fitting accuracy both in the stable zone and the optimal control zone. When the drive wheels exceed $\lambda_{\text{opt}}$, the ASR controller needs to be activated immediately. Therefore, a control approach is designed to solve the problem: when the VCU enters and exits ASR system. The strategy will be introduced in section 5...
between the two drive wheels. Therefore, the "high-election principle" is adopted to take the larger slip ratio as the control object, before which, the larger wheel angular speed has to be obtained as follows:

$$\omega_i = \max(\omega_l, \omega_r)$$  \hspace{1cm} (5)

where, $\omega_l$ and $\omega_r$ is the angular speed of the left and right wheel, respectively.

In the driving process, the relationship between $v_o$ and $v$ satisfies $v_o \geq v$ all the time. Therefore, the larger slip ratio can be obtained as follows:

$$\lambda_i = \frac{v_o - v}{v_o} = \frac{r\omega_l - v}{r\omega_l}$$  \hspace{1cm} (6)

### 3.1.2 Wheel dynamics model

In this paper, the following points are assumed when the vehicle is simplified into a single wheel reference model:

1. the vehicle’s air resistance and wheel rolling resistance is ignored;
2. the left and right wheels have the same physical characteristics, such as the moment of inertia and rolling radius.

As shown in Fig.4, the vehicle dynamics model is simplified into a drive wheel model and the wheel revolution formulation can be obtained as follows:

$$J_i \dot{\omega}_i = T_i - rF_{xi}$$  \hspace{1cm} (7)

$$M \ddot{v} = F_{xi}$$  \hspace{1cm} (8)

where, $J_i$ is the moment of inertia, $kg \cdot m^2$; $T_i$ is the driving torque on the wheels, $Nm$; $r$ is the rolling radius of the wheels, $m$; $F_{xi}$ is the wheel longitudinal force, $N$; and $M$ is 1/2 the mass of the whole vehicle, $kg$.

The relationship between the ground longitudinal force $F_{xi}$ and the ground normal force $F_{zi}$ is as follows:

$$F_{xi} = \mu \lambda_i \cdot F_{zi}$$  \hspace{1cm} (9)

where, $\mu \lambda_i$ is the fitting adhesion coefficient.

#### 3.2 Control law

Because of the inherent discontinuous switching characteristics, in an actual system, there must be jitter near the sliding mode surface. Besides, it is impossible to eliminate the jitter and it can only be weakened. For the ASR system applied to BEV, the high-frequency vibration generated by chattering will accelerate the wear of the transmission system, increase the load on the motor, and make the system unstable. Generally speaking, the chattering is caused by the following four parts:

- The time lag of the switch;
- The space lag of the switch;
- The effect of system inertia;
- Discrete system itself.

In short, when the system reaches the switching surface, the moving point passes through it due to the system inertia. Finally, chattering is formed and superimposes on the ideal sliding mode [25]. Therefore, the chattering can be weakened effectively by reducing the reaching speed and system inertia.

The derivative of Eq (6) can be obtained as follows:

$$\dot{\lambda}_i = \frac{\dot{\omega}_i v - \omega_i \dot{v}}{r \omega_l^2}$$  \hspace{1cm} (10)

Substituting Eq (7), Eq (8), and Eq (9) into Eq (10):

$$\dot{\lambda}_i = \frac{vT_i}{J_i r \omega_l^2} - \mu \lambda_i \left(\frac{F_{ziv}}{J_i \omega_l^2} + \frac{F_{zi}}{rM \omega_l}\right)$$  \hspace{1cm} (11)
Substituting Eq (3) into Eq (11):

\[
\dot{\lambda}_i = \frac{vT_i}{J_1r_2\omega_i^2} - 2\mu_{\text{max}}\lambda_i\lambda_{\text{opt}} \left( \frac{F_{zi}v}{J_1\omega_i^2} + \frac{F_{zi}}{rM\omega_i} \right) \tag{12}
\]

The sliding mode surface is defined as:

\[
S = \lambda_i - \lambda_{\text{opt}} \tag{13}
\]

In order to meet the arrival conditions while considering approach speed and approach quality together, this paper uses an improved exponential approach law as follows:

\[
\dot{S} = -\varepsilon(S/\varphi)^2 \text{sgn}(S) - kS \tag{14}
\]

where, \( \varphi \) is the thickness of the boundary layer, and the \( \text{sgn} \) function is defined as follows:

\[
\text{sgn}(x) = \begin{cases} 
1, & x > 0 \\
0, & x = 0 \\
-1, & x < 0
\end{cases} \tag{15}
\]

Multiply formula (13) by formula (14) to get the following equation:

\[
S \cdot \dot{S} = -\varepsilon S^3 \text{sgn}(S) - kS^2 < 0 \tag{16}
\]

The approach law satisfies the existence conditions of the generalized sliding mode. Therefore, it satisfies the reachability condition of the sliding mode.

\[
\dot{S} = \dot{\lambda}_i - \dot{\lambda}_{\text{opt}} = -\varepsilon(S/\varphi)^2 \text{sgn}(S) - kS \tag{17}
\]

Substituting Eq (12) into Eq (17) to obtain the control law:

\[
T_i = \frac{2\mu_{\text{max}}\lambda_{\text{opt}}\lambda_i}{\lambda_{\text{opt}}^2 + \lambda_i^2} \cdot rF_{zi} + \frac{2\mu_{\text{max}}\lambda_{\text{opt}}\lambda_i}{\lambda_{\text{opt}}^2 + \lambda_i^2} \cdot \frac{J_1F_{zi}\omega_i}{Mv} - \frac{rJ_1\omega_i^2}{v} \left( \varepsilon(S/\varphi)^2 \text{sgn}(S) + kS \right) \tag{18}
\]

As can be seen from the formula above, the sliding mode control law consists of three parts: The first part \( \frac{2\mu_{\text{max}}\lambda_{\text{opt}}\lambda_i}{\lambda_{\text{opt}}^2 + \lambda_i^2} \cdot rF_{zi} \) is the main part of the acceleration torque; the second part \( \frac{2\mu_{\text{max}}\lambda_{\text{opt}}\lambda_i}{\lambda_{\text{opt}}^2 + \lambda_i^2} \cdot \frac{J_1F_{zi}\omega_i}{Mv} \) is the influence of the wheel inertia on the driving torque. The wheel moment of inertia is usually very small, which results in that this term has very little effect on the driving torque; the third part

\[
\frac{rJ_1\omega_i^2}{v} \left( \varepsilon(S/\varphi)^2 \text{sgn}(S) + kS \right)
\]

is the control adjustment item. When the wheel slip ratio exceeds the threshold, this item will provide a negative torque to correct the total driving force, and vice-versa.

### 3.3 Fuzzy controller

In this section, through a quantitative analysis of the relationship between the approach law coefficient (ALC) and chattering, an adaptive correction algorithm based on a fuzzy controller is proposed. When the error of the wheel angular acceleration and slip ratio is large, a larger ALC will be selected to achieve a faster approach speed and a smaller ALC will be selected when the error is small. The dynamical adjusting of the ALC can make the motion trajectory more suitable for the sliding mode surface, which can ensure the dynamic quality of the arrival process and reduce the high-frequency chattering of the system at the same time.

Substituting Eq (6) into Eq (10) and sort it out:

\[
\dot{\lambda}_i = \frac{(1 - \lambda_i) r\omega_i - \dot{v}}{r\omega_i} \tag{19}
\]

When the wheel slip ratio is stable at a certain value, \( \dot{\lambda} \) is 0, and the above formula becomes:

\[
\dot{\lambda}_i = \frac{\dot{v}}{(1 - \lambda_i) r} \tag{20}
\]

![Figure 5 Membership function of variables](image-url)
It can be seen from the equation above that when the slip ratio is stable, the wheel angular acceleration is correlated positively with the vehicle acceleration. Therefore, when the vehicle has the maximum acceleration, the optimal reference value of wheel angular acceleration can be obtained.

From Eq (8) and Eq (9), the maximum acceleration of a certain road is:

\[
\dot{v}_{\text{max}} = \frac{F_{r\text{max}}}{M} = \frac{\mu \lambda_{\text{opt}} F_{zi}}{M} \tag{21}
\]

Therefore, the reference optimal wheel angular acceleration is:

\[
\alpha_{\text{opt}} = \dot{\omega}_{\text{opt}} = \frac{\mu \lambda_{\text{opt}} F_{zi}}{(1 - \lambda_{\text{opt}})Mr} \tag{22}
\]

The first input variable is \(\Delta E_1 = \lambda_i - \lambda_{\text{opt}}\). Based on theoretical derivation and practical experience, in order to provide sufficient coverage, \(\Delta E_1\) is divided into five fuzzy subsets: [NS, ZO, PS, PM, PB]. The membership function is shown in Fig. 5 (a). 'N' and 'P' stand for 'Negative' and 'Positive', respectively. 'B', 'M', and 'S' stand for 'Big', 'Middle', and 'Small', respectively.

The second input variable is \(\Delta E_2 = \dot{\omega}_i - \dot{\omega}_{\text{opt}}\) and it is divided into seven fuzzy subsets: [NS, NM, NB, ZO, PS, PM, PB]. The membership function is shown in Fig. 6 (b).

In the fuzzy controller, the Mamdani method is used to perform the fuzzy logic calculation. The fuzzy rules are shown in Tab. 2. The defuzzification method is the gravity center. Based on multiple simulation results, the factors are divided into five fuzzy subsets: [A, B, C, D, E], and the membership function is shown in Fig. 7 (c) and (d).

4 Controller analysis

4.1 Stability analysis

In order to verify the stability of the proposed closed-loop controller, the Lyapunov function is defined as follows:

\[
V = (\lambda_i - \lambda_{\text{opt}})^2 \tag{23}
\]

It can be seen from Eq (23) that the control system energy function is positive definite. To ensure the asymptotic stability of the system, the derivative of \(V\) should be negative definite, which can be calculated as follows:

\[
\dot{V} = 2(\lambda_i - \lambda_{\text{opt}}) \tag{24}
\]

Substituting Eq (14) into Eq (24) and the following formula can be obtained:

\[
\dot{V} = -2(\lambda_i - \lambda_{\text{opt}}) \left\{ \varepsilon (\lambda_i - \lambda_{\text{opt}})/\varphi \right\}^2 \text{sgn}(\lambda_i - \lambda_{\text{opt}}) + k(\lambda_i - \lambda_{\text{opt}}) \right\}
\]

In Eq (25), the ALCs are set to be positive definite. Therefore, it can be inferred:

\[
\begin{cases}
\dot{V} \leq 0, & \lambda_i \leq \lambda_{\text{opt}} \\
\dot{V} \leq 0, & \lambda_i > \lambda_{\text{opt}}
\end{cases}
\tag{26}
\]

As can be seen from Eq (26) that \(\dot{V}\) is negative definite, so the proposed sliding mode controller is stable asymptotically.

4.2 Robustness analysis

In fact, BEV is a non-linear and uncertain system. Therefore, it is necessary to perform a robust analysis. In this section, for the errors of the motor torque and the road adhesion coefficient, the robustness of the sliding mode controller is analyzed.

4.2.1 Motor torque error

There is an error between the actual output torque and the required value. Assuming that \(\Delta T_i\) is the torque error, so the actual torque is:

\[
T_{ia} = T_i + \Delta T_i \tag{27}
\]

where, \(T_{ia}\) is the real output torque. Substituting Eq (11) into Eq (24) and the following formula can be obtained:

\[
\dot{V} = 2(\lambda_i - \lambda_{\text{opt}}) \left\{ \frac{T_i v}{J_i r \omega_i^2} - \mu \lambda_i \left( \frac{F_{zi} v}{J_i r \omega_i^2} + \frac{F_{zi}}{r M \omega_i} \right) \right\}
\]

Replacing \(T_i\) by Eq (27), and the following formula can be obtained:

\[
\dot{V} = 2(\lambda_i - \lambda_{\text{opt}}) \left\{ \frac{\Delta T_i v}{J_i r \omega_i^2} + \frac{T_i v}{J_i r \omega_i^2} - \mu \lambda_i \left( \frac{F_{zi} v}{J_i \omega_i^2} + \frac{F_{zi}}{r M \omega_i} \right) \right\}
\]

\[
\dot{V} = 2(\lambda_i - \lambda_{\text{opt}}) \frac{\Delta T_i v}{J_i r \omega_i^2} \tag{29}
\]
Substituting Eq (11) and Eq (17) into Eq (29) to get the following formula:

$$\dot{V} = 2(\lambda_i - \lambda_{opt}) \left\{ \frac{\Delta T_i v}{J_i r \omega_i^2} + \left\{ \varepsilon[(\lambda_i - \lambda_{opt})/\varphi]^2 \text{sgn}(\lambda_i - \lambda_{opt}) + k(\lambda_i - \lambda_{opt}) \right\} \right\}$$

(30)

It can be concluded that $\Delta T_i$ should satisfy the following equation:

$$|\Delta T_i| \leq \left| \frac{J_i r \omega_i^2}{v} \left[ -\varepsilon(S/\varphi)^2 \text{sgn}(S) - kS \right] \right|$$

(31)

Lyapunov function $V$ is less than 0 and it means that when the adhesion coefficient error satisfies Eq (35), the ASR controller is stable and robust, and vice versa.

4.2.2 Adhesion coefficient error

The road surface adhesion coefficient is usually obtained by some estimation methods. It will lead to an error between the actual adhesion coefficient and the estimating one. $\Delta \mu_i$ is assumed as the adhesion coefficient error. Therefore, the actual adhesion coefficient $\mu_{ia}$ can be defined as follows:

$$\mu_{ia} = \Delta \mu_i + \mu_i$$

(32)

Substituting Eq (32) for $\mu_i$ in Eq (28), and the following equation can be obtained:

$$\dot{V} = 2(\lambda_i - \lambda_{opt}) \left[ \frac{T_i v}{J_i r \omega_i^2} - (\Delta \mu_i + \mu_i) \left( \frac{F_z i v}{J_i r \omega_i^2} + \frac{F_z i}{r \omega_i M \omega_i} \right) \right]$$

(33)

Substituting Eq (11) and Eq (17) into Eq (33) to get the following equation:

$$\dot{V} = 2(\lambda_i - \lambda_{opt}) \left\{ \Delta \mu_{i\lambda} \left( \frac{F_z i v}{J_i r \omega_i^2} + \frac{F_z i}{r M \omega_i} \right) + \left\{ \varepsilon[(\lambda_i - \lambda_{opt})/\varphi]^2 \text{sgn}(\lambda_i - \lambda_{opt}) + k(\lambda_i - \lambda_{opt}) \right\} \right\}$$

(34)

Therefore, it can be inferred that $\Delta \mu_i$ should satisfy the following equation:

$$|\Delta \mu_{i\lambda}| \leq \left| \frac{J_i r \omega_i^2 M}{F_z (r M v + J_i \omega)} \left[ -\varepsilon(S/\varphi)^2 \text{sgn}(S) - kS \right] \right|$$

Lyapunov function $V$ is less than 0 and it means that when the adhesion coefficient error satisfies Eq (35), the ASR controller is stable and robust, and vice versa.

5 Control approach

In this section, a brief description of the control approach is presented. As illustrated in Fig.6, the VCU is in the DCS by default. Before entering the ASR system, two judgments need to be performed.

Case 1: due to the property of the slip ratio formula, the measurement errors of the wheel speed sensors are amplified when the vehicle drives in a low speed, which will cause a slip ratio mutation. Therefore, by setting a speed threshold $v_{th}$, the misjudging of the slip status can be eliminated.

Case 2: when $\lambda_{max}$ is bigger than $\lambda_{th}$, the wheel is considered in a slippage state. With a view to the capacity limit, the smaller value between the ASR controller torque $T_{ASR}$ and $T_{Cap}$ is chosen as the final target torque $T_{TAR}$.

When the vehicle is in the ASR control mode, another two conditions will be judged in a loop.

Case 3: the driver’s acceleration intention is estimated by comparing $C_{DR}$ with a threshold $\sigma$.

Case 4: the research in this paper focuses on the straight driving condition.

When the vehicle steers or the left and right adhesions are different, a slip difference between the two drive wheels will be generated. If Case 3 or Case 4 is not satisfied, the VCU will exit the ASR system immediately.

Before entering another mode, the VCU will always stay in the current state. During each judgment, anti-shake process is required and the time threshold is 10 cycles (100ms). The two control systems, DCS and ASR, can be switched in time according to different
working conditions, which makes the ASR system become a practical active safe driving assistance system.

6 Simulation validation

In this section, in order to verify the performance of the ASR system under different conditions, two simulation scenarios of low and high adhesion roads are performed. The parameters of the simulation model is set to be the same as the rear-drive BEV. A part of these parameters are shown in Tab.3. The simulation step size is fixed-step, fundamental sample time is 10 ms and the solver is discrete.

6.1 Low adhesion condition

Table 3 Vehicle model parameters.

| Description                  | Value        |
|------------------------------|--------------|
| Gross vehicle weight         | 2200 (kg)    |
| Radius of tire               | 364 (mm)     |
| Wheelbase                    | 3360 (mm)    |
| Moment of inertial of wheel  | 1 (kg/m²)    |
| Transmission ratio           | 13.25 (-)    |
| Unsprung mass of front axle  | 400 (kg)     |
| Unsprung mass of rear axle   | 500 (kg)     |

In this section, an ice road is selected as the simulation verification environment. $\mu_{\text{max}}$ is set to 0.1, $\lambda_{opt}$ is set to 0.1 and $\varphi$ is set to 0.01.
As can be seen from Fig.7 (a) and (c), at the moment when vehicle starts, the torque increases rapidly. At the same time, there is a sudden change of the slip ratio from 0 to about nearly 0.9. Due to the definition formula of slip ratio, it is close to 1 when the vehicle speed approximates 0, which is consistent to the simulation result. About 0.25s later, the torque enters into an adjustment process and stabilizes gradually. It can be seen from Fig.7 (b) that $F_x/F_z$ is close to $\mu_{max}$ (0.1), which means that the drive wheels can make the best use of the ground adhesion. As illustrated in Fig.7 (d), the vehicle speed and wheel speed increase steadily. Based on the analysis above, it can be concluded that under a low adhesion condition, the ASR controller can prevent wheels from slipping effectively. Besides, the acceleration process has good straight-line driving stability. The slip ratio can be controlled in the optimal control zone within 0.3s and the quick response is proved.

6.2 High adhesion condition

In this section, a dry cement pavement is selected as the simulation verification environment. $\mu_{max}$ is set to 0.8, $\lambda_{opt}$ is set to 0.17, and $\varphi$ is set to 0.01. As can be seen from Fig.8 (a) and (c), at the moment when vehicle starts, the torque increases rapidly. Meanwhile, there is a sudden change of the slip ratio from 0 to about 1 and it drops to about 0.15 quickly. After 0.2s, the torque stabilizes gradually and enters an adjustment process with small jitter. At the same time, the slip ratio stabilizes near the optimal value of 0.17. It can be seen from Fig.8 (b) that $F_x/F_z$ is close to $\mu_{max}$ (0.17), which indicates that the wheels can make the best use of the ground adhesion. As shown in Fig.8 (d), the vehicle speed and the wheel speed increase smoothly during the entire acceleration process without obvious fluctuation. Finally, the vehicle speed reaches 8.9km/h.

Based on the analysis above, it can be concluded that under a condition of high adhesion road, the ASR controller can reduce the wheel slip effectively and obtain high adhesion. Besides, the acceleration process has good straight-line driving stability. The slip ratio can be controlled in the optimal control zone within 0.18s and the quick response is proved.

7 Vehicle road test

Due to the power limitation of the motor, it is difficult to slip on dry roads continuously. Some verification tests with a kind of slippery road condition are performed in this section. Low ABS sensor accuracy is difficult to obtain an accurate slip rate during low vehicle speed. Therefore, the BEV is accelerated to a certain speed before testing first. For the same reason, the slip rate may approach 1 at low speed Fig.10 (b). The ASR controller model is embedded into the VCU as a subsystem of the main functions. The VCU is shown in Fig.9 (a). The ABS is equipped with wheel speed sensors and the signal frequency of sampling is 100Hz. The ABS is shown in Fig.9 (b). Computer sampling results have a high frequency of 100Hz.

Some parameters used for vehicle road test are shown in Tab.3. The road condition is a slippery pavement with a maximum adhesion coefficient of 0.4 and an optimal slip ratio of 0.15. The thickness of boundary layer is set to 0.01. Due to the limited length of the road on the test site, the maximum speed should not exceed 60km/h to reserve a sufficient safety distance. Owing to the lack of a ground normal force sensor, compared to the simulation results, the item $F_x/F_z$ is missing.

It can be seen from Fig.10 (a) and (b) that, in the absence of ASR system, the output torque of the DCS increases to the maximum value quickly and then decreases to the rated torque with the increasing speed. The slip ratio increases to about 0.7 and remains at a high level for a while. However, with the ASR system, the VCU is in the DCS by default. When the driver presses the accelerator pedal, the slip ratio and the vehicle speed exceed their thresholds respectively and the VCU enters the ASR controller.
With the torque adjustment, the slip ratio is changing up and down along the optimal slip ratio all the time. When the vehicle speed is close to 60 km/h, the driver releases the accelerator pedal gradually and the VCU enters the DCS. As indicated in Fig.10 (c), when with the ASR system, the wheel speed is higher than vehicle speed slightly throughout the acceleration process. After 6.5s, the vehicle speed reaches 52km/h. The acceleration curves are shown in Fig.10 (d), from which it can be seen that, compared to the DCS, has better continuous acceleration performance. In addition, the vehicle does not show a tendency to side-slip and the vehicle posture is stable.

8 Conclusion

In order to solve the problem of wheel slip of battery electric vehicle, a sliding mode control algorithm is proposed. Considering the phenomenon that SMC is prone to system oscillation, a correction algorithm on the basis of fuzzy controller is established to adjust its key parameters. The effectiveness of the proposed control law has been verified by co-simulation. The results of vehicle tests show that the proposed controller has a small overshoot and a quick response. And the control approach is feasible during the vehicle road tests.

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Availability of data and materials

All data generated or analysed during this study are included in this published article [and its supplementary information files].

Authors’ contributions

The author’s contributions are as follows: Lin He, Qin Shi and Bingzhao Gao were in charge of the whole trial and provide ideas and inspiration; Cheng Yao wrote the manuscript; Zejia He assisted with sampling and analyses.

Competing interests

The authors declare no competing financial interests.

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