Two-Time-Scale Braking Controller Design with Sliding Mode for Electric Vehicles over CAN

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ABSTRACT This paper proposed a braking torque controller via two-time-scale design with sliding mode for electric vehicles with four in-wheel motors. According to the different rates of change between the vehicle and wheel motion during the braking process, the design of braking controller is implemented in two steps. Without considering the tire-road friction, a nominal braking controller is first developed over the slow-time scale. Then, a tire-road friction observer is adopted in the fast-time scale to recover the performance of the nominal braking controller. Owning to the high nonlinearity and complexity of the braking system, a sliding mode surface is further added in the nominal braking controller to ensure the stability and robustness of proposed braking controller. A braking supervisor is adopted to enable the proposed braking controller, which is based on the wheel slip as well as vehicle speed condition. And a torque allocation scheme is presented for the coordination between the regenerative braking system and the friction braking system in each wheel. Co-simulation is conducted using MATLAB/Simulink and CarSim. The effectiveness of proposed controller under different braking conditions is fully validated. A delicate controller area network (CAN) bus model is developed via SimEvent, by which the robust performance of proposed braking controller against CAN-induced time-varying delays is also investigated.

INDEX TERMS Braking control; two-time-scale design; sliding mode; CAN-induced delays; electric vehicle.

I. INTRODUCTION Braking system is of great importance to the ground vehicles [1], which directly determines the active safety performance. From traditional internal combustion engines vehicles [2] to elective vehicles including battery electric vehicle (BEV) [3][4] and hybrid electric vehicle (HEV) [5][6], and to the latest autonomous vehicles [7], braking control is always a concerned research hot spot. Johansen et al. [2] presents a gain-scheduled braking controller design with experimental validation, in which a Mercedes E220 is adopted as the test vehicle. Mi et al. [8] proposes an iterative learning braking control for electric vehicles, which has good adaptability to various road conditions. And Guo et al. [9] presents a coordinated braking and steering control for autonomous vehicles. Generally, there are two difficulties in the braking control of ground vehicles. The first one is the accurate acquisition of vehicle speed, which can directly affect the performance of braking controller. Although accurate vehicle speed information can be obtained by using high end navigation system [10], considering the cost, it is preferred that real-time vehicle speed can be estimated for market vehicles [11][12]. The second challenge is the strong nonlinearity of the tire-road friction, which shows complicated relations with wheel slip ratio, road condition, tire sideslip angle and temperature [2]. Generally, there are two methods to calculate the tire-road friction in the braking control. One is using the empirical tire-road models, in which magic formula [13], LuGre friction dynamics [14] and Burckhardt’s tire model [15] are widely adopted. As accurate parameters of these empirical models are generally difficult to obtain, tire-road friction or friction coefficient estimation is utilized as the second approach. It not only can help get more accurate tire-road friction information but also enhance the adaptivity of the braking controller [16]. Hoang et al. [17] designed a friction observer based on the Burckhardt’s tire model. Chen et al. [18] investigated the ABS with an adaptive observer for...
the internal state of the LuGre model. Besides, In [19] and [20], the Burckhardt’s model was approximated with linear parameters i.e. linear parameterization (LP). Hoseinnezhad et al. [21] directly maximize the longitudinal component of tyre-road friction.

To design the braking system controller, different approaches have been proposed such as PI [22], LQR [2], MPC [22][23], and so on. Han et al [24] presented an adaptive individual brake torque estimation method. And the $H_\infty$ braking controller was adopted in [25]. In [26] and [27], the neural networks was applied to EVs. The approaches of fuzzy sliding mode were proposed in [28] and [29]. De Castro et al. [30] proposed a sliding mode control (SMC) approach based on field programmable gate array technology while Okyay et al. [1] proposed an SMC structure employing derivative switching function with integral sliding surface. The SMC is widely used in braking systems to overcome the uncertainty [31], and has high robustness to the disturbance [32], external noise [33] and even unmodeled dynamics [34]. However, it is generally tedious in controller design for the braking system. Considering the different changing rates between the vehicle and wheel motion during the braking process, time-separately design is an effective method for the braking controller [35]. A two-time-scale braking controller, where a sliding mode surface is further designed into the nominal time scale to ensure the stability and robustness, is proposed in this paper.

The braking system of the EVs is composed of the friction braking system and the regenerative braking system [36]. Comparing with the traditional internal combustion engines vehicles, the allocation between these two braking systems is required to be further addressed. Wang et al. [37] proposed a braking allocation scheme for hydraulic and regenerative braking systems, where the hydraulic braking system generates a relatively stable pressure while the regenerative braking system make suitable and rapid response to the remained braking torque requirement. According to different frequency responses of the braking actuators, De Castro et al. [38] proposed a comprehensive braking allocation scheme, which can also handle actuator constraints and realize collaborative optimization between energy efficiency and dynamic performance. Heydari et al. [39] proposed a maximizing regenerative braking energy recovery of electric vehicles through dynamic low-speed cutoff point detection. Moreover, Shuai et al. [40] proposed a quadratic programming (QP) based torque allocation algorithm. In this paper, the torque was briefly allocated that the regenerative torque was maximized for the regeneration brake when the controller was triggered and the rest of the torque that demands was compensate by the friction braking system as it is not the emphasis of the paper.

The EV with four in-wheel motors is a research hotspot owing to its relatively simple powertrain connections [41] [42] and actuator redundancy [41]. The braking torque and mode of each wheel can be controlled independently, which increase the signal exchange requirement in control systems. With the development of in-vehicle network and x-by-wire technologies[43], the messages are delivered via a communication network, such as CAN, in vehicles [44]. Thus, the system of EVs with four in-wheel motors is network controlled instead of centralized. Due to the contradiction between the increasing requirement of message delivery and the limitation of the network bandwidth, the network induced time-varying delays are difficult to avoid. These network-induced delays would degrade or even deteriorate the performance of braking control [45][46]. Several studies have dealt with this issue on the stabilization of EVs with four in-wheel motors using the CAN control systems. Shuai et al. [40] considered the message scheduling. Caruntu et al. [45] present the impact of CAN-induced delays to the oscillations of a vehicle drivetrain. And the impact of CAN-induced delays to the lateral motion of vehicles was taken into consideration [46]. Thus, in order to further test the robust performance of the proposed controller against CAN-induced delays, a detailed model of a CAN bus was developed, and the CAN-induced delays were discussed in co-simulation via MATLAB/Simulink and CarSim, which is considered as the verification of the proposed controller in this paper. The contributions of this paper are two-fold:

1. A TTS braking controller with a sliding mode surface added to enhance the stability and robustness of the braking system was designed.

2. Detailed CAN bus model was established via SimEvent adopted to the simulation. And the robust performance of proposed braking controller against CAN-induced time-varying delays was investigated in co-simulation.

The rest of this paper is as follows: In Section II, problem formulation and dynamical modeling are presented. SMC with TTS torque controller design is presented in Section III. Co-simulation results via MATLAB/Simulink and CarSim are presented in Section IV and concluding remarks are summarized in Section V.

II. PROBLEM FORMULATION

A. ARCHITECTURE OF THE PROPOSED BRAKING CONTROL SYSTEM

![FIGURE 1. Scheme of the proposed braking torque control system](image)

The scheme of proposed braking torque control system is shown in Fig. 1. The braking torque is directly related to the
pedal travel at the beginning, and the braking control supervisor calculates the slip ratio, which is sent to high-level controller with the vehicle velocity to decide whether the braking controllers of each wheel would be triggered or switched off. The braking controllers output the braking torque to vehicle model. All the messages are transmitted through CAN bus, and the ‘background traffic’ is immitted into CAN bus to reach the upper limit of bandwidth which would generate the CAN-induced delays.

**B. VEHICLE LONGITUDINAL DYNAMICAL MODELING**

![FIGURE 2. Dynamics model of quarter vehicle](image)

As is shown in Fig. 2, dynamic model of the electric vehicle during the braking process can be derived as

\[
\begin{align*}
M \ddot{v} &= -F_z, \\
J \dot{\omega} &= rF_z - T_b, \\
\lambda &= \frac{v - \omega}{v}
\end{align*}
\]

where \(M\) is the quarter vehicle mass, \(v\) is the vehicle velocity, \(F_z\) is the braking force from road, \(J\) is the wheel rotational inertia, \(\omega\) is the wheel angular speed, \(r\) is the radius of wheel, \(T_b\) is the braking torque and \(\lambda\) is the tire slip ratio, defined by \(v\) and \(\omega\), which represents the slip condition of the wheel in a braking maneuver. \(\lambda = 0\) indicates the wheel is at a pure rotation state without slip while \(\lambda = 1\) indicates the wheel is completely locked.

It can be reached from (1) that

\[
\lambda = -\frac{(1 - \frac{r^2}{v^2}) F_z + r \omega T_b}{r (1 + (1 - \lambda) \frac{J}{Mr^2} r F_z - T_b)}
\]

where \(\dot{\lambda}\) is the first derivative of \(\lambda\) versus time, and consider \(\lambda\) as the control target. Considering the wheel inertia is usually much smaller than the equivalent vehicle inertia, i.e. \(J \ll Mr^2\) [47], dynamics (2) can be simplified as

\[
\dot{\lambda} = \frac{r}{vJ} (T_b - r F_z)
\]

**C. THE TYRE-ROAD FRICTION UNDER DIFFERENT CONDITIONS**

The tyre-road friction coefficient under different conditions is illustrated in Fig. 3. The \(\mu(\lambda)\) is in concerned with road surface [48], vehicle velocity [49], side slips [48] and temperatures [27]. The braking force is essentially the friction between the road and the tire, which is defined as

\[
F_z = F_z(\lambda, \mu, T, v, \alpha)
\]

where \(F_z\) is the wheel vertical load and \(\mu\) a nonlinear term, is the friction coefficient, which relates with \(\lambda\), the slip ratio, \(\mu,\) the road condition, \(T\), the tire temperature, \(v\), the vehicle velocity, and \(\alpha\), the wheel side slip angle. Thereinto, \(\mu\) is intimately influenced by \(\lambda\). Thus, \(\mu(\lambda, \mu, T, v, \alpha)\) is going to be written as \(\mu(\lambda)\) for conciseness, and (3) can be written as

\[
\dot{\lambda} = \frac{r}{vJ} (T_b - r F_z(\lambda))
\]

Notice that \(\mu = \mu(\lambda)\) is a nonlinear item, and the \(\mu - \lambda\) curve is complicate. Thus, it is a challenge to formulate \(\mu(\lambda)\) to achieve the precise value. A faster time-scale observer to circumvent the intricate \(\mu - \lambda\) curve and estimate the value of \(\mu(\lambda)\) is designed in SMC with TTS.

**III. SMC WITH TTS CONTROLLER DESIGN**

First of all, consider the nominal time-scale term in (4) before designing the faster part. Leaving out term \(-r F_z(\lambda)^{\ast}\), which is denoted as \(\psi(\lambda) = -r F_z(\lambda)\) and the slip dynamics degrades to the nominal part, which is

\[
\dot{\lambda} = \frac{r}{vJ} T_{\text{nom}}
\]

Define a sliding mode surface with integration as

\[
s = k \int e + e
\]

where \(e\) is the tracking error of \(\lambda\) to its reference value \(\lambda^\ast\), i.e. \(e = \lambda - \lambda^\ast\). It is easy to obtain that

\[
\dot{e} = \dot{\lambda} - \dot{\lambda^\ast} = \frac{r}{vJ} T_{\text{nom}} - \dot{\lambda^\ast}
\]

The first derivative of \(s\) versus time is

\[
\dot{s} = ke + \dot{e}
\]

Make the reaching law an exponential one, which is

\[
\dot{s} = -\alpha s + \beta s^2
\]

where \(\alpha > 0\) and \(\beta > 0\) are two parameters to be design. Choose Lyapunov candidate as \(V_i = \frac{1}{2} s^2\), which yields

\[
\dot{V_i} = s \dot{s} = -\alpha |s| - \beta s^2 < 0
\]
Consider $T_{\text{bnom}}$ as the controlled value, and it can be obtained that
\[
T_{\text{bnom}} = \frac{V_{\text{f}}}{r} \left( \lambda^* - ke - \beta s - \alpha \text{sgn} \ s \right)
\]  
(11)

Then, take the term $\psi(\lambda)$, the faster time-scale part, into consideration. To estimate it, an observer is designed as
\[
\dot{\lambda} = \frac{r}{v_{\text{f}}} T_b - \frac{1}{e} \left( \lambda - \hat{\lambda} \right)
\]  
(12)

and thereinto,
\[
\dot{\lambda} = \frac{r}{v_{\text{f}}} T_b - \frac{1}{e} \left( \lambda - \hat{\lambda} \right)
\]  
(13)

where $\hat{\lambda}(0) = \lambda(0)$, and $\varepsilon > 0$ is a small parameter to be determined.

The proposed controller should be
\[
\hat{T}_b = T_{\text{bnom}} - \psi(\lambda)
\]  
(14)

and the proof comes at following.

Denote $e^\sigma = \frac{1}{e} \left( \lambda - \hat{\lambda} \right)$, and it can be obtained from (12) and (13) that
\[
e^\sigma \dot{e}^\sigma = -\sigma - \frac{r}{v_{\text{f}}} \psi(\lambda)
\]  
(15)

which indicates that $e$ has a faster time response than that of $\lambda$ because of the small parameter $\varepsilon$. Manifold $\sigma$ as
\[
\sigma = -\frac{r}{v_{\text{f}}} \psi(\lambda)
\]  
(16)

Define the manifold error as
\[
\eta = \sigma + \frac{r}{v_{\text{f}}} \psi(\lambda)
\]  
(17)

Then, redefine $e_2 = \lambda - \lambda^*$ and mark it as $e_2$, and the error dynamics can be obtained as
\[
\dot{e}_2 = \frac{r}{v_{\text{f}}} \left( T_{\text{bnom}} - rF_{\mu}(\lambda) \right) - \dot{\lambda}^*
\]  
\[
= \frac{r}{v_{\text{f}}} \left( \dot{\lambda} - ke - \beta s - \alpha \text{sgn} \ s \right) + \frac{r}{v_{\text{f}}} l + \psi(\lambda) - \dot{\lambda}^*
\]  
(18)

Define a new sliding mode surface as
\[
s_2 = k \begin{bmatrix} e_2 + e_2 - \int \eta \end{bmatrix}
\]  
(19)

\[
\hat{s}_2 = ke_2 + \dot{e}_2 - \eta
\]  
(20)

Consider it as exponential reaching law
\[
\hat{s}_2 = -\alpha \text{sgn} s_2 - \beta s_2
\]  
(21)

Choose Lyapunov candidate as $V_2 = \frac{1}{2} s_2^2$, which yields
\[
\dot{V}_2 = s_2 \hat{s}_2 = -\alpha |s_2| - \beta s_2^2 < 0
\]  
(22)

It can be inferred via $\hat{s}_2$ that
\[
ke_2 + \dot{e}_2 - \eta = -\alpha \text{sgn} s_2 - \beta s_2
\]
\[
\dot{e}_2 = \eta - ke_2 - \alpha \text{sgn} s_2 - \beta s_2
\]  
(23)

Remark: Since $\dot{e}_2 = \dot{e} + e$, $e$, and $e_2$ can be considered as similarity as long as $\eta \rightarrow 0$, i.e. $e \rightarrow 0$ and $e_2 \rightarrow 0$.

**IV. CO-SIMULATION AND ANALYSIS**

**A. PROCESS OF THE SIMULATION**

The simulation model is composed as follows. First, the control signal of braking, like other signals in the vehicle, is transmitted through communication protocol and bus, CAN bus mainly, thus a detailed model of CAN bus is established in Simulink. Then, the control system will not be triggered at the beginning of a braking process while the braking torque is linearly related to the pedal travel, which imitates the driver’s behavior. Furthermore, the controller will be triggered when the slip ratio increases too much while it will be switched off when the vehicle velocity is small i.e. the control process is not necessary, and the maximal torque will be applied to lock the wheel until complete stop. Besides, a torque allocator is adopted to allocate the braking torque into the regenerative torque and the friction torque. The motors output the maximum of the electric torque for the regeneration brake, and the friction torque compensate the rest that demands. The friction torque would fill the whole demanded braking torque when the braking controller were not triggered. The co-simulation model is given in Fig. 4.
The detailed model of CAN bus is built in Simulink given in Fig. 5. There are four nodes with different priorities which decrease progressively from node 1 to node 4. Node 1 is for the proposed braking controller, node 2 is for the vehicle model which is co-simulated with CarSim, node 3 is for the braking control supervisor and the node 4 is for the background traffic generating the CAN-induced delays. The CAN Node module is shown in Fig. 6. A Gaussian noise generator is added to present the inevitable measurement errors caused by the sensors. The main parameters of the vehicle model and simulation are listed in Tab. I, where the vehicle mass, wheel inertia and radius are cited from [35].

### TABLE I

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass $(M)$</td>
<td>1230</td>
<td>kg</td>
</tr>
<tr>
<td>Wheel inertia $(J)$</td>
<td>0.6</td>
<td>kg.m^2</td>
</tr>
<tr>
<td>Wheel radius $(r)$</td>
<td>0.31</td>
<td>m</td>
</tr>
<tr>
<td>Referential slip ratio $(\lambda^*)$</td>
<td>0.16</td>
<td>-</td>
</tr>
<tr>
<td>Initial vehicle speed $(v(0))$</td>
<td>100</td>
<td>km/h</td>
</tr>
<tr>
<td>Vehicle speed threshold $(v_s)$</td>
<td>5</td>
<td>km/h</td>
</tr>
<tr>
<td>Step size of solver $(s)$</td>
<td>0.5</td>
<td>ms</td>
</tr>
<tr>
<td>Channel capacity</td>
<td>50000</td>
<td>bit/s</td>
</tr>
<tr>
<td>Loss probability</td>
<td>0.01%</td>
<td>-</td>
</tr>
<tr>
<td>Gaussian Variance</td>
<td>0.005*0.005</td>
<td>-</td>
</tr>
<tr>
<td>Gaussian Initial Seed</td>
<td>41</td>
<td>-</td>
</tr>
<tr>
<td>Parameter $k$</td>
<td>20</td>
<td>-</td>
</tr>
<tr>
<td>Parameter $\varepsilon$</td>
<td>0.08</td>
<td>-</td>
</tr>
<tr>
<td>Parameter $\alpha$</td>
<td>1.5</td>
<td>-</td>
</tr>
<tr>
<td>Parameter $\beta$</td>
<td>15</td>
<td>-</td>
</tr>
</tbody>
</table>

A pedal-travel subsystem is to simulate the driver’s reaction when the brake is demanded. The braking torque is related to the pedal travel, and the driver would press the pedal to the bottom in order to gain the maximum of braking torque in emergency. The reaction-imitated braking torque is shown in Fig. 7.

**B. CONTRAST SIMULATION WITH SMC WITH LP**

The linear parameterization (LP) adopted to deal with $\mu(\lambda)$ in SMC with LP is taken as a contrast. And the LP of $\mu(\lambda)$ is

$$\mu(\lambda)=\varphi^T\Phi(\lambda)$$  \hspace{1cm} (24)

Where $\varphi$ is the fitting parameters and $\Phi(\lambda)$ is the regression term in LP approximation model [19], which are

$$\varphi=[1.22\ -0.45\ 0.18\ -1.19\ -0.25]$$  \hspace{1cm} (25)

$$\Phi(\lambda)=[1\ \lambda\ e^{-9.94\lambda}\ e^{-18.43\lambda}\ e^{-65.62\lambda}]$$  \hspace{1cm} (26)

Thus, the SM controller with LP should be

$$T_s=\frac{N}{r}\left(\lambda^* - k\epsilon - \beta s - \alpha sgn(s)\right) + r F_s \varphi^T \Phi(\lambda)$$  \hspace{1cm} (27)

**C. CARSIM-MATLAB/SIMULINK Co-simulation**

(1). Ideal braking with constant $\mu$

(a). Without CAN-induced delays: The vehicle will brake longitudinally on the homogeneous road from 100 km/h without CAN-induced delays. The results are given in Fig. 8-11.
It is easy to know from Fig. 8 that the utilization of CAN bus is about 15% and there is less delay with no background-traffic disturbance from Fig. 9, which can be considered as no CAN bus induced delay in this simulation.

(b). With CAN-induced delays: This braking process is the same as the former one except the added background-traffic disturbance at 0.6s in CAN bus. The results are present from Fig. 12 to 18.

The delays are generated at 0.6s, which can be seen in Fig. 12 and Fig. 13. The utilization of CAN bus is gradually risen to about 60% when the disturbance is added, which generates the time-varying delays randomly around to 4ms, which is 8 times to the solver step.

The braking distance with SMC with TTS is shorter than that with LP than that with SMC with LP from Fig. 10. And Fig. 11 indicates that it is imprecise for the SM controller with LP, which has a big overshoot, to regulate the slip ratio to its reference well due to the nonlinear error and that the PI controller can be effective with appropriate PI parameters while it is not robust enough to deal with the impact of CAN-induced delays, which can be obtained from the following braking condition.

The braking distance with SMC with TTS is shorter than that with SMC with LP from Fig. 14, which indicates that the braking performance of proposed controller is better than the SM controller with LP.
In Fig. 15, the wheel slip ratio is out of control through PI controller while the SM controller remains robust with either TTS or LP, which verifies the robustness of SMC. Furthermore, the proposed controller presents a better performance than the other one for tracking the slip ratio well to its reference, 0.16.

The error between the estimated $\mu$ and the real $\mu$ of the proposed controller is shown in Fig. 16. It can be indicated that the estimation of $\mu$ is reliable from both front and rear wheel, where the error at both ends is big because the controller is switched off.

It can be seen from Fig. 17 that the braking torque in front wheel is larger than the braking torque in the rear wheel due to the effect of weight transfer [20].

The wheel velocity of SMC with TTS is more stable, and the vehicle took shorter time to stop through proposed controller shown in Fig. 18.

(2). Split $\mu$ braking with CAN-induced delays: This process simulates that vehicle brakes with separate road condition between left and right wheels ($\mu_l = 0.5$ on the left while $\mu_l = 0.7$ on the right). The results are present from Fig. 19-23.

Fig. 20 indicates that the SMC with LP failed to regulate the wheel slip ratio while it is still under controlled through SMC with TTS shown in Fig. 22.
FIGURE 20. Wheel slip ratio of SMC with LP (solid: right wheel; dashed: left wheel)

FIGURE 21. Wheel slip ratio of SMC with TTS (solid: right wheel; dashed: left wheel)

FIGURE 22. Braking torque of SMC with TTS (solid: front left wheel; dashed: front right wheel; dotted: rear left wheel; dashed and dotted: rear right wheel)

FIGURE 23. Velocity of split $\mu$ (solid: SMC with TTS; dashed: SMC with LP; dotted: PI controller).

From Fig. 22, it can be concluded that the four-wheels controllers counterbalance against each other to keep the vehicle from drift. There are some oscillations in the left wheel shown in Fig. 21 and 23 due to the comparatively small $\mu_H$, and it does not degrade the braking performance, which indicates the robustness of the proposed controller under this condition.

(3). Opposite $\mu$ braking with CAN-induced delays: This process simulates that the road condition turns into another while vehicle is braking. (it turns from $\mu_H = 0.5$ at start into $\mu_H = 0.7$ then in simulation). The results are present from Fig. 24-27.

FIGURE 24. Braking distance (solid: SMC with TTS; dashed: SMC with LP)

FIGURE 25. Wheel slip ratio (solid: front wheel of SMC with TTS; dashed: rear wheel of SMC with TTS; dotted: front wheel of SMC with LP; dashed and dotted: rear wheel of SMC with LP)

The performance of SMC with LP is not stable enough under this condition. In addition, the crest of front wheel appears before that of rear one, which confirms to the assumption of opposite $\mu$. 
It is obvious that the SMC with TTS presents a better performance than that with LP. Furthermore, the braking distance and time via different controllers under all 4 braking conditions is synthesized in Tab. II and III, which indicate that the performance of SMC will be improved with TTS generally, and the proposed controller is more robust.

### TABLE II

<table>
<thead>
<tr>
<th>Conditions</th>
<th>SMC with LP</th>
<th>SMC with TTS</th>
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</thead>
<tbody>
<tr>
<td>Without delay (m)</td>
<td>82.89</td>
<td>79.16</td>
</tr>
<tr>
<td>With delays (m)</td>
<td>83.20</td>
<td>78.18</td>
</tr>
<tr>
<td>Split $\mu$ (m)</td>
<td>84.99</td>
<td>77.46</td>
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<tr>
<td>Opposite $\mu$ (m)</td>
<td>69.32</td>
<td>63.63</td>
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### TABLE III

<table>
<thead>
<tr>
<th>Conditions</th>
<th>SMC with LP</th>
<th>SMC with TTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without delay (s)</td>
<td>5.69</td>
<td>5.04</td>
</tr>
<tr>
<td>With delays (s)</td>
<td>5.71</td>
<td>5.05</td>
</tr>
<tr>
<td>Split $\mu$ (s)</td>
<td>6.10</td>
<td>5.93</td>
</tr>
<tr>
<td>Opposite $\mu$ (s)</td>
<td>5.01</td>
<td>4.82</td>
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</table>

### REFERENCES


### V. CONCLUSIONS

In this paper, a TTS-designed torque controller is proposed for electric vehicle with four in-wheel motors. It is combined with two subsystems over different time scales, which is designed separately with vehicle motion over the slow time scale and the slip-friction model over the fast time scale. The wheel slip ratio can be regulated via a sliding mode and the intricate nonlinear term $\mu(\lambda)$ is circumvented by an observer to estimate the value of friction. The braking control supervisor is a combination of slip ratio calculator and the trigger of proposed controller. This proposed controller is robust to friction uncertainties, measuring error, different road conditions and even message delivery delays caused by CAN traffic. Through co-simulating with MATLAB/Simulation and CarSim, all the brake torques and wheel slip ratios are well constrained.
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