# Sliding Mode Slip Suppression Control of Electric Vehicles

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Keywords: Electric Vehicle, Slip Ratio, Sliding Mode Control, Energy Saving.

Abstract: In this paper, a new SMC (sliding mode control) method for the slip suppression control of EVs (electric vehicles) is proposed. The proposed method aims to improve the maneuverability, the stability and the low energy consumption of EVs by controlling the wheel slip ratio. The proposed method is the extended SMC method adding the integral term to improve the control performance. There also include numerical simulation results to demonstrate the effectiveness of the method.



## 1 INTRODUCTION

In recent years, Electric Vehicles (EVs) have attracted great interests as a powerful solution against the environment and energy problems(Brown et al., 2010; Mousazadeh et al., 2009; Hirota et al., 2011; Kondo, 2011).

EVs are automobiles which are propelled by electric motors, using electrical energy stored in batteries or another energy storage devices. Electric motors have several advantages over (internal-combustion engines) ICEs:

- Energy efficient. Electric motors convert 75% of the chemical energy from the batteries to power the wheels - ICEs only convert 20% of the energy stored in gasoline.
- Environmentally friendly. EVs emit no tailpipe pollutants, although the power plant producing the electricity may emit them. Electricity from nuclear-, hydro-, solar-, or wind-powered plants causes no air pollutants.
- Performance benefits. Electric motors provide quiet, smooth operation and stronger acceleration and require less maintenance than ICEs.
- Reduce energy dependence. Electricity is a domestic energy source.

The travel distance per charge for EV has been increased through battery improvements and using regeneration brakes, and attention has been focused on improving motor performance. The following facts are viewed as relatively easy ways to improve maneuverability and stability of EVs.

1. The input/output response is faster than for gasoline/diesel engines.
2. The torque generated in the wheels can be detected relatively accurately
3. Vehicles can be made smaller by using multiple motors placed closer to the wheels.

Much research has been done on the stability of general automobiles, for example, ABS (Anti-lock-Braking Systems), TCS (Traction-Control-Systems), and ESC (Electric-Stability-Control)(Zanten et al., 1995) as well as VSA (Vehicle-Stability-Assist)(Kin et al., 2001) and AWC (All-Wheel-Control) (Sawase et al., 2006). What all of these have in common is that they maintain a suitable tire grip margin and reduce drive force loss to stabilize the vehicle behavior and improve drive performance.

When the vehicle is starting off or accelerating, particularly on a slippery or wet road surface, the wheel spins easily, which causes unstable driving situation and large waste of energy. Therefore, it's important to keep the optimal driving force in all driving situation for motion stability and saving energy. During acceleration, the driving force of wheel directly depends on the friction coefficient between road and tire, which is in accordance with the wheel slip and road conditions. For this reason, it becomes possible to give the adequate driving force by controlling the wheel traction.

Several methods have been proposed for the trac-

Li S. and Kawabe T..
Sliding Mode Slip Suppression Control of Electric Vehicles.
DOI: 10.5220/0004411000110018
In *Proceedings of the 10th International Conference on Informatics in Control, Automation and Robotics* (ICINCO-2013), pages 11-18
ISBN: 978-989-8565-71-6

tion control by using slip ratio of EVs (Kodama et al., 2004), (Mubin et al., 2006), (Fujii and Fujimoto, 2007), such as the method based on Model Following Control (MFC) in (Hori, 2000) and Model Predictive PID method (MP-PID) in (Kawabe et al., 2011). Both of these methods show good performances under the nominal conditions where the situation, for example, mass of vehicle, road condition, and so on, is not changed. To meet the high performance even variation happened in the conditions, it is significant to construct the robust method against the situation changing. About this point, Sliding Mode Control (SMC) has been performed good robustness for the systems with uncertainties or nonlinearities. However, for slip ratio control with the conventional SMC, the control performance will get degradation due to the chattering which always occurs because of switching the control inputs due to the structure of SMC. To overcome such disadvantages of conventional SMC method, new SMC method with introducing the integral term to the design of the sliding surface in order to get better control performance and save more energy for slip suppression of EVs with changing the mass of vehicle and road condition is proposed. The numerical examples show the effectiveness of the proposed method.

## 2 PRELIMINARIES OF SMC

Consider the single input nonlinear system (Slotine et al., 1991)

$$x^{(n)} = f(x) + b(x)u \tag{1}$$

where $x = \left[x\ \dot{x}\ ...\ x^{(n-1)}\right]^T$ is the state vector and $u$ is the control input. In general, the function $f(x)$ and the control gain $b(x)$ are not exactly known but the extents of the imprecision on $f(x)$ and $b(x)$ are upper bounded by known. The control problem is to seek a solution that is robust to uncertainties in $f(x)$ and $b(x)$. Firstly, we defined a time-varying surface $s(x;t)$ in the state space $R^{(n)}$ by

$$s(x;t) = \left(\frac{d}{dt} + \alpha\right)^{n-1} \tilde{x} = 0,\ \alpha > 0 \tag{2}$$

where $\tilde{x} = x - x^* = \left[\tilde{x}\ \dot{\tilde{x}}\ ...\ \tilde{x}^{(n-1)}\right]^T$ is the error between the output state $x$ and the desired state $x^*$. The problem of tracking $x = x^*$ is equivalent to remaining on the surface $s$ for all $t > 0$. When $s = 0$, that is to say, the output state reaches the surface which represents the error is zero. Here, $s = 0$ is called sliding surface. On this surface the error will converge to zero exponentially. When $\dot{s} = 0$, the state is controlled to slide on the sliding surface, which is described that the system is in sliding mode.

The SMC law contains two parts, the equivalent control $u_{eq}$ and the hitting control $u_{ht}$, which is defined as follows,

$$u = u_{eq} + u_{ht}. \tag{3}$$

$u_{eq}$ can be interpreted as the continuous control law which would maintain $\dot{s} = 0$ when the dynamics are exactly known. When the dynamics are not exactly known, such as the uncertainties occur in the system or the state of system is off the sliding surface, $u_{ht}$ acts to bring the state back to the sliding surface and keeps it in sliding mode. Generally, $u_{ht}$ uses a discontinuous function to realize the switching action on sliding surface.

For choosing the control input $u$, it is necessary to consider the sliding condition (Eker and AKinal, 2008), which is defined as

$$\frac{1}{2}\frac{d}{dt}s^2 \leq -\eta|s| \tag{4}$$

where $\eta > 0$. From eq. (4) , $s^2$ shows that the squared "distance" to the sliding surface, which decreases along all system trajectories. Particularly, once the states reach the surface, the system trajectories remain on the surface. In other words, satisfying the sliding condition makes the trajectories reach the surface in finite time, and once on the manifold, it cannot leave it. Furthermore, eq. (4) also implies that some dynamic uncertainties can be tolerated while still keeping the surface an invariant set.

To realize the concept of SMC, we always need to follow two steps:

**[Step** 1 ] Design a sliding surface $s$ which is invariant of the controlled dynamics.

**[Step** 2 ] Choose the control input $u$ which drives the states to the sliding surface in sliding mode in finite time.

## 3 ELECTRIC VEHICLE DYNAMICS

As a first step toward practical application, this paper restricts the vehicle motion to the longitudinal direction and uses direct motors for each wheel to simplify the one-wheel model to which the drive force is applied. In addition, braking was not considered this time with the subject of the study being limited to only when driving.

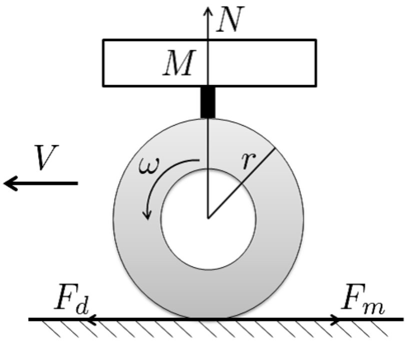


Figure 1: One-wheel car model.

From fig. 1, the vehicle dynamical equations are expressed as eqs. (5) to (7).

$$M\frac{dV}{dt} = F_d(\lambda) - F_a - \frac{T_r}{r} \quad (5)$$

$$J\frac{d\omega}{dt} = T_m - rF_d(\lambda) - T_r \quad (6)$$

$$F_m = \frac{T_m}{r} \quad (7)$$

$$F_d = \mu(c,\lambda)N \quad (8)$$

Where $M$ is the vehicle weight, $V$ is the vehicle body velocity, $F_d$ is the driving force, $J$ is the wheel inertial moment, $F_a$ is the resisting force from air resistance and other factors on the vehicle body, $T_r$ is the frictional force against the tire rotation, $\omega$ is the wheel angular velocity, $T_m$ is the motor torque, $F_m$ is the motor torque force conversion value, $r$ is the wheel radius, and $\lambda$ is the slip ratio. The slip ratio is defined by (9) from the wheel velocity ($V_\omega$) and vehicle body velocity ($V$).

$$\lambda = \begin{cases} \dfrac{V_\omega - V}{V_\omega} & \text{(accelerating)} \\ \dfrac{V - V_\omega}{V} & \text{(braking)} \end{cases} \quad (9)$$

$\lambda$ during accelerating can be shown by (10) from fig. 1.

$$\lambda = \frac{r\omega - V}{r\omega} \quad (10)$$

The frictional forces that are generated between the road surface and the tires are the force generated in the longitudinal direction of the tires and the lateral force acting perpendicularly to the vehicle direction of travel, and both of these are expressed as a function of $\lambda$. The frictional force generated in the tire longitudinal direction is expressed as $\mu$, and the relationship between $\mu$ and $\lambda$ is shown by (11) below, which is a formula called the Magic-Formula(Pacejka and Bakker, 1991) and which was approximated from the data obtained from testing.

$$\mu(\lambda) = -c_{road} \times 1.1 \times (e^{-35\lambda} - e^{-0.35\lambda}) \quad (11)$$

Where $c_{road}$ is the coefficient used to determine the road condition and was found from testing to be approximately $c_{road} = 0.8$ for general asphalt roads, approximately $c_{road} = 0.5$ for general wet asphalt, and approximately $c_{road} = 0.12$ for icy roads. For the various road conditions ($0 < c < 1$), the $\mu - \lambda$ surface is shown in fig. 2. It shows how the friction coefficient

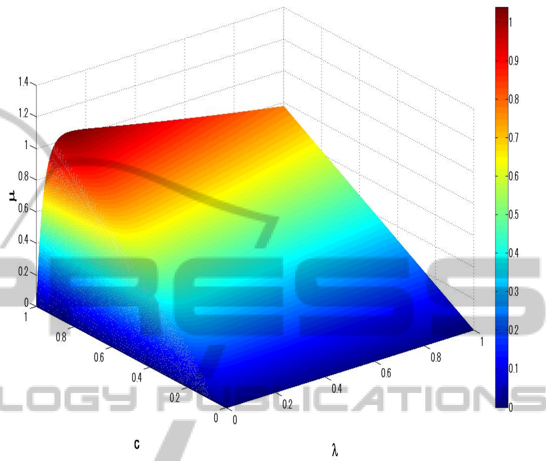


Figure 2: $\lambda$-$\mu$ surface for road conditions.

$\mu$ increases with slip ratio $\lambda$ ($0.1 < \lambda < 0.2$) where it attains the maximum value of the friction coefficient. As defined in (8), the driving force also reaches the maximum value corresponding to the friction coefficient. However, the friction coefficient decreases to the minimum value where the wheel is completely skidding. Therefore, to attain the maximum value of driving force for slip suppression, it should be controlled the optimal value of slip ratio. the optimal value of $\lambda$ is derived as follows.

Choose the function $\mu_c(\lambda)$ defined as

$$\mu_c(\lambda) = -1.1 \times (e^{-35\lambda} - e^{-0.35\lambda}). \quad (12)$$

By using (12), (11) can be rewritten as

$$\mu(c,\lambda) = c_{road} \cdot \mu_c(\lambda). \quad (13)$$

Evaluating the values of $\lambda$ which maximize $\mu(c,\lambda)$ for different $c (c > 0)$, means to seek the value of $\lambda$ where the maximum value of the function $\mu_c(\lambda)$ can be obtained. Then let

$$\frac{d}{d\lambda}\mu_c(\lambda) = 0 \quad (14)$$

and solving equation (14) gives

$$\lambda = \frac{log100}{35 - 0.35} \approx 0.13. \quad (15)$$

Thus, for the different road conditions, when $\lambda \approx 0.13$ is satisfied, the maximum driving force can be gained.

Namely, from (11) and fig. 2, we find that regardless of the road condition (value of $c$), the $\lambda-\mu$ surface attains the largest value of $\mu$ when $\lambda$ is the optimal value 0.13.

# 4 SMC METHOD WITH INTEGRAL ACTION FOR SLIP SUPPRESSION

In this section, for slip suppression of EVs, the proposed control strategy based on SMC with introducing the integral term is explained. Without loss of generality, one wheel model mentioned above is used for design of the control laws. The system dynamics can be written as

$$\dot{\lambda} = f + bT_m \tag{16}$$

where $\lambda \in R$ is the state of system representing the slip ratio of driven wheel which is defined as eq. (10) for the case of acceleration. $T_m$ is the control input.

Differentiating eq. (10) with respect to time

$$\dot{\lambda} = \frac{-\dot{V} + (1-\lambda)\dot{V}_w}{V_w} \tag{17}$$

and substituting eqs. (5), (6) and (8) into eq. (17), the following equations can be attained,

$$f = -\frac{g}{V_w}\left[1 + (1-\lambda)\frac{r^2 M}{J_w}\right]\mu(c,\lambda) \tag{18}$$

$$b = \frac{(1-\lambda)r}{J_w V_w}. \tag{19}$$

The control objective is to control the value of the slip ratio to the constant reference value $\lambda^*$.

Actually, the mass of vehicle often changes with the number of passengers and the weight of luggage. Besides, the vehicle has to always travel on many kinds of road surfaces. As a result, the controller needs to perform much robustly with the uncertainties happened in the mass of vehicle and road surface conditions which are represented by $M$ and $c$ respectively. The ranges of variation of $M$ and $c$ are set as

$$\begin{gathered} M_{min} \le M \le M_{max} \\ c_{min} \le c \le c_{max}. \end{gathered} \tag{20}$$

Consider the system eq. (16), the nonlinear function $f$ is not exactly known, but it can be estimated as $\hat{f}$. The estimation error on $f$ is assumed to be bounded by a known function $F = F(\lambda)$,

$$\left|\hat{f} - f\right| \le F. \tag{21}$$

The uncertainty in $f$ is due to the parameter $M$ and $c$. Accordingly, by using eq. (18) the estimation of $f$ can be defined as

$$\hat{f} = -\frac{g}{V_w}\left[1 + (1-\lambda)\frac{r^2\hat{M}}{J_w}\mu(\hat{c},\lambda)\right] \tag{22}$$

where $\hat{M}$ is the estimated value of $M$ and $\hat{c}$ is estimated for $c$.

Here, we define the estimated values of these parameters respectively by using the arithmetic mean of the value of the bounds as

$$\begin{aligned} \hat{M} &= \frac{M_{min} + M_{max}}{2} \\ \hat{c} &= \frac{c_{min} + c_{max}}{2}. \end{aligned} \tag{23}$$

From these definitions, the error in estimation can be given by

$$\begin{aligned} \left|f - \hat{f}\right| \le \frac{g}{|V_w|}\Big\{&\left|\mu(c_{max},\lambda) - \mu(\hat{c},\lambda)\right| \\ &+ (1-\lambda)\frac{r^2}{J_w}\left|M_{max}\mu(c_{max},\lambda) - \hat{M}\mu(\hat{c},\lambda)\right|\Big\} \end{aligned} \tag{24}$$

Then, let

$$\begin{aligned} F = \frac{g}{|V_w|}\Big\{&\left|\mu(c_{max},\lambda) - \mu(\hat{c},\lambda)\right| \\ &(1-\lambda)\frac{r^2}{J_w}\left|M_{max}\mu(c_{max},\lambda) - \hat{M}\mu(\hat{c},\lambda)\right|\Big\}. \end{aligned} \tag{25}$$

## 4.1 Design of Sliding Surface

Letting $\tilde{\lambda}$ be the variable of interest, then the order of system is assumed to be one. The sliding function of conventional SMC can be given by

$$s_c(\lambda,t) = \tilde{\lambda} \tag{26}$$

where $\tilde{\lambda}$ is the error between the actual slip ratio and the reference value, which is defined as $\tilde{\lambda} = \lambda - \lambda^*$.

By adding an integral item to the sliding function $s_c$, the new sliding function $s$ can be designed as

$$s(\lambda,t) = \tilde{\lambda} + K_i\int_0^t \tilde{\lambda}(\tau)d\tau \tag{27}$$

where $K_i$ is the integral gain, $K_i > 0$.

## 4.2 Derivation of Control Law

In this section, the sliding mode controller is derived to make the slip ratio $\lambda$ to track the reference slip ratio $\lambda^*$. The sliding mode occurs when the state $\lambda$ reaches the sliding surface defined by $s = 0$. The dynamics of sliding mode is governed by

$$\dot{s} = 0. \tag{28}$$

Differentiating eq. (27) and substituting the result into eq. (28) give

$$\left(\dot{\lambda}-\dot{\lambda^*}\right)+K_i\left(\lambda-\lambda^*\right)=0. \quad (29)$$

The reference slip ratio $\lambda^*$ is a constant, thus $\dot{\lambda^*}=0$. Substituting eq. (16) into eq. (29) gives

$$f+bT_m+K_i(\lambda-\lambda^*)=0 \quad (30)$$

and solving eq. (30) gives equivalent control input as

$$T_{meq}=\frac{1}{b}\left[-f-K_i\left(\lambda-\lambda^*\right)\right] \quad (31)$$

then the estimate of the equivalent control input can be obtained as

$$\hat{T}_{meq}=\frac{1}{b}\left[-\hat{f}-K_i\left(\lambda-\lambda^*\right)\right]. \quad (32)$$

For satisfying sliding condition (make state in the sliding mode) despite uncertainty on the dynamics $f$, the hitting control input is defined as

$$T_{mht}=\frac{1}{b}\left[-K\mathrm{sgn}(s)\right] \quad (33)$$

where

$$\mathrm{sgn}(s)=\begin{cases}-1 & s<0\\ 0 & s=0\\ 1 & s>0\end{cases} \quad (34)$$

and $K$ is called sliding gain. Thus, the control law can be given by

$$\begin{aligned}T_m &= \hat{T}_{meq}+T_{mht}\\ &= \frac{1}{b}\left[-\hat{f}-K_i(\lambda-\lambda^*)-K\mathrm{sgn}(s)\right]. \quad (35)\end{aligned}$$

When no uncertainty in the system (i.e., no variation in $c$ and $M$), $T_{mht}$ is desired to be 0. Because eq. (35) contains the estimate of the equivalent control $\hat{T}_{meq}$, $T_m$ keeps the state on the sliding surface ($s=0$ i.e., $\lambda=\lambda^*$). Because of the uncertainties in the system, the state $\lambda$ could deviate from the sliding surface. The hitting control acts to return the state back to the sliding surface which implies the robustness of SMC.

Here, the sliding gain $K$ is chosen as

$$K = F+\eta \quad (36)$$

with the value of $F$ given by eq. (25).

Then choose a Lyapunov function as

$$V = \frac{1}{2}s^2 \quad (37)$$

and differentiate eq. (37) with respect to time, that gives

$$\begin{aligned}\dot{V} &= \frac{1}{2}\frac{d}{dt}s^2\\ &= s\dot{s}. \quad (38)\end{aligned}$$

Substituting eqs. (16), (28), (32), (33) and (35) into eq. (38) yields

$$\begin{aligned}\dot{V} &= s\dot{s}\\ &= s\left[f-\hat{f}-K\mathrm{sgn}(s)\right]\\ &= s(f-\hat{f})-K|s|\\ &\le F|s|-K|s|\\ &\le -\eta|s|. \quad (39)\end{aligned}$$

Thus, the control law introduced in eq. (35) can guarantee the stability of the system in the Lyapunov sense under variations. Concretely, the stability of the system is guaranteed with an exponential convergence once the sliding surface is encountered, if the sliding condition is satisfied. So eq. (39) guarantees the strategy can converge to the sliding surface in finite time if the error is not zero, that is to say, slip ratio can be controlled to the reference value in finite time whenever the uncertainties occur in the system.

### 4.3 Chattering Reduction

For sliding mode control design, the switched controller limits switching to a finite frequency, which produces chattering. To reduce the chattering, the hitting control $T_{mht}$ can be rewritten by using the saturation function

$$T_{mht}=\frac{1}{b}\left[-K\mathrm{sat}\left(\frac{s}{\Phi}\right)\right] \quad (40)$$

where $\Phi>0$ is a design parameter representing the width of the boundary layer around the sliding surface $s=0$ and the saturation function is defined as

$$\mathrm{sat}\left(\frac{s}{\Phi}\right)=\begin{cases}-1 & s<-\Phi\\ \frac{s}{\Phi} & -\Phi\le s\le\Phi\\ 1 & s>\Phi\end{cases}. \quad (41)$$

Thus, using eqs. (35), (36) and (40), the control law of the system by the proposed SMC can be rewritten as

$$T_m=\frac{1}{b}\left[-\hat{f}-K_i\left(\lambda-\lambda^*\right)-(F+\eta)\mathrm{sat}\left(\frac{s}{\Phi}\right)\right]. \quad (42)$$

## 5 NUMERICAL EXAMPLES

This section shows the numerical simulation results to demonstrate the effectiveness of the proposed method. In all simulations, we consider three different road conditions, a dry asphalt for $t\in[0,2)[s]$, an icy road for $t\in[2,8)[s]$ and a wet asphalt for $t\in[8,10][s]$. The width of the boundary layer $\Phi$ defied in eq. (40) is set to 1. In eq. (42), the proposed SMC law can be calculated with the values of design parameters $K_i$ and $\eta$, which both impact on the steady state accuracy. Here, for confirm the energy conservation performance of the proposed method, the values of both parameters are set $K_i = 10$ and $\eta = 5$, which are determined by several tests.

By using eqs. (26), (28), (33) and (36), the control law of the conventional SMC can be derived as

$$T_{mc} = \frac{1}{b}\left[-\hat{f} - (F+\eta)\text{sat}\left(\frac{s}{\Phi}\right)\right]. \tag{43}$$

In the conventional SMC, the parameters $\eta = 1$ and $\Phi = 1$. The value of parameters used in the simulations are listed in Table 1.

Table 1: Parameters used in the simulations.

| | |
|---|---|
| $M$:Mass of vehicle | $1100[kg]$ |
| $J_w$:Inertia of wheel | $21.1[kg/m^2]$ |
| $r$:Radius of wheel | $0.26[m]$ |
| $\lambda^*$:Reference slip ratio | $0.13$ |
| $g$:Acceleration of gravity | $9.81[m/s^2]$ |



As the input to the simulation of system, the torque is produced by the constant pressure on the accelerator pedal, which is decided on the vehicle speed desired by the driver. Here, the vehicle speed is desired to achieve $180[km/h]$ in $15[s]$ by a fixed acceleration after starting the car. The range of variation in mass of vehicle $M$ and road condition coefficient $c$ are imposed as $M_{max} = 1400[kg]$, $M_{min} = 1000[kg]$, $c_{max} = 0.9$ and $c_{min} = 0.1$ respectively. So the nominal values of mass and road condition coefficient can be obtained as $\hat{M} = 1200[Kg]$ and $\hat{c} = 0.5$ .

## 5.1 Robust Performance

In order to verify the robustness of proposed SMC with variation both in the mass of vehicle and road condition, the variation in the mass of vehicle is made by assigning the value of $M$ to $1000[kg]$, $1100[kg]$, $1200[kg]$, $1300[kg]$ and $1400[kg]$ respectively. Fig. 3 shows the responses of slip ratio with different masses can converge to the reference value under the variation in the road condition. It is known that when the mass gets the nominal value $1200[kg]$, in the first $2[s]$, the response is more accurately than the car with other mass. But after $2[s]$, the performance drops down with the mass increases.

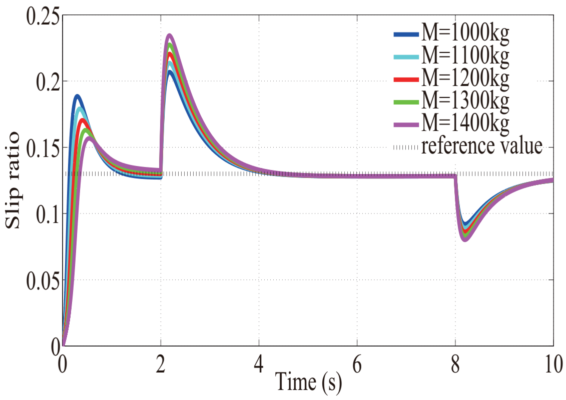


Figure 3: Slip ratio by the proposed method.

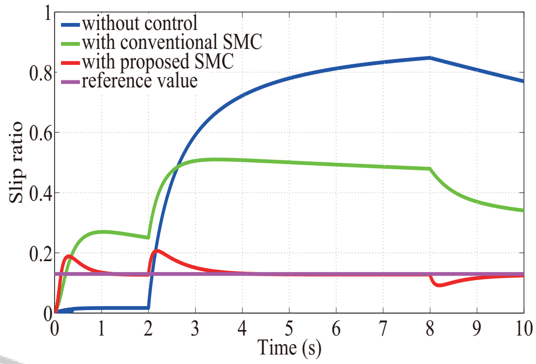


Figure 4: Slip ratio ($M = 1000[kg]$).

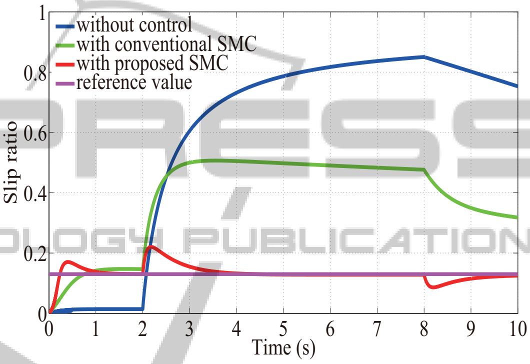


Figure 5: Slip ratio ($M = 1200[kg]$).

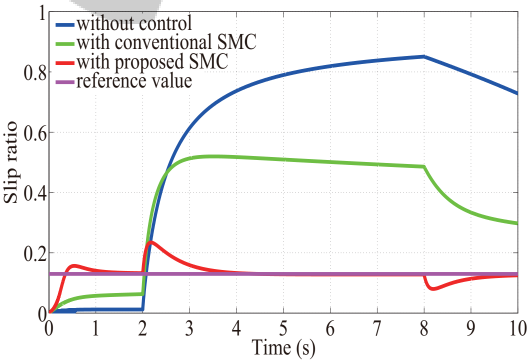


Figure 6: Slip ratio ($M = 1400[kg]$).

Next, we compared the proposed SMC with the conventional SMC and no control. Figs. 4~6 show the responses of slip ratio under three different road conditions for three different masses respectively.

The responses with proposed SMC can suppress the slip ratio to the reference value 0.13 accurately in a very short time whenever both of the mass and road condition are changing. In addition, the slip ratio with the conventional SMC does not converge to the reference value because of the steady state error. When the car starts off at $0[s]$ or runs into an icy road at $2[s]$, the slip ratio response using control method grows with the increasing wheel speed as a result of too much torque generated. As the car travels from icy road to wet asphalt in $8[s]$, the slip ratio decreases with the decreasing wheel speed, when the torque generated at that time cannot satisfy the one required on the wet asphalt. The car without control is to make the slip ratio to 0, so at the first stage the response is converged to 0. However, when the car runs into the ice road at $2[s]$, the wheel spins out of control resulting that the wheel speed in-creasing suddenly, which leads to a large slip ratio value. Therefore, we can see that the proposed SMC has a good performance against the variation in both of the mass of vehicle and road condition.

## 5.2 Acceleration Performance

It is different from the simulation condition described in previous that the simulations are executed under unchanging road condition with mass every time.

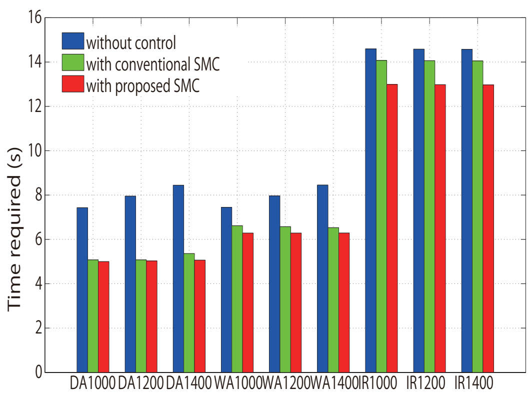


Figure 7: Acceleration performance.

Fig. 7 shows the time required for $100m$ by the car with different control method. The x-axis label indicates the cases of different road condition and mass, for example, $DA1000$ says the car with $M = 1000[kg]$ is driving on the dry asphalt, $WA1200$ shows the case with $M = 1200[kg]$ on the wet asphalt and $IR1400$ is the case with $M = 1400[kg]$ on the icy road. As shown in fig. 7, it takes minimum time by the proposed method in every case. So we can see that the car with proposed SMC have gained the best acceleration. In other words, the results also indicate the vehicle with the proposed SMC can keep the loss of driving force at a minimum.

## 5.3 Energy Consumption

To confirm the effectiveness of the proposed SMC for energy saving, we compare it to the conventional SMC and no control method. Generally, It's difficult to evaluate the energy consumption accurately without measurement by experiments on the real vehicle. In this paper, therefore, we estimate the energy cost by calculating the rotational energy of motor. As a beginning, we give the following assumptions;

Assumption 1: the electric power is all used to drive the wheel.

Assumption 2: the power consumed by the vehicle is in proportional with the rotational energy due to the rotation of driven wheel.

The rotational energy $E_r$ is defined by the rotational inertia of wheel $J_w$ and the angular velocity $w$ as

$$E_r = \frac{1}{2} J_w w^2. \tag{44}$$

Under these assumptions, we calculate the energy consumed in the simulations in 5.1.

Fig. 8 shows the results of electric energy consumed by different mass case. From this fig., we can see that the proposed SMC consumes minimum energy in every case. The car without control takes most energy because the spin of wheel on the icy road in $t \in [2, 8)[s]$ leads to much energy loss. As the mass increases, the amount of energy cost decreases because the car suppresses the spin of wheel by increment of mass to get more driving force. Conversely, the energy consumption with the proposed SMC and conventional SMC increases due to the rising cost of control as the mass increases. From this perspective, it also shows that an EV should be made more light to save more energy.

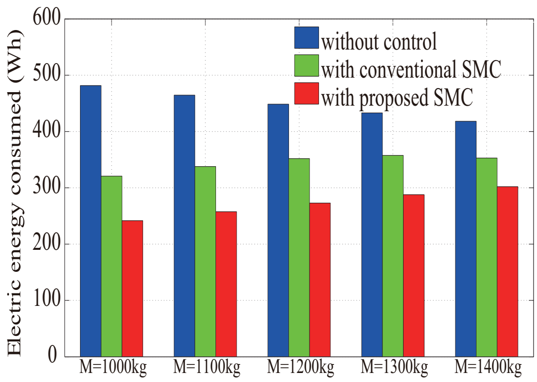


Figure 8: Energy consumption.

## 6 CONCLUSIONS

This paper proposes new SMC method with the integral action for EV traction control. The method can improve the robust performance of EV traction by controlling the slip ratio with low energy consumption against the variation of mass of vehicle and road conditions. We can verified that the the proposed method shows good robust performance with low energy consumption by comparing to conventional method.

As future works, in this paper, the gain $K_i$ of integral action added in the sliding function was determined by trial and error, so it is necessary to develop a systematic method to find the optimal value of $K_i$. Moreover, this paper was limited to showing the results with some example conditions using a simplified one wheel model, but to make the method practical, for a variety of road conditions and mass of vehicle must be verified for more detailed two-wheel and four-wheel models. In addition, the suitability of the proposed method must be studied not only for the slip suppression addressed by this paper but also for overall driving including during braking.

Even for this issue, however, the basic framework of the proposed method can be used as is and can also be extended relatively easily to form a foundation for making practical high performed robust traction control systems with low energy consumption for EVs by promoting further progress.

## ACKNOWLEDGEMENTS

This research was partially supported by Grant-in-Aid for Scientific Research (C) (Grant number: 24560538; Tohru Kawabe; 2012-2014) from the Ministry of Education, Culture, Sports, Science and Technology of Japan.

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