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Electric vehicle yaw moment control based on the body sideslip estimation

***Abstract**. The direct yaw moment control (DYC) presents an operative solution to improve the stability and road holding of vehicles, in particular electric vehicles equipped with independent motors. The torque control applied to each wheel can improve the handling performance of a vehicle making it safer and faster and in critical driving situations. In this article, a new method proposed for the control of the direct yaw moment based on the sliding mode control. This method uses a new design of switching function to simultaneously track the desired yaw rate and the side slip of the vehicle. The lateral sideslip angle of the vehicle is estimated by using a Kalman filter. The results of the comparative simulations show the effectiveness of the proposed method with the other conventional methods in terms of following the reference yaw rate, the vehicle trajectory and the vehicle skidding in various difficult driving scenarios*

***Streszczenie.** Bezpośrednie sterowanie momentem odchylenia (DYC) stanowi funkcjonalne rozwiązanie poprawiające stabilność i przyczepność pojazdów, w szczególności pojazdów elektrycznych wyposażonych w niezależne silniki. Kontrola momentu obrotowego zastosowana do każdego koła może poprawić właściwości jezdne pojazdu, czyniąc go bezpieczniejszym i szybszym w krytycznych sytuacjach na drodze. W niniejszym artykule zaproponowano nową metodę sterowania bezpośrednim momentem odchylającym w oparciu o sterowanie trybem ślizgowym. Metoda ta wykorzystuje nową konstrukcję funkcji przełączania do jednoczesnego śledzenia pożądanej wartości odchylenia i poślizgu bocznego pojazdu. Kąt bocznego znoszenia pojazdu jest szacowany za pomocą filtra Kalmana. Wyniki symulacji porównawczych pokazują skuteczność proponowanej metody z innymi metodami konwencjonalnymi w zakresie śledzenia referencyjnego kursu zbaczania, toru jazdy i poślizgu pojazdu w różnych trudnych scenariuszach jazdy (**Sterowanie momentem odchylenia pojazdu elektrycznego w oparciu o oszacowanie poślizgu bocznego nadwozia**)*

Keywords: Electric Vehicle, Sliding Mode Control, yaw moment control, Kalman Filter.
Słowa kluczowe: pojazd elektryczny, sterowanie ślizgowe, filtr Kalmana

Introduction

Recently and with the rapid development of electric vehicles, a new generation of DYC methods has been developped based on generating a yaw moment using the individual tractive forces produced by independently driving electric motors. This new type DYC stands as a promising addition to conventional DYC systems for several reasons: First, DYC differential braking only works during critical driving conditions when the vehicle is about to lose control, while the system based on DYC traction is capable of continuously producing differential traction forces to regulate vehicle movement. Second, the motor torque is measurable and its generation is fast and precise [1]. In the electric braking mode, the motor torque becomes negative, assists the hydraulic braking system, and improves energy efficiency through regenerative braking. In general, the DYC solution use yaw rate, vehicle sideslip body, or both states simultaneously as the primary control variable [2]. The yaw rate plays a crucial role in controlling vehicle dynamics and it should be selected as one of the primary control variables in vehicle stability and handling control for the following reasons: Firstly, the steady state yaw rate is directly dependent on the driver's steering input. Secondly, this steady-state value defines the steer characteristic (i.e., under, over, or neutral steer) of the vehicle. However, for the continuous control of the dynamics of the vehicle, knowledge of the parameters is essential. For reasons of cost and other practical issues, some parameters such as the sideslip angle of the body β must be estimated [3]. To improve and ensure the robustness of DYC, sliding mode control (SMC) has been widely adopted, which is robust and suitable for nonlinear systems such as vehicles with considering the presence of model uncertainties, system parameter variations and external disturbances [4]. The main contribution of this paper is proposing a basic method to estimate the sideslip angle of the body β, based on a Kalman filter using the linear bicycle model. This document is organised as follows. In the second section, the vehicle and electric motor and drive models are presented. A controller design based on sliding mode control is discussed in the third section. The estimation of the sideslip angle using the Kalman filter is discussed in the fourth section, and in the fifth section, the simulation results are provided. Finally, the findings are presented in the sixth section.

Overall System Modeling

A. Vehicle model

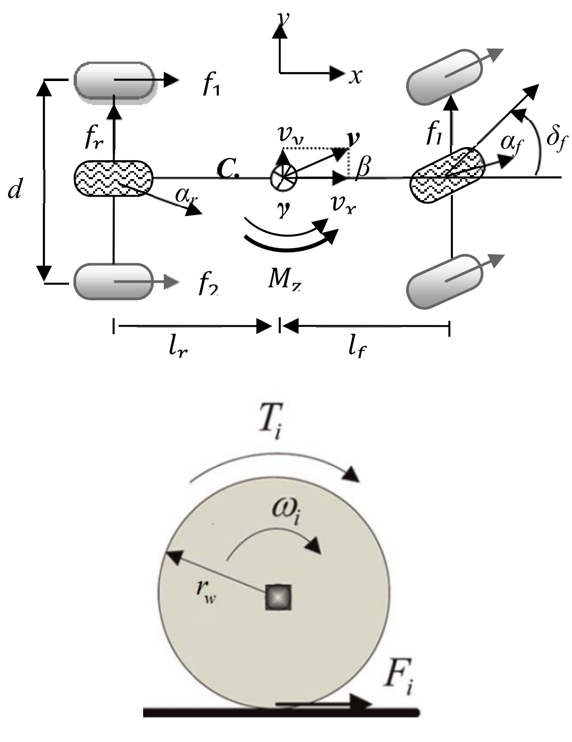


Fig 1. Model of the rear wheel drive vehicle and applied forces

Figure (1) shows the planar model of an electric vehicle with two electric motors placed at the rear wheels. This vehicle model has two degree-of-freedom (DOF), i.e., the yaw and lateral motions. However, the roll, vertical, and pitch motions are not considered here due to neglecting the suspension system.

As expected, the electric vehicle is propelled only by the two rear driving wheels; we are interested in the rear traction forces f_1 and f_2. For the design of the control system, the governing equations for lateral and yaw dynamics are:

(1) $$\begin{cases} m(\dot{v}_x - v_y\gamma) = f_1 + f_2 \\ m(\dot{v}_y - v_x\gamma) = f_r(\alpha_r) + f_f(\alpha_f) \\ I_z\dot{\gamma} = l_f f_f(\alpha_f) - l_r f_r(\alpha_r) + \frac{d}{2}(f_1 - f_2) \end{cases}$$

For small steering and sideslip angles, we have:

(2)
$$f_f = -C_f\alpha_f; \qquad f_r = -C_r\alpha_r$$
$$\alpha_f = \frac{v_y + l_f\gamma}{v_x} - \delta_f; \quad \alpha_r = \frac{v_y - l_f\gamma}{v_x}; \quad \beta \simeq v_y/v_x$$

Eqs (1) and eqs (2) yield a state-space equation as follows:

(3) $$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx \end{aligned}$$

Where

(4) $$\begin{bmatrix}\dot{\beta} \\ \dot{\gamma}\end{bmatrix} = \begin{bmatrix} -\frac{C_r + C_f}{mv_x} & \frac{C_r l_r - C_f l_f}{m{v_x}^2} - 1 \\ \frac{C_r l_r - C_f l_f}{I_z} & -\frac{C_r {l_r}^2 + C_f {l_f}^2}{I_z} \end{bmatrix} \begin{bmatrix}\beta \\ \gamma\end{bmatrix} + \begin{bmatrix} \frac{C_f}{mv_x} & 0 \\ \frac{C_f l_f}{I_z} & \frac{1}{I_z} \end{bmatrix} \begin{bmatrix}\delta_f \\ M_z\end{bmatrix}$$

ad

(5) $$M_z = \frac{1}{2}d(f_1 - f_2)$$

m denotes the mass of the body, I_z the moment of inertia concerning the yaw motion, β the side slip angle, $\dot{\gamma}$ the yaw angular acceleration, v_x the longitudinal velocity, δ_f the steering angle of front wheel, l_f, l_r the distances from the center of gravity to the front and rear axles respectively, C_f, C_r the cornering stiffness coefficients of the front and rear wheels respectively M_z the yaw moment applied by differential braking, which must be determined from the control law.

B. Electric motor and drives model

Furthermore, since the dynamic responses of modern motor drives are much faster than wheel dynamics, and considering the dominant poles of the closed loop system, an electric motor and its drive can be simply modeled as follows [5]:

(6) $$\frac{T_i}{T_i^*} = \frac{k}{(1+s\tau_o)(1+s\tau_e)}$$

where τ_o is the delay due to inverter and τ_e is the electrical time constant of electric motor.

The tractive and braking forces transmitted from the road to tires are the products of the torques being attached to the drive wheels. Therefore we have to add two dynamic equations of rear wheels to the system (1). Each of them is commonly described by:

(7) $$J_\omega\dot{\omega} = T_i - f_i(\lambda)r_\omega$$

where J_ω, is inertia moment of the wheel; T_i is wheel torque drive of the *i*th wheel; *w* is rotary speed; *r*, is radius of the wheel; $f_i(\lambda)$ is the driving force of the *i*th wheel and depending on the slip ratio:

(8) $$\lambda = (r_\omega\omega - v_x)/\max(r_\omega\omega, v_x)$$

Moreover, considering that the friction coefficient μ_i between tire and road is a function of λ_i, the driving force can be represented as:

(9) $$f_i = \mu_i(\lambda)N_i$$

where N_i normal force.

Controller Designer

A. Reference model:

In order to improve stability of the vehicle, the sideslip angle and the yaw rate of the vehicle are controlled to trace their desired values. The direct yaw moment generated by the longitudinal forces is employed as the control input to make actual responses approach the desired values.
Rewrite equation (1) as:

(10) $$\dot{x} = Ax + BM_z + H\delta_f$$

Where :

(11) $$x = \begin{bmatrix}\beta \\ \gamma\end{bmatrix}, A = \begin{bmatrix}a_{11} & a_{12} \\ a_{21} & a_{22}\end{bmatrix}, B = \begin{bmatrix}b_1 \\ b_2\end{bmatrix}, H = \begin{bmatrix}h_1 \\ h_2\end{bmatrix}$$

The reference model is designed to acquire the zero side slip angle at the center of gravity with respect to arbitrary front steering angle. But in the reference model the yaw rate response is approximated to be the first order system as [4]:

(12) $$\gamma_d(s) = \frac{k_1}{(1+k_2 s)}\delta_f(s); \qquad k_1 = \frac{-h_1}{a_{12}}, k_2 = \frac{-1}{a_{22}}$$

Then the reference model can be expressed by the state space form:

(13) $$\dot{x}_d = A_d x_d + H_d\delta_f$$

where

$$x_d = \begin{bmatrix}\beta_d \\ \gamma_d\end{bmatrix}, A_d = \begin{bmatrix}0 & 0 \\ 0 & -1/k_2\end{bmatrix}, H_d = \begin{bmatrix}0 \\ k_1/k_2\end{bmatrix}$$

B. Sliding mode control:

Before we design a sliding mode controller, we review the basic design methodology, (Slotine and Lee, 1991). Consider the single-input nonlinear system:

(14) $$y^{(n)} = f(x) + b(x).u$$

where *u* is the control input and $x = [x, \dot{x}, \dots x^{(n-1)}]^T$ is the state vector. *f(x)* is a nonlinear function of the state and *b* is the controller gain. The control objective is to track a desired trajectory,

$x_d = \left[x_d, \dot{x}_d, \dots x_d^{(n-1)}\right]^T$ [6].

Define a time-varying surface *S(t)* in the state-space by a scalar function *s(x ,t)=0*, where

(15) $$(x,t) = \left(\frac{d}{dt} + \varepsilon\right)^{n-1} e$$

where ε is a positive constant and tracking error can be expressed as:

(16) $$e = x - x_d$$

The problem of tracking $x = x_d$ is equivalent to remain on the surface *S(t)* for all *t>0*. The problem of keeping the scalar function s(x,t) at zero can be achieved by choosing the control law *u* such that:

(17) $$\frac{1}{2}\frac{d}{dt}s^2 \le -\eta|s|$$

where η is a design parameter that decides the response time of the system.

The sliding mode controller consists of two components, one that keeps the error at zero and one that guarantees the error reaches zero. The sliding mode control law is

(18) $$u = u_{eq} + k_s.sgn(s)$$

The striking control k_s is solved from (14) [Lyapunov condition]:

(19) $$\frac{1}{2}\frac{d}{dt}s^2 = \dot{s}.s \le -\eta|s|$$

Substituting the terms (16) at the right side of (10) and (13):

$$\dot{e} = \dot{x} - \dot{x_d} = (Ax - A_d x_d) + BM_z + (H - H_d)\delta_f$$

(20) $$= A(x - x_d) + (A - A_d)x_d + BM_z + (H - H_d)\delta_f$$
$$= Ae + (A - A_d)x_d + BM_z + F\delta_f$$

where $F = (H - H_d) = \begin{bmatrix} h_1 \\ h_2 - \frac{a_{22}}{a_{12}} h_1 \end{bmatrix}$

The first object to design a SM controller is to choose a sliding surface. Note that the actual sideslip angle and the yaw rate should approach their ideal value. Hence, the sliding surface of SMC is defined as:

(21) $$s = e_2 + \varepsilon e_1 = (\gamma - \gamma_d) + \varepsilon(\hat{\beta} - \beta_d)$$

The equivalent u_{eq} , is solved from:

(22) $$\dot{s} = \dot{e} = \dot{e}_2 + \varepsilon\dot{e}_1 = 0$$

Using equations (18),(19) and (21) into (22), the sliding mode control law can be designed as:

(23) $$M_z = \frac{-1}{b_2}\left[|(\varepsilon a_{11} + a_{21})e_1| + |(\varepsilon a_{12} + a_{22})e_2| + |(\varepsilon f_1 + f_2)\delta_f| + |(\varepsilon a_{12}\gamma_d)| + \eta\right] sgn(s)$$

To smooth out the input signal, *Sgn(s)* is replaced with sat(s/Φ), as suggested in (Slotine and Lee, 1991), where Φ is the boundary layer thickness [7].

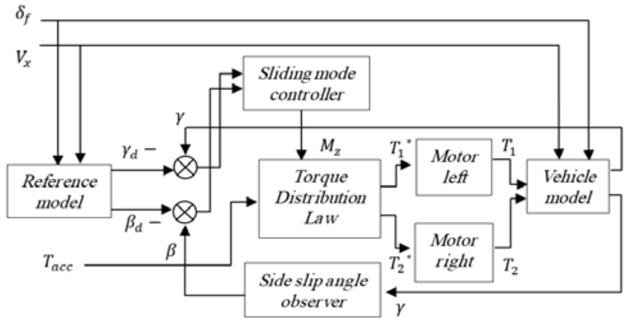


Fig. 2. Structure of control system.

C. Torque distribution law

The control input M_z, obtained by Sliding mode controller is distributed to two motors based on the following equations:

(24) $$\begin{cases} M_z = \frac{1}{2}d({f_1}^* + {f_2}^*) \\ T_{acc} = r_w({f_1}^* + {f_2}^*) \end{cases}$$

d is the half distance between left and right wheels.

By solving the simultaneous equation (23), the force command signals to left and right wheels (${f_1}^*; {f_2}^*$) can be determined from M_z, and accelerator command T_{acc}. Thus, the torque commands to two in-wheel motors can be calculated as [8]:

(25) $$\begin{cases} {T_1}^* = r_w {f_1}^* \\ {T_2}^* = r_w {f_2}^* \end{cases}$$

Using eq (25) into eq(24), we can obtain:

(26) $$\begin{cases} {T_1}^* = \frac{d}{2r_w}(M_z - T_{acc}) \\ {T_2}^* = \frac{d}{2r_w}(M_z + T_{acc}) \end{cases}$$

Side Slip Angle Estimator

In many ways, Kalman filter is the dual of LQ control, which is a celebrated “optimal” observer/filter/prediction design method.

In this section, a basic dynamic-model-based method based on a Kalman filter utilizing the linear bicycle model given in Eq. (4) will be introduced [9].

As the system model contains uncertainties and the sensor measurements are contaminated by noise, process noise $v(t)$ and measurement noise $w(t)$ are also included as:

(27) $$\dot{x}(t) = Ax(t) + Bu(t) + v(t)$$
$$y(t) = Cx(t) + w(t)$$

Recall that for time invariant systems, the design method for state feedback can be used to design a state estimator (observer).

In According to the linear control theory, the sideslip angle observer can be constructed as:

(28) $$\dot{\hat{x}}(t) = (A - LC)\hat{x}(t) + Ly(t) + Bu(t)$$

where :

$$\hat{x} = \begin{bmatrix} \hat{\beta} \\ \hat{\gamma} \end{bmatrix}, A = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, L = \begin{bmatrix} L_1 \\ L_2 \end{bmatrix}.$$

Estimation error can be expressed by:

(29) $$e(t) = \hat{x}(t) - x(t)$$

Using eq(27) and eq(28) into eq(29), the estimation error can be designed as:

(30) $$\dot{e}(t) = (A - LC)e(t) + Lw(t) - v(t)$$

In order to reduce the estimation error, the matrix gain L may be set large, At the same time, the observation noise (sensor noise) is emphasized. It is necessary to determine the optimum value.

The properties of process noise and observation noise are supposed to be uncorrelated with each other, their mean values are zero, and the covariance matrices are Q and R, respectively. The setting of this covariance matrix is related to the estimated speed of the Kalman filter [10].

The Kalman filter gain L is the optimum observer gain that minimizes the squared average error of the difference $e(t)$ between the estimated value of the state quantity and the true value in infinite time.

By solving the Riccati equation, it can be obtained as follows [11].

(31) $$PA^T + AP - PC^T R^{-1} CP + Q = 0$$

where P is solution of Riccati equation.

The Kalman Filtering gain can be designed as:

(32) $$L = PC^T R^{-1}$$

Results and discussions

In order to verify the effectiveness of proposed control algorithm, two steering wheel manoeuvres change on the right and the left, are investigated by Matlab/Simulink simulation. Vehicle parameters for simulation are shown in Table1 and electric drive (DC motor) parameters for simulation are shown in Table 2.

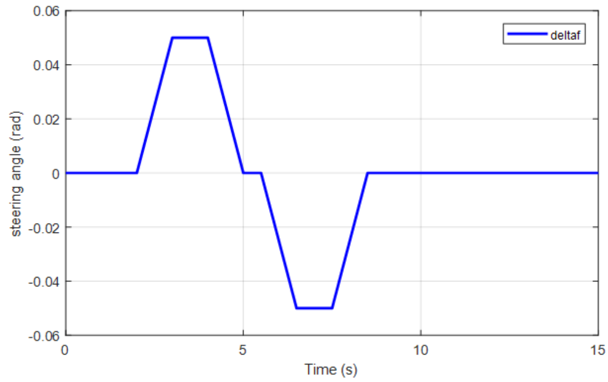
The simulation results are obtained for a severe cornering manoeuvre. The vehicle speed is 80 km/h. The steering angle input is shown in Fig.3(a). The parameters of the whole vehicle model and electric motor in simulation are shown in Table 1 and 2 respectively. Additionally, due to the discontinuous property of SMC, controller appears heavy chattering. The simulation results of Fig.3 (b) and fig.3 (c) indicate that the state values of the yaw rate and slip angle for the sliding mode controlled are considerably less than those for the uncontrolled vehicle. The yaw rate response can follow the desired response. Therefore, the sliding mode controller can considerably improve the safety of the vehicle. Fig.3(e) shows the yaw moment control input changing between [-4000,4000 Nm]. The rear torque motor and motor speed shows in Fig.3(e) and 4(f) proved a good control of the electric vehicle on the steering manoeuvre. The blue solid line in Fig.3(d) Illustrates the simulation results of body-slip-angle estimation using a Kalman filter with correct vehicle parameters that improve estimation performance of this filter.

Table 1. Vehicle parameters.

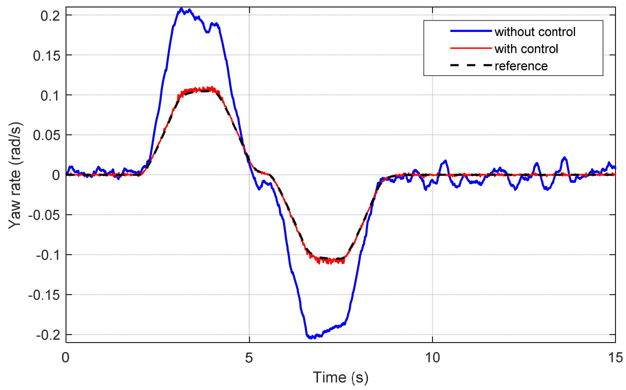
Parameters	Design
m = 1980	Vehicle mass
iz = 3758	Vehicle yaw moment of inertia
lf = 1.358	Distance from CG to front axle
lr = 1.472	Distance from CG to rear axle
cf = 41000	Front tire cornering stiffness
cr = 74000	Rear tire cornering stiffness
n = 20	Steering gear ratio
v = 80/3.6	Vehicle speed
d = 1.7	Vehicle wide

Table 2. Electric drives parameters

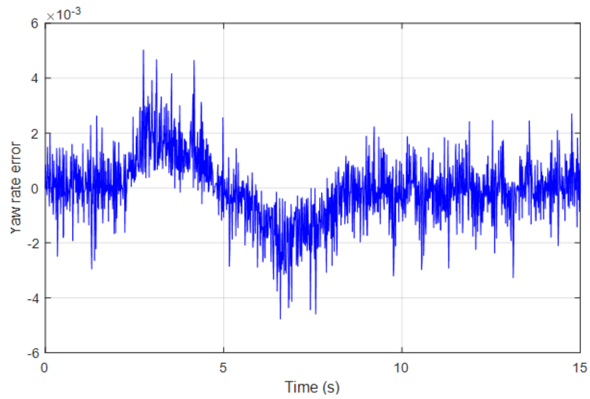
Parameters	Design
R=0.1 Ω	Resistance
L=0.01 H	inductance
K=0.01	EMF constant
J=0.8	Wheel inertia.
Gr=1:10	Gear box ratio



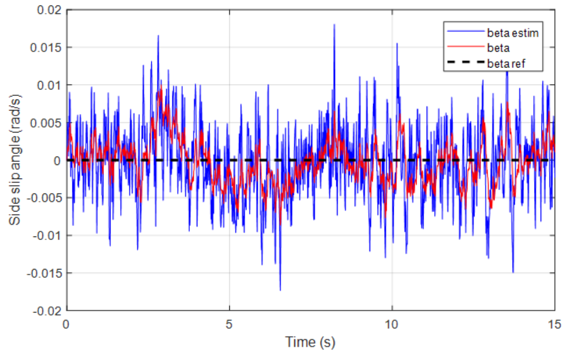
(a) Steering angle



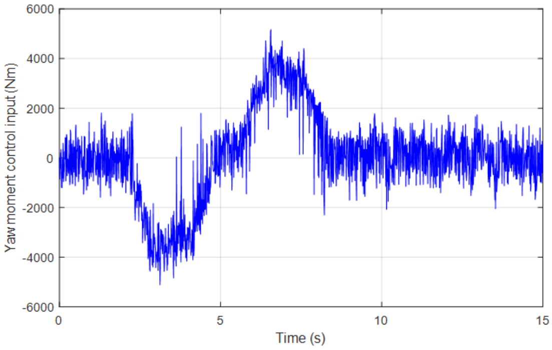
(b) Yaw rate



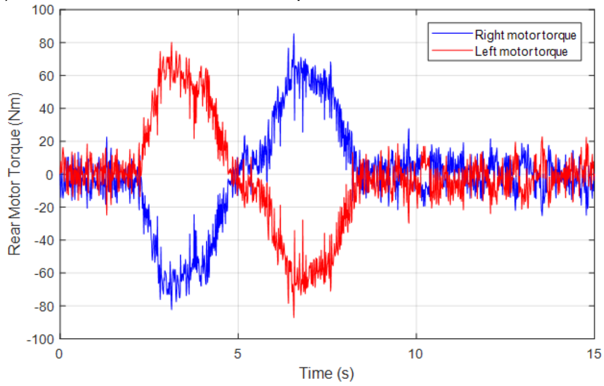
(c) Yaw rate error



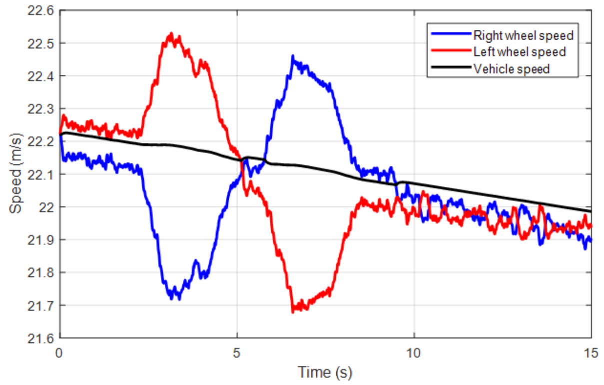
(d) Side slip angle



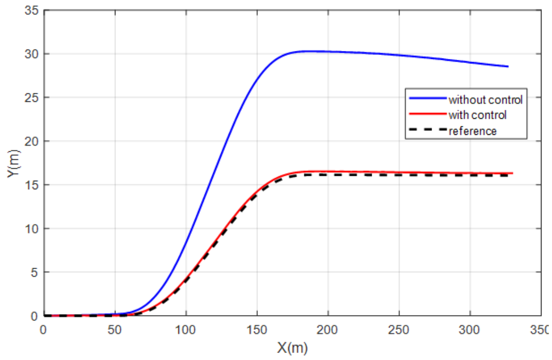
(e) Yaw moment control input



(f) Rear Motors torque



(g) Vehicle and wheel drive speed



(h) Trajectories of vehicle

Fig 3. Vehicle response under severe cornering.

Conclusion

The vehicle controller presented in this paper only requires the braking or driving force to generate the direct yaw moment control input. Considering the development of electric vehicles, the proposed control approach becomes very promising and more practical.

Because the corrective yaw moment can be generated by controlling the driving/braking torques of individual motors, the motor-drive EVs are considered as the best choice in terms of yaw stability control in different driving conditions. The simulation results show better performances in the yaw rate and slip angle response, and robustness ensured by the yaw moment control, therefore the sliding mode controller can improve the safety of the vehicle effectively. It was found that the lateral acceleration is reduced, and the yaw rate can perfectly track the reference one. Also, the vehicle sideslip angle is reduced and restricted which is of central importance in terms of vehicle handling stability. As a result, it can be concluded that the vehicle handling stability is greatly improved using proposed controller.

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