A MODEL BASED ON PREVIEW MECHANISM OF DRIVER FOR THE ELECTRONIC STABILITY PROGRAM

Gu JUN[1], Lin XIAONING[2], Huang MANHONG[3]

Reference model is the benchmark for vehicle dynamics control. However, the time delay from the driving intention to the reference model output will aggravate the instability of vehicle under limit working conditions. Therefore, a reference model based on preview mechanism of driver is proposed to reduce the influence of the time delay. Firstly, the preview mechanism of driver is used to compute the predicted steering wheel angle, which can enhance the driving intention. Then, the reference model is established by combining the steering angle of driver input and the predicted steering wheel angle. Secondly, the yaw moment controller is designed according to the sliding control theory and the required yaw moment is allocated according to the quadratic programming theory. Finally, simulation results show that the vehicle stability control system based on the proposed reference model has a better performance than the traditional control system, and the proposed method is able to improve the control intervention instant.

Keywords: Electronic Vehicle, ESP System, Reference Model, Driving Intention

1. Introduction

Electronics Stability Program (ESP) system is one kind of new active security technology. Recently, ESP system has been a new vehicle safety standard for all light vehicle. During the transient steering process, the output value of reference model is regarded as the most important driving intention variable to realize the driving operation, which is the tracking goal for ESP system.

In order to obtain a good maneuverability, the reference model has been studied and different models were put forward by the researchers [1-3]. According to human drivers are familiar with the linear handling characteristics, Nagai *et al*. [1] proposed a reference model for ESP system, which is based on the Tow Degree of Freedom (2DOF) vehicle model. Moreover, Geng C *et al*. [2] modified

[1] College of Information and Mechanical and Electrical Engineering, Jiangsu Open University, Nanjing Jiangsu 210019, China; E-mail:gujun_nj@126.com
[2] College of Information and Mechanical and Electrical Engineering, Jiangsu Open University, Nanjing Jiangsu 210019, China.E-mail:linhsiaon@126.com
[3] College of Energy and Power Engineering, Nanjing University of Aeronautics and Astronautics, Nanjing Jiangsu 210016, China.E-mail:huangmanhong@yeah.net

the 2DOF reference model by considering the road adhesion to limit the lateral acceleration. But these reference models cannot stabilize the vehicle adequately under limited working conditions. Therefore, Jonasson *et al*. [3] proposed a reference model that is constrained by side slip angle to improve the vehicle stability.

However, as the reference models continuous improvement, the vehicle has still happened to instability under limited working conditions. The reasons for this situation are the fuzziness of driving intention, the parameter uncertainty with the vehicle and so on. Therefore, the research scholars proposed the other reference models aiming to solve these problems [4-5]. Raksincharoensak *et al*. [4] used Hidden Markov Model to predict the driver's steer intention, and enhanced the reference model's driving intention via this information. Xiong *et al*. [5] proposed a tire cornering stiffness estimation method to ensure the control robustness against environmental disturbances as well as parameter uncertainty, which can modify the output value of reference model. However, even if the reference model could be properly modified, a vehicle dynamics stability controller might still not achieve a satisfactory result, because there is a delay while the output value of reference model response to the driving intention.

A number of tests have shown the existence of that delay. The delay mainly caused by the neural response of driver and steering system of vehicle, and varies from 100ms to 300ms depending on the driver's reaction time, maneuvering conditions and the road conditions [6]. However, the control intervention instant can be brought forward to some degree with a driver model. Chen *et al*. [7] proposed a new method to enhance the driving intention of reference model based on a single point preview driver model [8], which can predict the driver's intention. However, papers about the driver model to improve the control intervention are seen rarely. Thus, this paper proposes a novel reference model based on the driver's preview mechanism. The reference model can enhances its driving intention by introducing prediction of steering wheel angle, which can be obtained via a direction preview driver model [9]. In order to minimize the influence on the driver's operation, it uses fuzzy control method to establish the reference model.

This paper is arranged as follows. The second section describes the specific method of the proposed ESP system. Simulation results of the two ESP systems are shown, and a comparison is made between these two kinds of ESP system in the third section. Finally, conclusions are drawn in the fourth section.

2. ESP System

The proposed ESP system utilized in this paper is illustrated in Figure 1. The system includes: prediction of steering wheel angle algorithm, novel reference model, yaw moment controller and control allocation algorithm. The prediction of steering wheel angle algorithm obtains the predict steering wheel angle based on the driver model, and is used to compensate the reference model's driving intention. Then the novel reference model can be established by introducing the predicted steering wheel angle.

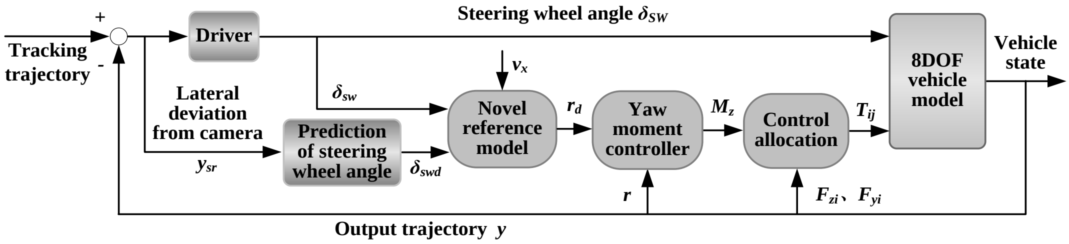


Fig. 1. Structure of ESP system

2.1. Prediction of steering wheel angle algorithm

Due to the driver model can reflect the human driver's behavior, which is based on the human driver, therefore it can be used to compensate the driving intention for the reference model. It can obtain the prediction of steering wheel angle via the preview mechanism of driver which is according to a direction preview driver model [9-10].

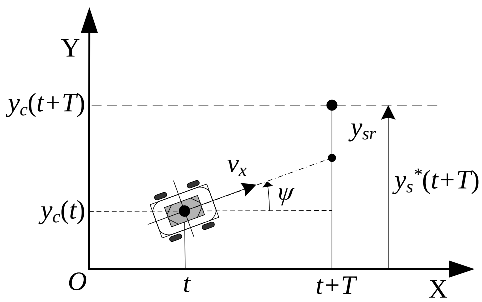


Fig. 2. The preview mechanism of driver

From Figure 2, the human driver takes a preview decision for the tracking path before the steering wheel angle has to be decided. The predicted lateral displacement at the distance in front of vehicle of l_s can be expressed as follows

$$y_c(t+T)=y_c(t)+T\cdot\dot{y}_c(t)+\frac{T^2}{2}\cdot\ddot{y}_c(t) \tag{1}$$

Where T is the preview time and is equivalent to l_s/v_x, l_s is the preview distance, ψ is the heading angle of the vehicle, $y_c(t)$ is the location of the vehicle, $y_c(t+T)$ is the desired path after T.

In the process of ideal tracking, the desired lateral displacement at the preview point and predicted lateral displacement must be equal $(y_s*(t+T)=y_c(t+T))$. Therefore, the required lateral acceleration can be calculated as follows

$$\ddot{y}_c(t)=\frac{2v_x^2}{l_s^2}\left[y_s^*(t+T)-\left(y_c(t)+l_s\frac{\dot{y}_c(t)}{v_x}\right)\right]=\frac{2v_x^2}{l_s^2}y_{sr}(t) \tag{2}$$

Where $y_{sr}(t)$ is the vehicle's lateral deviation, which is quite possible to detect the road information in advance using modern sensors. In this paper, the road information acquisition algorithm is not mentioned, and $y_{sr}(t)$ is considered to be known.

When the body side slip angle is negligible, the kinematics relationship between the yaw rate and lateral acceleration can be expressed as $\ddot{y}_c(t)=v_x\cdot r(t)$, by substituting it into Eq.(2), the desired yaw rate can be obtained by the lateral deviation as follows

$$r^*(t)=\frac{2v_x}{l_s^2}y_{sr}(t) \tag{3}$$

Using the 2DOF vehicle model, the transfer function of the steady state gain from steering wheel angle input to yaw rate can be calculated as follows

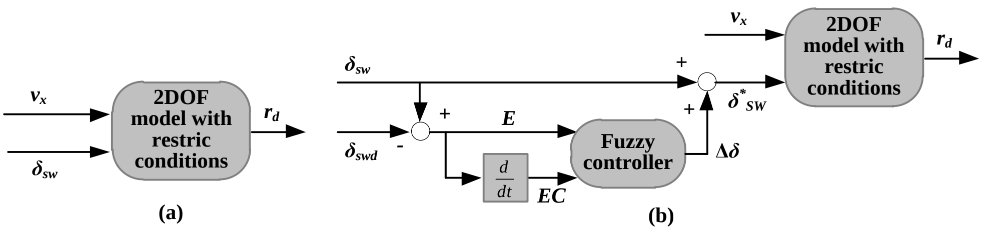
$$\frac{r}{\delta_{sw}}=\frac{v_x}{n\left(l+K_v{v_x}^2\right)} \tag{4}$$

Where $K_v=\frac{m}{l}\left(\frac{b}{C_f}-\frac{a}{C_r}\right)$ denotes the understeer gradient.

Considering Eq. (3) and (4) lead to prediction of steering wheel angle as follows

$$\delta_{swd}=\frac{2n}{l_s^2}\left(l+K_v{v_x}^2\right)y_{sr}(t) \tag{5}$$

2.2. Proposed reference model



(a) the traditional reference model; (b) the novel reference model

Fig. 3. Difference between two reference models

The role of the reference model is to calculate the desired yaw rate, which can provide the tracking goal for the yaw moment controller. In Figure 3(a), 2DOF model is widely used as a traditional reference model because human drivers are very familiar with the linear handling characteristics [1]. Based on this model, the desired yaw rate r_d can be derived from the steering wheel angle δ_{sw} and longitudinal vehicle velocity v_x as

$$r_d = \frac{v_x}{\left(l + K_v v_x^{\ 2}\right)} \cdot \frac{\delta_{sw}}{n} \tag{6}$$

Besides, assuming the friction coefficient is μ, and in order to avoid large lateral acceleration that exceeds tire cornering capability, the desired yaw rate is constrained as

$$|r_d| \le 0.85 \frac{\mu g}{v_x} \tag{7}$$

In fact, the yaw rate response to the steering angle is the first-order delay system. Therefore, the desired yaw rate can be described as follows

$$r_d = \min\left\{\left|\frac{v_x}{l + K_v v_x^2}\frac{\delta_{sw}}{n}\right|, \left|0.85\frac{\mu g}{v_x}\right|\right\} \cdot \mathrm{sgn}\left(\delta_{sw}\right) \cdot \frac{1}{1+\tau_r s} \tag{8}$$

Where τ_r the delay is time and **sgn** is a sign function.

The traditional reference models depend on the driver's output steering wheel angle, which characterizes the driving intention. Although this steering wheel angle has strong robustness and adapts to different road conditions, it limits the driver's physiological conditions and vehicle structure. This means that δ_{sw} doesn't reflect the driver's intention adequately. Compared with δ_{sw}, the predicted steering wheel angle δ_{swd} is not constrained by these conditions and is an ideal tracking goal for the vehicle, which can reduce the influence of these conditions.

However, if the predicted steering wheel angle can combine with the steering wheel angle by human driver, it could not only enhance the driving intention of reference model, but also improve the stability of vehicle. Therefore, the novel reference model proposed in this article uses fuzzy control method as it is shown in Figure 3(b). Due to fuzzy logic control can be used to deal with complicated non-linear dynamic control problems, which is a non-linear control method [11-12]. There are two main advantages of fuzzy models in comparison with conventional mathematical models. The one is the possibility of elaborating them on the basis of far fewer amounts of information about a system. The other one is that fuzzy logic enables the heuristic rule-based techniques to be extended for use in the continuously variable situation without significantly. In this context, the number of applications of fuzzy logic to vehicle has increased significantly over the last years with good results [13].

The architecture of fuzzy logic controller consists of three steps:

(i) Fuzzification

Two input variables for the fuzzy logic control are the error between the human driver's steering wheel angle and predicted steering wheel angle, i.e. $E = \delta_{sw} - \delta_{swd}$, and the error between the human driver's steering wheel angle rate and predicted steering wheel angle rate, i.e. $EC = \dot{\delta}_{sw} - \dot{\delta}_{swd}$. The output variable is the steering wheel angle adjustment $\Delta\delta$.

To provide enough rule coverage, seven fuzzy sets are used for input variables: PB (positive big), PM (positive medium), PS (positive small), ZO (zero), NB (negative big), NM (negative medium), and NS (negative small). Similarly, the output variables are also fuzzified into seven fuzzy sets: {NB, NM, NS, ZO, PS, PM, PB}.

(ii) Fuzzy decision process

It processes a list of rules from the knowledge base using fuzzy input from the previous step to produce the fuzzy output. Table 1 shows rules for the proposed fuzzy logic controller. These rules are introduced based on expert knowledge and extensive simulations performed in this study. The steering wheel angle adjustment $\Delta\delta$ mainly influences on steering wheel angle δ_{sw}. For this reason, the rules follow the next criteria:

- If $E = \delta_{sw} - \delta_{swd}$ is positive that means the vehicle is oversteer, it occurs that:
  - $\delta_{sw} > 0$ and $\delta_{swd} > 0$ with $\delta_{sw} > \delta_{swd}$. In this case, it is necessary to generate a negative $\Delta\delta$ that δ_{swd} to decrease δ_{sw}.
  - $\delta_{sw} < 0$ and $\delta_{swd} < 0$ with $\delta_{sw} < \delta_{swd}$. In this case, it is necessary to generate a positive $\Delta\delta$ that δ_{swd} to increase δ_{sw}.

- If $E = \delta_{sw} - \delta_{swd}$ is negative that means the vehicle is understeer, it occurs that:
  - δ_{sw} > 0 and δ_{swd} > 0 with $\delta_{sw} < \delta_{swd}$. In this case, it is necessary to generate a positive $\Delta\delta$ that δ_{swd} to increase δ_{sw}.
  - δ_{sw} < 0 and δ_{swd} < 0 with $\delta_{sw} > \delta_{swd}$. In this case, it is necessary to generate a negative $\Delta\delta$ that δ_{swd} to decrease δ_{sw}.
- The higher the difference between the steering wheel angle and predicted steering angle, the higher the steering wheel angle is adjusted. Meanwhile the error rate EC will be considering into the adjusting process, which is characterized the degree of steering urgency.

Table 1.

Fuzzy control rule for $\Delta\delta$

EC	E						
	PB	PM	PS	ZO	NS	NM	NB
PB	NB	NB	NB	NB	NM	NS	ZO
PM	NB	NB	NB	NM	NM	NS	PS
PS	NB	NB	NM	NS	ZO	PS	PS
ZO	NB	NM	NS	ZO	PS	PM	PB
NS	NS	NS	ZO	PS	PM	PB	PB
NM	NS	PS	PM	PM	PB	PB	PB
NB	ZO	PS	PM	PB	PB	PB	PB

Rules based on the criteria above, we can tune considering the effect on the vehicle state's response lots of simulations. The fuzzy controller uses the Mamdani Fuzzy Inference System (FIS), which is characterized by the following fuzzy rule schema:

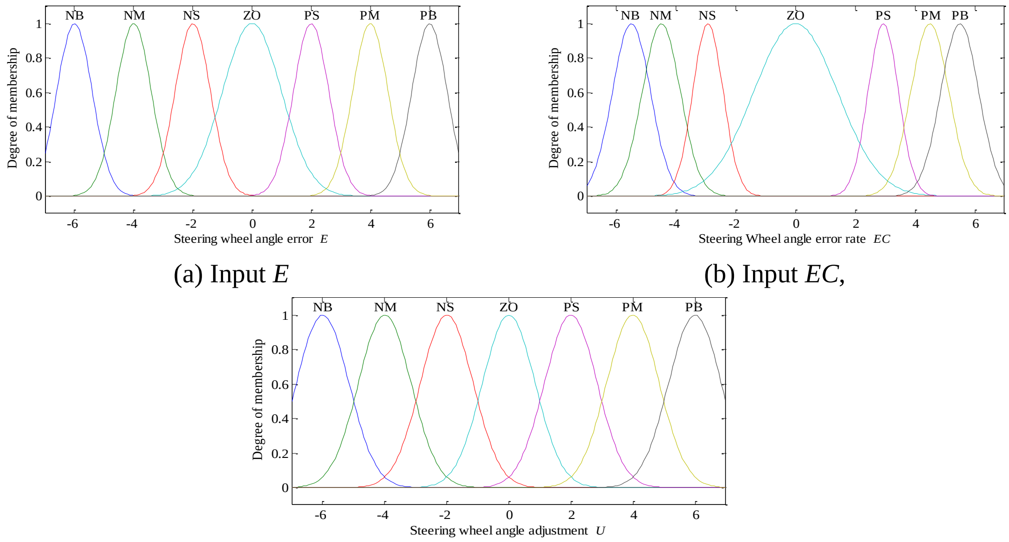
IF E is A and EC is B THEN $\Delta\delta$ is C

Where A and B are fuzzy sets defined on the input and output domains respectively.

(iii) Defuzzification

It scales and maps the fuzzy out from fuzzy decision process to produce an output, which is the input to the system controller. In this case, the steering wheel angle adjustment. The defuzzification method used in this project is the center of area. This method determines the center of the area below the combined membership function.

Figure 4 shows the membership functions and ranges of values of E, EC and $\Delta\delta$ respectively. The universe of discourse of inputs and output are normalized in the range of [-7 7].



(a) Input E (b) Input EC,

(c) Output $\Delta\delta$

Fig. 4. Membership functions of input and output variables:

After the Fuzzy controller outputs the steering wheel angle adjustment, the desired steering wheel angel can be written as

$$\delta_{sw}^{*} = \delta_{sw} + \Delta\delta \tag{9}$$

Combined with Eq. (9) and (8), the desired yaw rate described as follows

$$r_d^{*} = \min\left\{ \left| \frac{v_x}{l + K_v v_x^2} \frac{\delta_{sw}^{*}}{n} \right|, \left| 0.85 \frac{\mu g}{v_x} \right| \right\} \cdot \operatorname{sgn}\left(\delta_{sw}^{*}\right) \cdot \frac{1}{1 + \tau_r s} \tag{10}$$

2.3 Yaw Moment Controller

The yaw moment controller is to ensure that the desired yaw rate is realizable. Considering the nonlinear characteristics of vehicle, the sliding control theory is beneficial to solve this problem [14]. Therefore, this paper uses the sliding control theory to design the controller.The sliding surface S is defined as $S = r - r_d$, differentiating it and combining Eq. (10) yields, it can be written as

$$\dot{S} = \dot{e} = \dot{r} - \dot{r}_d = \frac{1}{I_z}\left(\left(c_r b - c_f a\right)\beta - \frac{c_f a^2 + c_r b^2}{v_x} + c_f b \frac{\delta_{sw}}{n} + M_z \right) - \dot{r}_d \tag{11}$$

To make the system slide to the sliding surface S, the exponential converging velocity is defined as $\dot{S} = -KS - \varepsilon \operatorname{sgn}(S)$, where K and ε are the positive gains.

Then, implementing the convergence condition of sliding mode control, the required yaw moment M_z is obtained from Eq. (11) as follows

$$M_z = I_z\left(-\frac{c_r b - c_f a}{I_z}\beta + \frac{c_f a^2 + c_r b^2}{I_z v} - \frac{c_f a}{I_z}\frac{\delta_{sw}}{n} + \dot{r}_d - KS - \varepsilon\,\mathrm{sgn}(S)\right) \tag{12}$$

According to Lyapunov stability criterion, the stability of control system needs to be proved. The Lyapunov function is structured as $V_{L2} = S_2 / 2$, then make a derivation to it and combine $S = r - r_d$ yields

$$\dot{V}_{L2} = S\dot{S} = S\left(\frac{c_r b - c_f a}{I_z}\beta - \frac{c_f a^2 + c_r b^2}{I_z v_x} + \frac{c_f b}{I_z}\frac{\delta_{sw}}{n} + \frac{M_z}{I_z} - \dot{r}_d\right) \tag{13}$$

Substituting Eq. (12) into Eq. (13) with proper simplification as follows

$$\dot{V}_{L2} = S\dot{S} = S(-KS - \varepsilon\,\mathrm{sgn}(S)) \tag{14}$$

Because S, **sgn(*s*)**, K and ε are the positive, therefore the V_{L2} is a negative and this can be proved the control system is stable.

Besides, the controller uses a saturation function to alleviate clutch chattering and improve its robustness by substituting sign function. In a similar way, Eq. (12) can be written as follows

$$M_z = I_z\left(-\frac{c_r b - c_f a}{I_z}\beta + \frac{c_f a^2 + c_r b^2}{I_z v} - \frac{c_f a}{I_z}\frac{\delta_{sw}}{n} + \dot{r}_d - KS - \varepsilon\,\mathrm{sat}(S)\right) \tag{15}$$

The saturation function is designed as follows

$$\mathrm{sat}(S/\Delta) = \begin{cases} \mathrm{sgn}(S) & |S| \geq \Delta \\ S/\Delta & |S| < \Delta \end{cases} \tag{16}$$

Where Δ is the boundary layer thickness.

2.4. Control allocation algorithm

The control allocation algorithm is designed to generate the total longitudinal force F_{xd} and yaw moment M_{zd}. It requires for the controlled vehicle to follow the reference model responses as well as to meet with the driving intention. In order to allocate reasonable to realize the control target, this paper uses the quadratic programming method to design the optimal allocation, which is an effectively method to solve the controlled problem of over-actuated systems [15].

From 8DOF vehicle model, assuming that δ_f is a small value, then $\sin\delta_f \approx 1$ and $\cos\delta_f \approx 1$. The total longitudinal force as well as the yaw moment generated by tire longitudinal forces can be expressed as follows

$$\begin{cases} F_{xfl}+F_{xfr}+F_{xrl}+F_{xrr}=F_{xd} \\ \left(F_{xfr}-F_{xfl}\right)T_{wf}/2+\left(F_{xrr}-F_{xrl}\right)T_{wr}/2=M_{zd} \end{cases} \quad \text{with } T_i=F_{xi}R_w, i=fl,fr,rl,rr \tag{17}$$

Where F_{xi} and T_i are denotes the longitudinal tire force and torque output of the in-wheel motor, respectively. T_{wf} and T_{wr} are the front and rear wheel tread, respectively.

In order to obtain an excellent control performance, the optimization objective is designed by introducing the characterization of whole vehicle road load conditions as follows

$$\min J=\sum_i^4\frac{c_iF_{xi}^2}{\mu^2F_{zi}^2}=\sum_i^4\frac{c_iT_i^2}{\mu^2R_w^2F_{zi}^2} \quad \text{with } i=fl,fr,rl,rr \tag{18}$$

Where c_i is weighting coefficient and μ is road adhesion coefficient.

The longitudinal tire forces are limited by the nominal friction coefficient and normal load. Combining Eq. (17), it can be written as follows

$$-A_i\le T_i\le A_i,\ \ (A_i=R_w\sqrt{\mu^2F_{zi}^2-F_{yi}^2}, i=fl,fr,rl,rr) \tag{19}$$

According to the above optimization objective and constrain conditions, the quadratic programming standard can be expressed as follows

$$\begin{cases} \min J=u^TWu \\ s.t.\ \ Bu=v,\ u_{\min}\le u\le u_{\max} \end{cases} \tag{20}$$

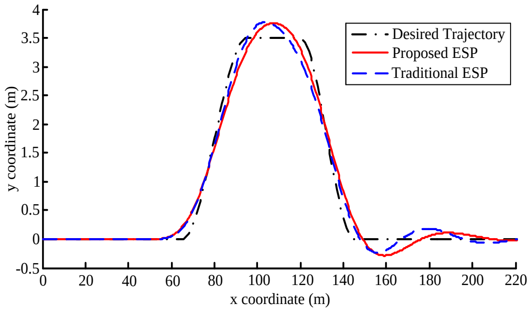
where $W=\mathrm{diag}\left(c_i/\left(uF_{zi}\right)^2\right)$, $v=(F_x \quad M_z)T$,

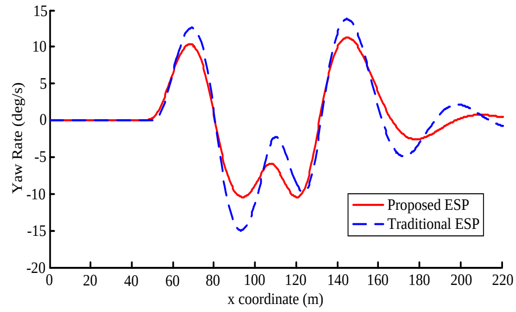
$u=(T_{fl} \quad T_{fr} \quad T_{rl} \quad T_{rr})T$, $B=\begin{pmatrix} 1/R_w & 1/R_w & 1/R_w & 1/R_w \\ -T_{wf}/2R_w & -T_{wf}/2R_w & -T_{wf}/2R_w & -T_{wf}/2R_w \end{pmatrix}$.

3. Simulation results and discussion

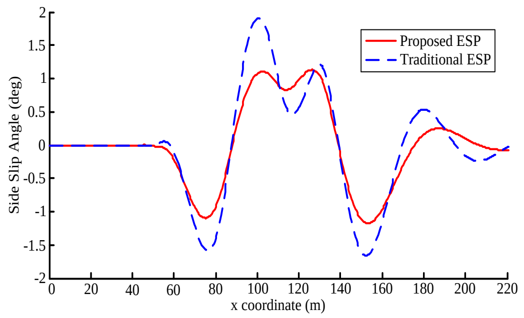
In order to analyze the performance of the strategy, computer simulation is performed utilizing the 8DOF nonlinear vehicle model [16], which established by the ‘Dugoff model’ tire model [17].

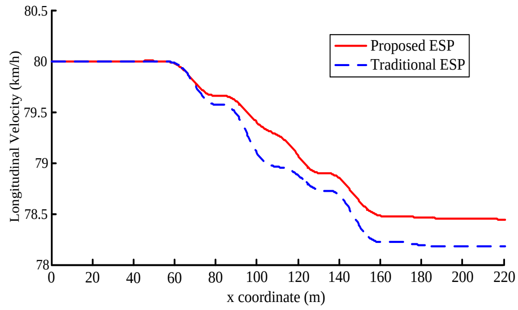
To validate the effectiveness of the strategy for ESP system, the following simulation is used, i.e. a double-lane change (DLC) simulation test, which can reflect that the driver under the emergency operation of obstacle avoidance to the response characteristics of vehicle. The responses of a vehicle with the proposed ESP system is compared with the traditional ESP system, which is used the traditional reference model as shown in Figure 5(a).



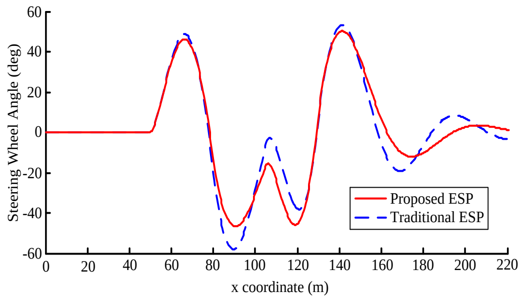


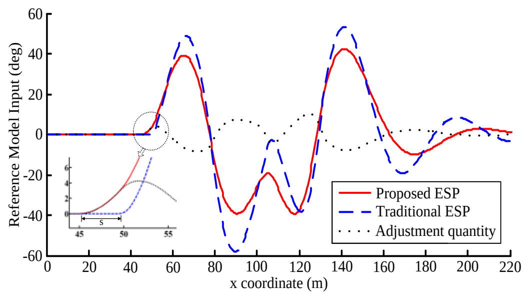
(a)Trajectory tracking, (b)Yaw rate,





(c)Side slip angle, (d) Longitudinal velocity,

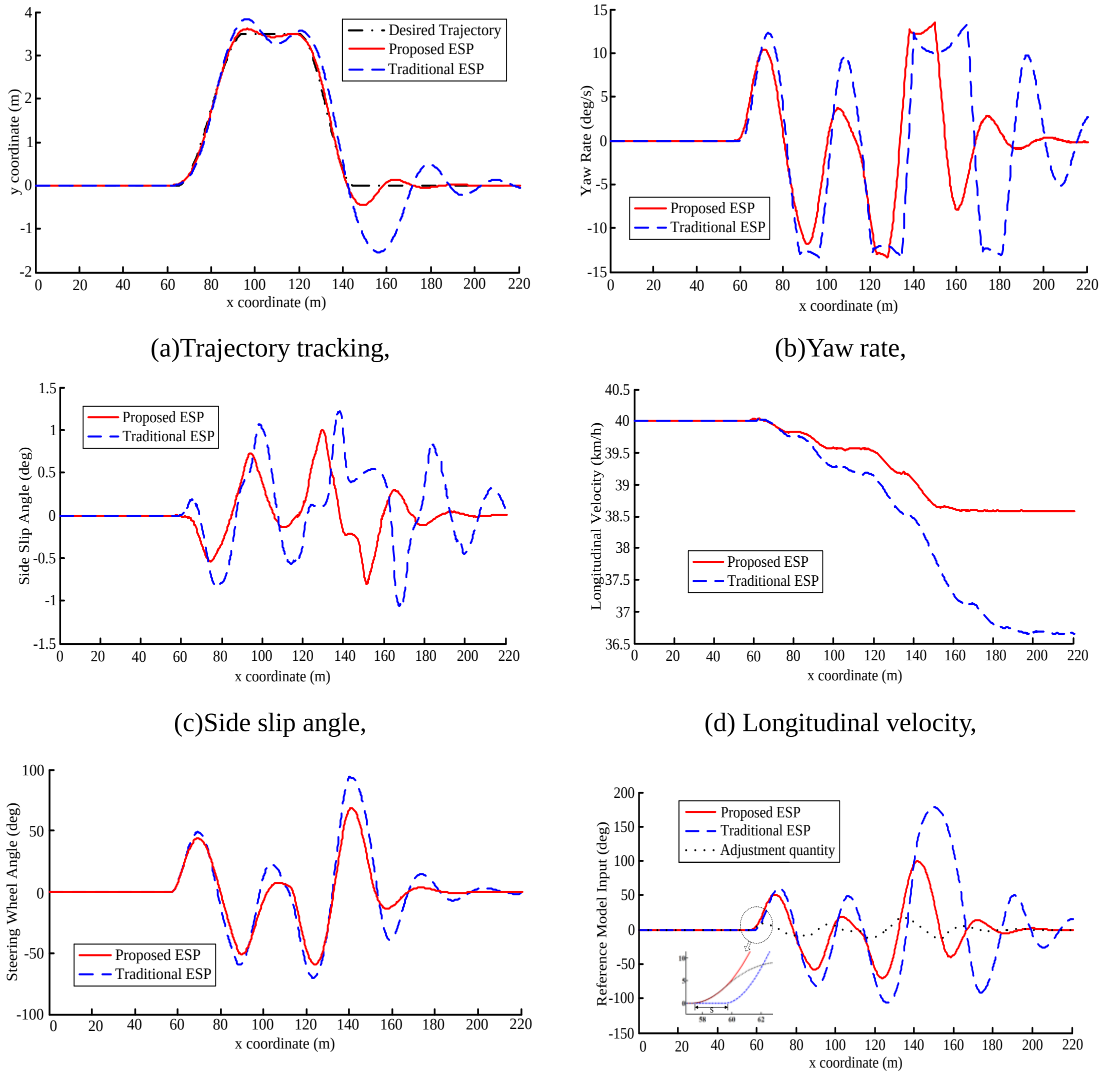




(e) Steering wheel angle, (f) Reference model input

Fig. 5. Simulation comparison (V = 80 km/m, μ = 0.8)

Fig. 5 displays the proposed and traditional ESP control results at an initial velocity of 80km/h and nominal friction coefficient μ is 0.8. From Fig. 5, both the proposed ESP and traditional system can stabilize the vehicle to pass the DLC path, the former obviously better than the latter such as Figure 5(a)~(c). Especially, in Figure 5(d) and (e), the vehicle final speed of the proposed ESP system is higher than that of the traditional ESP system and the driver manipulate the vehicle under the proposed ESP control with a smoother and smaller steering wheel angle than the traditional ESP control, which can help the driver to relieve the driving fatigue. It illustrates that proposed ESP had better performance than the tradition ESP by the timely intervention to control vehicle.



(a)Trajectory tracking, (b)Yaw rate,

(c)Side slip angle, (d) Longitudinal velocity,

(e) Steering wheel angle, (f) Reference model input

Fig. 6. Simulation comparison (V = 40 km/m, μ = 0.3)

Fig. 6 is the simulation results with an initial longitudinal velocity of

40km/h and nominal friction coefficient μ is 0.3. In this DLC condition, the proposed ESP system is evidently better than the traditional ESP system from Figure 6. As Figure 6(a)-(d) shown, the proposed ESP can have better maneuverability and more potential to stabilize the vehicle. In the low friction coefficient condition, the timely intervention made the great help for the vehicle to stabilized vehicle. And it shows that the proposed ESP system not only solves the time delay problem, but also improves the control intervention time.

4. Conclusions

(1) In this paper, a novel reference model is proposed based on the preview mechanism of driver. In order to enhance the traditional reference model's driving intention, it used the prediction of steering wheel angle to compensate the driving intention. The proposed reference model is established by the fuzzy control method, which introduced the predicted steering wheel angle.

(2) Simulation results of the ESP system based on the proposed reference model show a more responsive and better performance than a traditional method. The proposed reference model and the new ESP control system are able to improve the vehicle stability under critical situations.

Acknowledgements

This work was supported by the Natural Science Foundation of the Jiangsu Higher Education Institutions of China [Grant Number 13KJB510005].

REFERENCES

[1]. *Van Zanten A T.* Bosch ESP systems: 5 years of experience. SAE Technical Paper, 2000.

[2]. *Geng C, Mostefai L, Denaï M, et al.* Direct yaw-moment control of an in-wheel-motored electric vehicle based on body slip angle fuzzy observer, J. Industrial Electronics, IEEE Transactions on, 2009, 56(5): 1411-1419.

[3]. *Jonasson M, Andreasson J, Solyom S, et al.,* Utilization of actuators to improve vehicle stability at the limit: From hydraulic brakes toward electric propulsion. Journal of Dynamic Systems, Measurement, and Control, 2011, 133(5): 051003.

[4]. *Raksincharoensak P, Mizushima T, Nagai M.* Direct yaw moment control system based on driver behavior recognition, J. Vehicle System Dynamics, 2008, 46(S1): 911-921.

[5]. *Xiong L, Yu Z, Wang Y, et al.,* Vehicle dynamics control of four in-wheel motor drive electric vehicle using gain scheduling based on tire cornering stiffness estimation. Vehicle System Dynamics, 2012, 50(6): 831-846.

[6]. *Zhu H, Li L, Jin M, et al.* Real-time yaw rate prediction based on a non-linear model and feedback compensation for vehicle dynamics control[J]. Proceedings of the Institution of Mechanical Engineers Part D-Journal of Automobile Engineering, 2013, 227(10): 1431-1445.

[7]. *Chen Y, Hedrick J K, Guo K,* A novel direct yaw moment controller for in-wheel motor electric vehicles[J]. Vehicle System Dynamics, 2013, 51(6): 925-942.
[8]. *Guo K, Guan H,* Modelling of driver/vehicle directional control system. Vehicle System Dynamics, 1993, 22(3-4): 141-184.
[9]. *Y.S. Tan, H. Shen, M.H. Huang,* Internal model control driver model with direction preview. China Science paper. 2014, 9(11): 1275-1278.
[10]. *Le T P, Stiharu I.* An optimal preview driver model applied to a non-linear vehicle and an impaired driver. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of automobile engineering, 2013, 227(4): 536-548.
[11]. *Li Y, Tong S, Li T.* Observer-based adaptive fuzzy tracking control of MIMO stochastic nonlinear systems with unknown control directions and unknown dead zones. IEEE Transactions on Fuzzy Systems, 2015, 23(4): 1228-1241.
[12]. *Zhang RY, Huang H, Chen WW, et al.,* Coordinated control of ESP Based on Function Allocation and Multi-objective Fuzzy Decision. Chinese Journal of Mechanical Engineering, 2014, 50(6): 99-106.
[13]. *Aksjonov A, Augsburg K, Vodovozov V.* Design and Simulation of the Robust ABS and ESP Fuzzy Logic Controller on the Complex Braking Maneuvers. Applied Sciences, 2016, 6(12): 382.
[14]. *Liu X, Xiong L, Yu ZP, et al.* Experiment Research on Control Algorithm of ESP System[C]//Applied Mechanics and Materials. Trans Tech Publications, 2013, 427: 838-841.
[15]. *Klomp M.* Longitudinal force distribution using quadratic ally constrained linear programming. Vehicle System Dynamics, 2011, 49(12): 1823-1836.
[16]. *Hsiao T.* Robust wheel torque control for traction/braking force tracking under combined longitudinal and lateral motion. IEEE Transactions on Intelligent Transportation Systems, 2015, 16(3): 1335-1347.
[17]. *Liu W, He H, Peng J.* Driving control research for longitudinal dynamics of electric vehicles with independently driven front and rear wheels. Mathematical Problems in Engineering, 2013(2013)