Front wheel angle control of steering by wire system based on fuzzy adaptive PID algorithm

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Abstract: - Steer by wire system can be free to design steering force transmission characteristics and angular displacement transfer characteristics, which can realize the coordination of portability, the driver steering road feel and handling stability. The dynamics models of steer by wire and vehicle system are established. Control strategy of front wheel angle based on fuzzy adaptive PID control is put forward, and simulation in different working conditions and road test are conducted. The simulation results show that front wheel angle control based on fuzzy adaptive PID can make the system achieve good control stability in different conditions and different road test. The results of this study can provide certain theoretical basis for research and application of steer by wire system.

Key-Words: - vehicle; steering by wire; fuzzy adaptive PID; front wheel angle; control

1 Introduction

The mechanical connection between the steering wheel and front wheels of automotive steer-by-wire system (SBW) has been cancelled, which gets rid of the limitation of traditional steering system completely. Thus, it can design the force and displacement transmission characteristics freely, and resolve the contradiction between the portability and steering feel of the steering system [1]. In addition, the steer-by-wire system can realize active intervention control, which can improve the vehicle handling stability, and bring more development space for the design of steering characteristics [2].

The front steering angle of SBW system is mainly controlled by the steering angle motor, which can make it obtain more direct and accurate control [3-4]. Meanwhile, because of the influence of the vehicle structure on the transmission characteristics, the vehicle system has the hysteresis and nonlinearity in itself. When driving

in the complex environment, the vehicle is expected to not only have a real-time and accurate operation according to the intention of the driver, but also can automatically revert to the safe state in case of danger [5-6]. Therefore, it is necessary to control the front steering angle reasonably.

At present, research on the control of the front steering angle mainly focus on improving the vehicle handling stability [7-12]. However, they neither have a comprehensive analysis of the control effects under different road test, nor consider the driver factors in the whole closed loop system. Therefore, it is unable to analyze the influence of the SBW system on the vehicle dynamic characteristics comprehensively.

In this paper, the dynamic model of the vehicle, driver model and SBW system, which include the steering wheel subsystem model and the front wheel system subsystem model, are established. Considering the yaw rate gain and sideslip angle gain, the control strategy of the front wheel angle based on fuzzy adaptive PID is conducted, and the simulation analysis in different working

conditions and road test are carried out.

2 Dynamic model

2.1. Dynamic model of the vehicle

Assuming that the vehicle has only two DOF of the lateral and yaw motions, and the actual vehicle model is simplified to a two wheels model with only the front and rear wheels, then the two DOF vehicle model is obtained shown in figure 1.
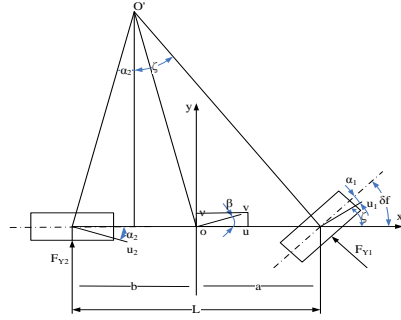
Figure 1 Two DOF vehicle model

The two DOF vehicle differential equations can be expressed as:

$$\begin{cases} (k_1 + k_2)\beta + \dfrac{1}{u}(ak_1 - bk_2)\omega_r - k_1\delta_f = m(\dot{v} + u\omega_r) \\ (ak_1 - bk_2)\beta + \dfrac{1}{u}(a^2k_1 + b^2k_2)\omega_r - ak_1\delta_f = I_z\dot{\omega}_r \end{cases} \quad (1)$$

where m is the vehicle total mass, kg; I_Z is vehicle moment of inertia around Z axis, kg·m^2; α_1, α_2 are the front and rear wheel side slip angle respectively, rad; k_1、k_2 are the front and rear tire cornering stiffness respectively, N·rad^{-1}; δ_f is the front wheel angle of the vehicle, rad; a、b are the distance of front and rear axle to center of mass respectively, m; u is the velocity of the vehicle along the x axis, that is, the forward speed of vehicle, m·s^{-1}; v is the lateral speed of vehicle, m·s^{-1}; ω_r is the yaw rate at steering, rad /s; β is sideslip angle, rad.

2.2 Steering wheel subsystem model

The direct connection preserved by the traditional vehicle between the steering system and the front wheels has been disconnected. Thus, the steering wheel subsystem model can be independent relatively, compared with the previous model. The simplified model of the steering wheel subsystem is shown in Figure 2.
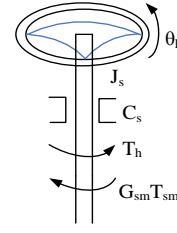
Figure 2 Steering wheel model

Then, the differential equations of steering wheel subsystem can be given as:

$$J_s\ddot{\theta}_h + C_s\dot{\theta}_h = T_h - T_{sm}G_{sm} \quad (2)$$

where J_s is the moment of inertia that the steering wheel subsystem equivalent to steering column kg·m^2; θ_h is the steering wheel angle, rad; C_s is the damping coefficient that the steering wheel subsystem equivalent to steering column, N·m/(rad/s); T_h is the steering wheel torque, N·m; T_{sm} is the force motor torque, N·m; G_{sm} is the transmission ratio of the force motor reducer.

2.3 Front wheel subsystem model

Ignoring the stiffness coefficient of the parts, and taking the moment of inertia and damping coefficient of the steering motor shaft equivalent to the steering wheel shaft, then the simplified front wheel subsystem model can be obtained as shown in figure 3.
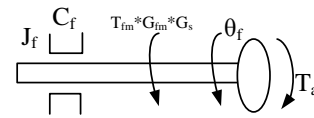
Figure 3 Front wheel system model

Based on figure 3，the differential equations of front wheel subsystem can be depicted as:

$$J_f\ddot{\theta}_f + C_f\dot{\theta}_f = T_{fm}G_{fm}G_s - T_a \quad (3)$$

where J_f is the moment of inertia that the steering wheel subsystem equivalent to the front wheel steering pin, kg·m^2; C_f is the damping coefficient that the steering wheel subsystem equivalent to the front wheel steering pin, N·m/(rad/s); T_{fm} is the steering motor torque of the front wheel, N·m; T_a is the front wheel aligning torque, N·m; θ_f is the front wheel angle, rad; G_s is the transmission ratio of the rack and pinion redirector; G_{fm} is the transmission ratio of the front wheel steering

motor reducer.

2.4 Driver model

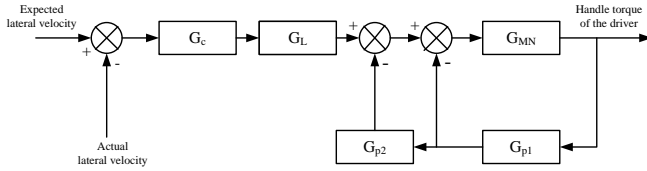The driver model which describes the manipulation behavior of the driver is used [13], as shown in figure 4.

Figure 4   Driver model

The driver model can be depicted as follows:

$$G_{NM} = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \qquad (4)$$

$$G_{p1} = \frac{K_1 s}{s + 1/T_1}, \quad G_{p2} = \frac{K_2}{s + 1/T_2} \qquad (5)$$

$$G_L = \frac{1}{T_3 s + 1}, \quad G_c = \frac{1 + mT_c s}{T_c s + 1} \qquad (6)$$

where G_{NM} is the transfer function of the driver's arm neuromuscular system; G_{P1} and G_{P2} is the feedback transfer function of the driver body and muscle movement respectively; G_L is the delay transfer function of the driver's brain information process; G_C is the lead compensation transfer function of the driving behavior in the low frequency.

3 Control strategy of the front wheel angle

3.1 Control structure

Figure 5 shows the block diagram of the front wheel steering angle control. The input of the driver model is the difference between the expected and actual lateral velocity, and the output is the steering wheel torque. T_m is the torque of feedback motor; $K_c(v)$ is the aligning torque coefficient related to the speed; K_T is the

feedback torque gain; δ_h is the steering wheel angle; δ^* is the expected front wheel steering angle; δ is the actual front wheel steering angle.

The difference between the expected and actual front wheel steering angle is input to the fuzzy controller, and the modified one is obtained. Next, it is output to the two DOF vehicle model to get the actual lateral velocity, which is then feedback to the driver model, so as to realize the closed-loop control of the driver-vehicle system.
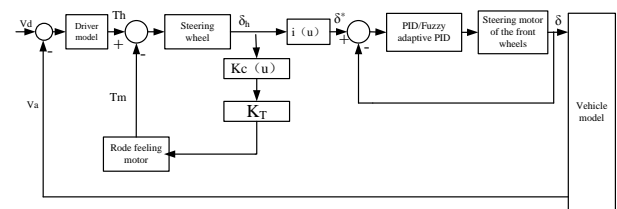

Figure 5 Front wheel steering angle control block diagram

3.2 Control method

In this paper, the fuzzy adaptive PID control method is adopted, whose control law is based on the PID control. Its general form can be expressed as:

$$u(t) = K_P e(t) + K_I \int_0^t e(t)dt + K_D \frac{de(t)}{dt} \qquad (7)$$

In the actual control process, it is often used the summation and difference quotient instead of integral and differential respectively, and the discretization equation can be given as:

$$u(k) = K_P e(k) + K_I \sum_{i=0}^{k-1} e(i) + K_D[e(k) - e(k-1)] \qquad (8)$$

Figure 6 shows the designed two-dimensional fuzzy control structure with double input and three outputs. The input of the fuzzy controller is the deviation e and deviation change rate ec between the expected and the actual front wheel angle. The output is the deviation of the proportional, integral and differential coefficient respectively, which can be expressed as Δkp ,

Δki , Δkd .

In the steering process, by constantly testing e and ec, the fuzzy controller can adjust the three parameters of the proportion, differential and integral coefficient kp , kp , kp , to meet the different requirements of e and ec. It can make the vehicle has a good response, and achieve the vehicle steering stability.
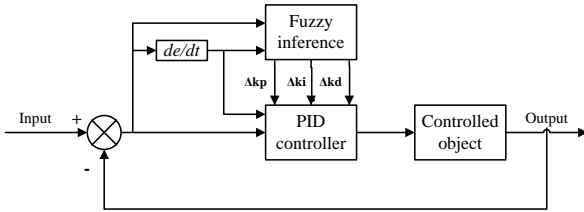

Figure 6 The structure of fuzzy adaptive PID controller

The same membership functions are used for the input and output fuzzy sets, with the beginning, middle and end part as the one of Z, triangle and S type respectively [14–15].

Considering the influence of the deviation change rate ec, the fuzzy control rules of the three parameters are established as shown in table 1 to table 3.

Table 1 Δkp fuzzy control rule

ec \ e	NB	NM	NS	ZO	PS	PM	PB
NB	PB	PB	PM	PM	PS	ZO	ZO
NM	PB	PB	PM	PS	PS	ZO	NS
NS	PM	PM	PM	PS	ZO	NS	NS
ZO	PM	PM	PS	ZO	NS	NM	NM
PS	PS	PS	ZO	NS	NS	NM	NM
PM	PS	ZO	NS	NM	NM	NM	NB
PB	ZO	ZO	NM	NM	NM	NB	NB

Table 2 Δki fuzzy control rule

ec \ e	NB	NM	NS	ZO	PS	PM	PB
B	NB	NB	NM	NM	NS	ZO	ZO
NM	NB	NB	NM	NS	NS	ZO	ZO
NS	NB	NM	PS	NS	ZO	PS	PS
ZO	NM	NM	NS	ZO	PS	PM	PM
PS	NM	NS	ZO	PS	PS	PM	PB
PM	ZO	ZO	PS	PS	PM	PB	PB
PB	ZO	ZO	PS	PM	PM	PB	PB

Table 3 Δkd fuzzy control rule

ec \ e	NB	NM	NS	ZO	PS	PM	PB
NB	PS	NS	NB	NB	NB	NM	PS
NM	PS	NS	NB	NM	NM	NS	ZO
NS	ZO	NS	NM	NM	NS	NS	ZO
ZO	ZO	NS	NS	NS	NS	NS	ZO
PS	ZO	ZO	ZO	ZO	ZO	ZO	ZO
PM	PM	NS	PS	PS	PS	PS	PB
PB	PB	PM	PM	PM	PS	PS	PB

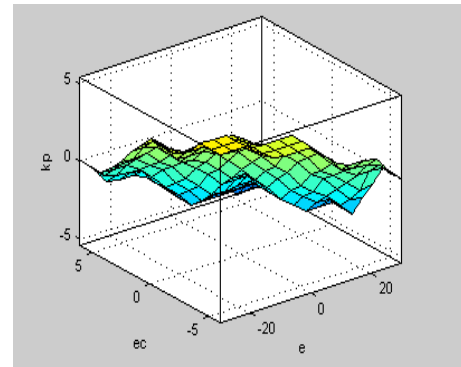The fuzzy control rule surfaces of the three parameters for the controller are shown in figure 7 to figure 9.

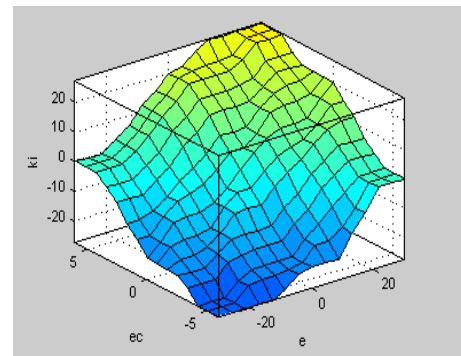Figure 7 Δkp fuzzy control rule surface
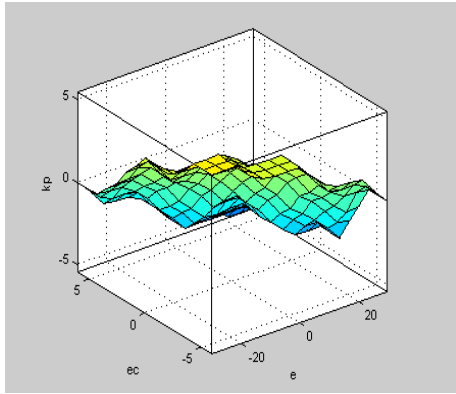

Figure 8 Δki fuzzy control rule surface

Figure 9 Δkd fuzzy control rule surface

3.3 Simulation analysis

The simulation model of the controller and the vehicle are established with Matlab/Simulink software, which are shown in figure 10 and figure 11. The simulations are carried out with the PID control and fuzzy adaptive PID control respectively. Table 4 shows the main simulation parameters.
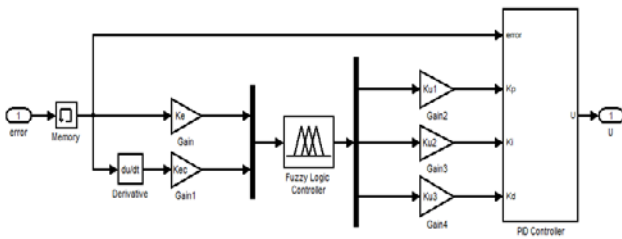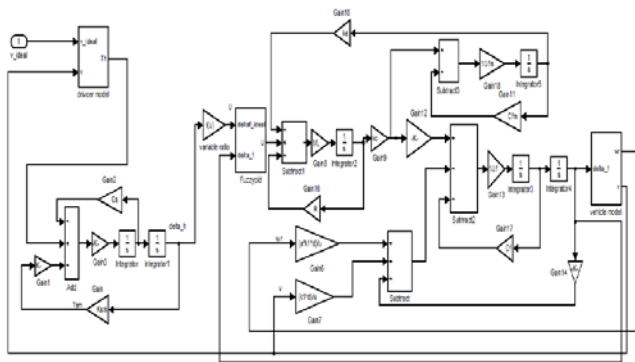

Figure 10 Simulation model of the controller


Figure 11 Simulation model of the vehicle

Table 4 Main simulation parameters

Name	Value	Name	Value
m/kg	1818.2	J_S / (kg·m^2)	0.04
I_Z/ (kg/m$^{2)}$	3885	J_f/ (kg·m^2)	0.0098
a/ m	1.463	C_S/(N·m/(rad/s))	0.225
b/ m	1.585	C_f/ (N·m/(rad/s))	0.03

k_1/ (N/rad)	-62618	G_s	20
k_2/ N/rad	-110185	G_{fm}	25
ξ	0.459	T_1(s)	2.5
w_n	9.2	T_2(s)	0.225
K_1	0.088	T_3(s)	0.3
K_2	1.6	T_c(s)	0.0109
α	2.3		

Setting the initial speed of the vehicle as 30km/h, 60km/h and 90km/h, the simulation analyses are conducted in the steering transient response test, double lane change test and serpentine road test.

It can be seen from figure 12 to figure 14 that the yaw rate and the sideslip angle are all increased with the increase of the speed, and the maximum values are all in a reasonable range, in the steering transient response test, double lane change test and serpentine road test. Thus, they meet the requirement of the vehicle handling stability. Moreover, compared with the traditional PID control, the response time of the system is improved in a certain extent, and the steady-state or peak value of the yaw rate and the sideslip angle are all reduced, with fuzzy adaptive PID control. Therefore, the system has better control effect with fuzzy adaptive PID control. It can further improve the vehicle driving stability, and reduce the driver's manipulating burden.
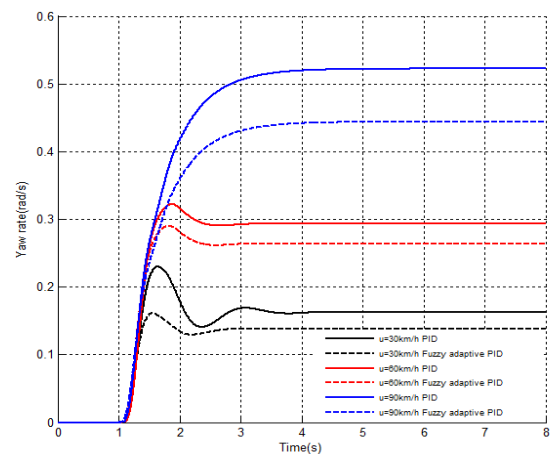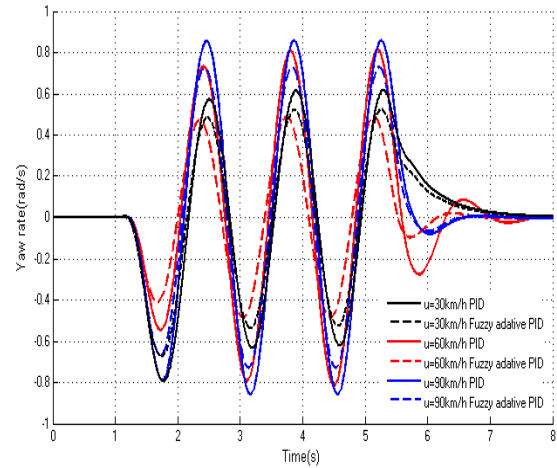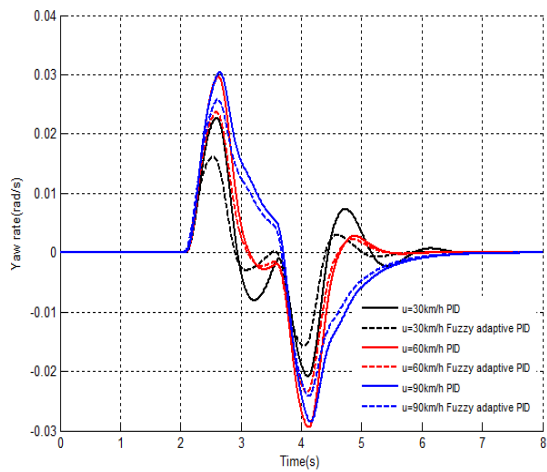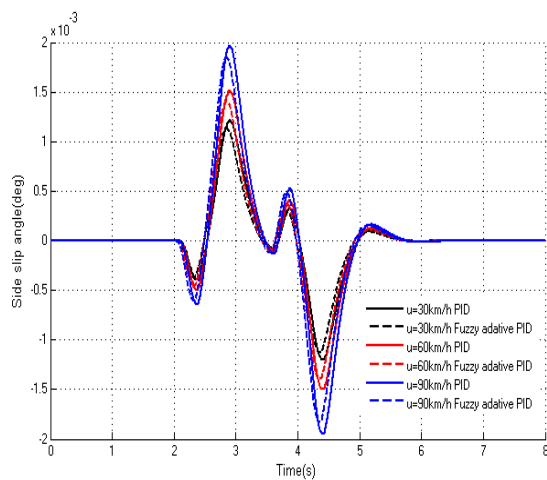

(a) Yaw rate
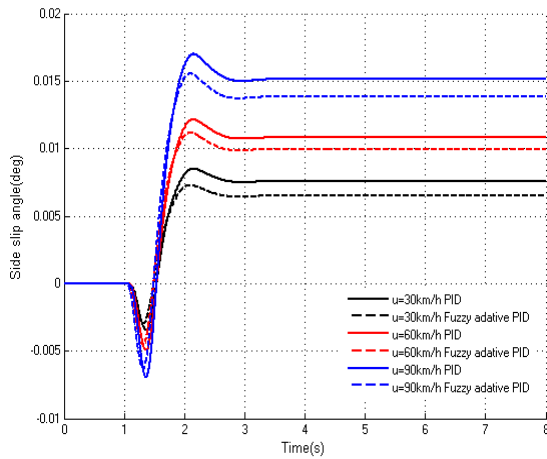
(b) Sideslip angle

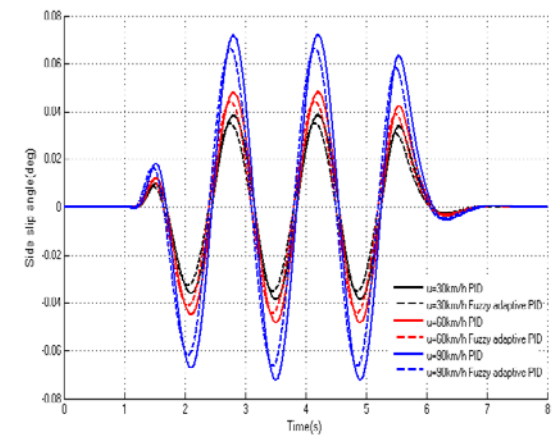Figure 12 Transient response test

(a) Yaw rate

(b) Sideslip angle

Figure 13 Double lane change test

(a) Yaw rate

(b) Sideslip angle

Figure 14 Serpentine road test

4 Conclusions

In this paper, the vehicle dynamic model, driver model and SBW system are established. Then, the control strategy of the front wheel angle based on fuzzy adaptive PID is studied, and the simulation analysis in different working conditions and road test are conducted. The simulation results show that the control strategy based on fuzzy adaptive PID control can ensure the vehicle has good handling stability under different conditions. The simulation results show that the control strategy based on fuzzy adaptive PID control can ensure the vehicle has good handling stability under different conditions. The results of this paper can provide certain theoretical basis for the research and application of the SBW system.

References:

[1] P. Yih, J. Ryu, J. C. Gerdes. Modification of vehicle handling characteristics via steer-by-wire. *IEEE transactions on control systems technology: A publication of the IEEE Control Systems Society,* Vol.13, No.6, 2005, pp. 965-976.

[2] N. Daher, M. Ivantysynova. A virtual yaw rate sensor for articulated vehicles featuring novel electro-hydraulic steer-by-wire technology. *Control Engineering Practice*, No.30, 2014, pp.45-54.

[3] A. E. Cetin, M. A. Adli, D. E. Barkana, et al. Adaptive on-line parameter identification of a steer-by-wire system. *Mechatronics*, Vol.22, No. 2, 2012, pp.152-166.

[4] X. Fu, F. Li. Research of automotive steer-by-wire control based on integral partition PID control. *Genetic and Evolutionary Computing. 3rd International Conference*, 2009, pp.561-564.

[5] B. Zheng, S. Anwar. Yaw stability control of a steer-by-wire equipped vehicle via active front wheel steering. *Mechatronics*, Vol.19, No.6, 2009, pp.799-804.

[6] M. Segawa, S. Nakano, O. Nishihara, et al. Vehicle stability control strategy for steer by wire system. *JSAE Review*, 2001, Vol.22, No. 4, pp.383-388.

[7] S. W. Oh, S. C. Yun, H. C. Chae, et al. The development of an advanced control method for the steer-by-wire system to improve the vehicle maneuverability and stability. *SAE Paper*, No.2003-01-0578

[8] V. Tavoosi, R. Kazemi, S. M. Hosseini. Vehicle handling improvement with steer-by-wire system using hardware in the loop method. *Journal of Applied Research and Technology*, Vol.12, No. 4, 2014, pp.769-781.

[9] Y. Jiangyun, K. Feng, W. Fangyuan. Control for front wheel angle of steer-by-wire based on intelligent variable steering ratio. *Computer Measurement & Control*, No.8, 2012, pp.20-30.

[10] Y. R. Li, J. M. Huang, Z. M. He. A nonlinear control method of steering angle following used in steer-by-wire system. *Applied Mechanics and Materials*, No.190, 2012, pp.890-893.

[11] S. W. Oh, H. C. Chae, S. C. Yun, et al. The design of a controller for the steer-by-wire system. *JSME International Journal Series C*, Vol.47, No.3, 2004, pp.896-907.

[12] H. Y. Zheng, C. F. Zong, C. W. Tang, et al. Control algorithm for steer-by-wire system with ideal steering ratio. *Journal of Jilin University (Engineering and Technology Edition)*, Vol.37, No.6, 2007, pp.1229-1235.

[13] A. Modjtahedzadeh, R. A. Hess. Model of driver steering control behavior for use in assessing vehicle-handling qualities. *Transactions of ASME*, Vol.115, No.3, 1993, pp.456-464.

[14] W. S. L Huashi. Yaw moment fuzzy control of four-wheel-steering vehicle based on co-simulation technology. *Transactions of the Chinese Society for Agricultural Machinery*, No.5, 2011, pp.2-9.

[15] D. Xiang, J. Yuan, W. Xu. Study on fuzzy PID algorithm for a new active front steering system[J]. *Journal of Control Engineering and Technology*, Vol.2, No.1, 2012, pp.24-29.