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Integrated Chassis Control Using ANFIS

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***Abstract* - A nonlinear control system that integrated active front steering system and vehicle dynamics control system based on a multi-body vehicle dynamic model is introduced in this paper. The multibody vehicle dynamic model based on ADAMS can accurately predict the dynamics performance of the vehicle. The control strategy consists of two control objectives (yaw velocity ω and sideslip angle β). An integrated controller is built according to the lateral acceleration to distribute the desired yaw moment to VDC and AFS. The VDC and AFS controllers are established using ANFIS (Adaptive-Network-based Fuzzy Inference System) and Fractional-order $PI^{\lambda}D^{\mu}$ control algorithm, respectively. The adaptive fuzzy controller can adjust the fuzzy control rules and membership functions through by training of ANFIS, and the parameters of fractional-order $PI^{\lambda}D^{\mu}$ are tuned using genetic algorithm (GA) optimization method. The co-simulation scenario takes place on an icy road at high speed with a step steer input. The simulation results show that integrated chassis control system can enhance vehicle handling stability and safety greatly.**

Index Terms* - *List key index terms here. No mare than 5.

I. INTRODUCTION

With reference to the automobile as an indispensable element of our personal mobility, the desire for more comfort and security have placed their mark in all areas of vehicle design. The vehicle chassis control is in general to control vehicle lateral, vertical and longitudinal motions in order to improve handling performance, ride comfort and traction/braking performance. To enhance such performance, it has basically been depending on the steering control, suspension control and traction/braking control.

During recent years, some electronically controlled systems such as four-wheel steering system (4WS), anti-lock braking system (ABS), active front steering system (AFS), vehicle dynamics control system (VDC), active suspension system (AS), traction control system and others, have been developed and significant betterment of vehicle performance have been demonstrated by them. However, The effect of chassis control can be further increased by the integration and coordination of those controls based upon a deeper observation and study of vehicle dynamics and tire characteristics. Fig. 1 shows the structure of integrated chassis control.

In Peter E. Rieth's paper [1], ESC II (ESC with active steering intervention) is presented, which considers the integration of brakes and steering. In [2], cooperation of 4WS and direct yaw moment control (DYC) has been considered, In which a linear 4WS controller designed independently of the DYC controller was used. Yu and Moskwa [3] proposed an integrated control system designed by using feedback linearization technique and sliding mode control theory. Mauro Velardocchia and Andrea Morgando [4] presented a VDC-4WS-Active Roll Control (ARC) integration based on one reference body yaw rate for all active systems. Yuqing Wang and Masao Nagai [5] introduced an integrated control system providing high performance within tires' strong nonlinear areas with an adaption to the changing road and other conditions, by optimally controlling the front and rear steering angles and the yaw moment, based on the information of system parameters identification. In Jiang Wei and Yu Zhuoping's paper [6], a chassis control system integrated AFS and DYC was presented using fuzzy logic controller to distribute the required yaw moment to AFS and DYC control system properly. Kazuya Kitajima and Huei Peng [7] designed two integration algorithms for vehicle chassis control systems—a feedforward integration method and an H∞ control algorithm aiming to coordinate VDC, 4WS and active suspension functions of ground vehicle. In G. Burgio and P. Zegelaar's publication [8], the feedback linearization technique was proposed for the design of the integrated vehicle controller, with steering (AFS, SBW) and brakes (or equivalently traction) actuators. E. M. Elbeheiry, Y. F. Zeyada and M. E. Elaraby [9] suggested the integration between Active Front Steering (AFS) and Active Roll Moment Control (ARMC) systems in order to enhance the vehicle controllability and the AFS system adopted a robust sliding mode controller (SMC). In E. Ono and Y. Hattori's article [10], the vehicle dynamics integrated control for four-wheel-distributed steering and four-wheel-distributed traction/braking systems based on friction circle of each wheel was proposed.

This paper focuses on the integrated chassis control of VDC and AFS, which consists of five sections. The second section presents the multibody vehicle dynamic model considering both AFS and VDC control system. In the third section, the integrated controller is designed to distribute the yaw moment to VDC and AFS. The VDC and AFS system are controlled by ANFIS and Fractional-order $PI^{\lambda}D^{\mu}$ method, respectively. The simulation results and conclusion are presented in the last two sections.

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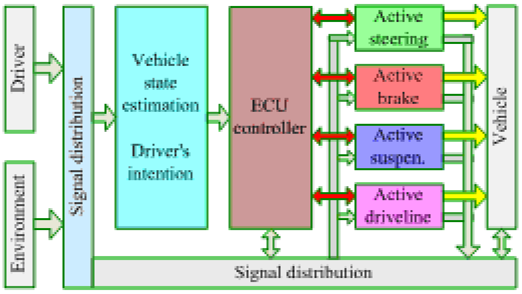

Fig. 1 Integrated chassis control

II. VEHICLE SYSTEM MODEL

In this section first, a linear 2-DOF model is described, which is proposed as and ideal reference model and then, a multi-body dynamic model which is considered for simulation is presented.

A. *Linear 2-DOF reference model*

We consider the multibody dynamic model as a whole-vehicle model and the 2-DOF model (Fig. 2) as an ideal reference model. This 2-DOF model represents the driver's desired vehicle performance and driving tracking. If the driver's desired vehicle performance and driving tracking are expressed as linear functions, the vehicle will be much easier and safer to drive. In Fig. 2, G denotes the ground reference, and the vehicle has a front-wheel steering system.

The vehicle dynamic equations for the ideal 2-DOF reference model are formulated as

$$\begin{cases} m(\dot{v}_y + u_c\omega) = -\dfrac{C_{af}+C_{ar}}{u_c}v_y - \dfrac{aC_{af}-bC_{ar}}{u_c}\omega + C_{af}\delta_f \\ I\dot{\omega} = -\dfrac{aC_{af}-bC_{ar}}{u_c}v_y - \dfrac{a^2C_{af}+b^2C_{ar}}{u_c}\omega + aC_{ar}\delta_f \end{cases} \tag{1}$$

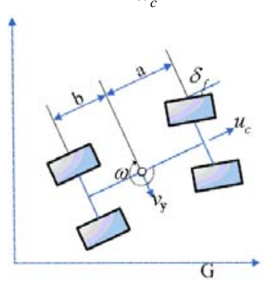

Fig. 2 A 2-DOF ideal vehicle model

where m denotes the mass of the vehicle; v_y is the lateral speed; ω represents the yaw velocity; u_c is vehicle forward velocity; C_{af} and C_{ar} represent the front- and rear-wheel cornering stiffness, respectively; δ_f is the front steering angle; I is the moment of inertia; and a and b are the length from the front and rear axles, respectively, to the center of gravity of the vehicle.

The parameters of the ideal 2-DOF vehicle are given as follows: m=1200Kg , I=2549kg•m^2 , a=1.256m^2 , b=1.368m^2 , C_{af}=75kN/rad and C_{ar}=100.4kN/rad .

B. ACTIVE FRONT STEERING SYSTEM OVERVIEW

Fig. 3 shows the principle of the active front steering system. The driver controls the vehicle via the hand steering wheel (the steering wheel angle is denoted by δ_s) and the actuator provides an additional steering wheel angle δ_a according to the signal from ECU. Both angles result in a pinion angle down at the steering track..

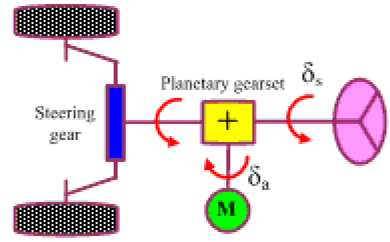

Fig. 3 Principle of the active front steering system

C. VEHICLE DYNAMICS CONTROL SYSTEM OVERVIEW

VDC can control the longitudinal and lateral stability synthetically to unify the ABS, TCS, and direct yaw moment control (DYC). Different companies have different names for their systems: Bosch calls it the electronic stability program (ESP) [11], Toyota calls it vehicle stability control (VSC) [12], and others call it interactive vehicle dynamics (IVD).

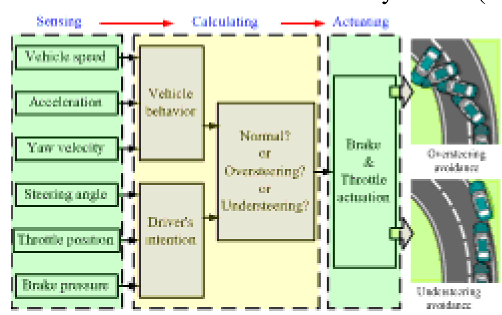

Fig. 4 Principle of the vehicle dynamic control system

Fig. 4 shows the principle of vehicle dynamics control system. Vehicle dynamics control is realized mainly by the differential controls on the brake forces on four wheels, which provides yaw moment to keep the vehicle running according to the driver's intent. In addition, it can prevent sharp turning when the driver operates too heavily. Vehicle dynamics control uses brake moment and engine moment to maintain stability and make it easy for the driver to steer the vehicle.

D. MULTIBODY VEHICLE DYNAMIC MODEL

Multibody simulation has been widely used to predict vehicle handling, safety, stability and performance, and the loads calculated from these simulations are used for subsequent stress and durability analysis. The actual multibody solution process involves numerically integrating the equations of motion. The mathematical algorithm used to derive the equations of motion was developed by Prof. Ed Haug and his team at the University of Iowa in the early 1980's [13]. It is based on using a maximal set of Cartesian "generalized" coordinates (X, Y, Z) and "Euler Parameters" (e0, e1, e2, e3) to form a system of differential-algebraic equations.

In this paper, the multibody vehicle dynamic model is built in the ADAMS/CAR environment. This model includes seven subsystems: the front suspension system, the rear suspension system, the brake system, the powertrain system, the steering system, the tire and the bodywork system, as shown in Fig. 5.

Fig. 5 Multibody vehicle dynamic model

E. Tire Model

In most time, the integrated chassis system has strong nonlinearity and we should adopt nonlinear tire model. Thus, it introduces Pacejka's Magic Formular Model [14] which has high precision for longitudinal force of wheel and side-force and also has better confidence level out of range of limit value. The Pacejka's Magic Formular Tyre Model can be described as follows:

$$y = D\sin(C\tan^{-1}\{Bx - E[Bx - \tan^{-1}(Bx)]\}) \qquad (2)$$

with

$$Y(X) = y(x) + S_V\,,\; x = X + S_H$$

Where Y represents the output variable (longitudinal force F_x or lateral-force F_y or self-aligning torque M_z), X is the input variable (sideslip angle β or longitudinal slip κ). The coefficients B, C, D, E are the stiffness factor, the shape factor, the peak value and the curvature factor, respectively; S_H, S_V denotes the horizontal and vertical shift, respectively. Here B, C, D, E, S_H and S_V are known as primary Magic Formular parameters.

III. Integrated VDC and AFS Control System

A. Integrated Chassis Control Structure

The simulation system is established by the combination of ADAMS and MATLAB. The structure of co-simulation is showed in Fig. 6. AFS can generate corrective large yaw moment in each direction when vehicle is driven straight; when the steer angle is large, AFS is more effective in controlling oversteer than understeer; when vehicle is at the limit of adhesion, the yaw moment generate by AFS is very weak and the VDC should be the main actor. In this paper, the yaw moment is distributed to AFS and VDC according to the value of lateral acceleration which is a main factor to judge the state of the vehicle.

The ANFIS and fractional-order $PI^{\lambda}D^{\mu}$ methods are applied on the control of VDC and AFS, respectively. First, the state of VDC and AFS controllers (on or off) are determined by the distributing controller, and $\Delta\beta$ and $\Delta\omega$ (sideslip angle discrepancy and yaw velocity discrepancy) can be obtained by comparing the real values of the sideslip angle and yaw velocity with the desired values. Then, we can calculate the corrective yaw moment for both VDC and AFS. In addition, the braking force on each wheel, the throttle opening and the additional front steering angle (δ_a) can be identified by the corrective yaw moment.

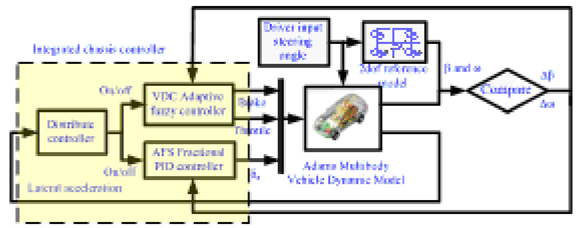

Fig. 6 The structure of integrated VDC and AFS control

The vehicle dynamic equations for the ideal 2-DOF reference

B. Adaptive Fuzzy Controller for VDC

The ANFIS is an attractive compromise between the adaptability of a neural network and interpretability of a fuzzy inference system. Fuzzy inference system doesn't maintain self-learning function, which limits its application; the artificial neural network liking a black box cannot express the linguistic fuzzy information and inference function. The ANFIS proposed by Jyh-Shing Roger Jang [15] can compensate the deficiency of fuzzy inference system by adopting the self-learning capability of neural network.

For the VDC controller, the Takagi-Sugeno (TS) fuzzy controller trained by ANFIS architecture is designed. When $-0.5 < \Delta\omega < 0.5$, it is in a stable state and different wheels can be controlled in the ranges $\Delta\omega < -0.5$ and $\Delta\omega > 0.5$. Two adaptive fuzzy controllers are built with two same input variables (yaw velocity discrepancy and sideslip angle discrepancy) and one output variable (Brake or Throttle). The structure of the adaptive fuzzy controller is showed in Fig. 7 and Fig. 8.

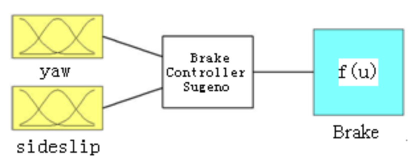

Fig. 7 Adaptive fuzzy controller to obtain brake force

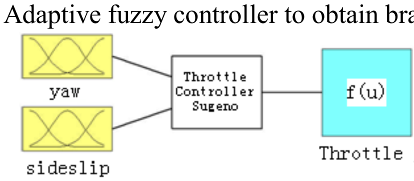

Fig. 8 Adaptive fuzzy controller to obtain throttle opening

Fig. 9 shows the general structure of ANFIS. The two input variables can be divided into 5 levels: negative big (NB), negative median (NM), zero (Z0), positive median (PM), and positive big (PB). The output variable (Brake or Throttle) is the linear function of the two input variables. The rule-base contains 25 fuzzy IF-THEN rules of TS type as follows:

Rule i : If X is A_i and Y is B_i , then $f_i = a_i\omega + b_i\beta + c_i$

Where X , Y and f are linguistic variables representing two inputs variables (sideslip angle discrepancy and yaw velocity discrepancy) and one output variable (Brake or Throttle), respectively. A_i and B_i are particular fuzzy subsets defined by nonlinear coefficient, namely premise parameters, while a_i , b_i and c_i are linear coefficients determining the output of each applied fuzzy rule and usually known as consequent parameters.

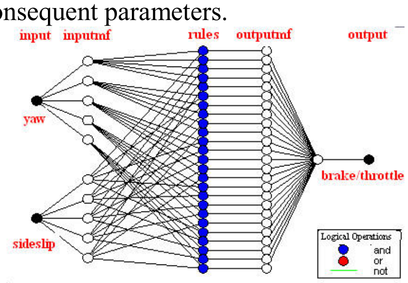

Fig. 9 The general structure of ANFIS

The training process of ANFIS is presented in Fig. 10. The training data set of sideslip angle discrepancy, yaw velocity discrepancy, brake force and throttle opening is obtained through the simulation based on multibody vehicle dynamic model built in ADAMS/CAR. These data are collected from various experiments at different situations like single lane change test, ISO lane change test, Brake in turn test, Constant-radius cornering test, Power-off cornering test, Fish-hook test, Step steer test, Swept sine steer test and so on. ANFIS takes the initial fuzzy model and tunes it by means of a hybrid technique combing gradient descent back-propagation and mean least-squares optimization algorithms. Fig. 11-14 shows the membership function after training and MF is the representation of membership function.

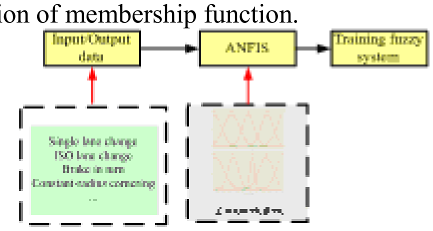

Fig. 10 The process of training fuzzy logic controller by ANFIS

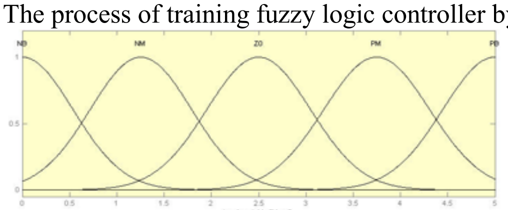

Fig. 11 MF of yaw velocity for BrakeController when $\Delta\omega > 0.5$

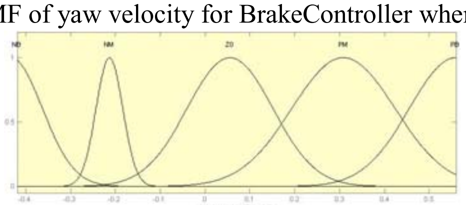

Fig. 12 MF of sideslip angle for BrakeController when $\Delta\omega > 0.5$

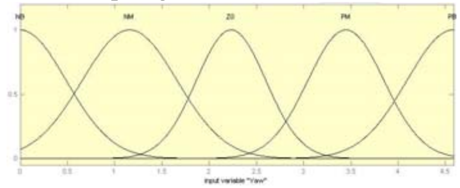

Fig. 13 MF of yaw velocity for ThrottleController when $\Delta\omega > 0.5$

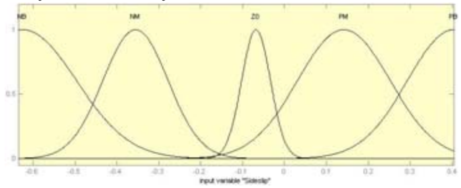

Fig. 14 MF of sideslip angle for ThrottleController when $\Delta\omega > 0.5$

C. Fractional-order $PI^{\lambda}D^{\mu}$ controller for AFS

VDC can control the longitudinal and lateral stability synthetically to unify the ABS, TCS, and direct yaw moment control (DYC). Different companies have different names for their systems: Bosch calls it the electronic stability program (ESP) [11], Toyota calls it vehicle stability control (VSC) [12], and others call it interactive vehicle dynamics (IVD).

1) Fractional-order $PI^{\lambda}D^{\mu}$ overview

In 1994, Podlubny [16] proposed a generalization of the PID controller, which is called the fractional-order $PI^{\lambda}D^{\mu}$ controller because of involving an integrator of λ and differentiator of order μ , and which also shows better performance than the classical PID controller. Fractional-order $PI^{\lambda}D^{\mu}$ controllers are described by fractional-order differential equations.

Fractional-order $PI^{\lambda}D^{\mu}$ control is the generalization and development of the integer-order PID control. It is described as follows [16]:

$$u(t) = k_p e(t) + k_i D_t^{-\lambda} e(t) + k_d D_t^{\mu} e(t) \qquad (3)$$

where $e(t)$ is the system error, λ is the fractional integral order, μ is the fractional derivative order. When $\lambda = 1$, and $\mu = 1$, classical integer order PID controller is obtained. $\lambda = 1$ and $\mu = 0$ define a PI controller; $\lambda = 0$ and $\mu = 1$ give a PD controller. $\lambda = 0$ and $\mu = 0$ give a gain.

2) Fractional-order PI^{λ} controller design and optimization

A PI^{λ} controller with yaw velocity discrepancy as input variable is designed to generate additional front steering wheel angle. The additional front steering wheel angle δ_a is described as follows:

$$\delta_a = K_{pFra} e_{\omega}(t) + K_{iFra} D_t^{-\lambda} e_{\omega}(t) \qquad (4)$$

where $e_\omega(t)$ is the yaw velocity discrepancy between the multi-body vehicle dynamic model and the 2-DOF ideal reference model. Parameters like K_{pFra}, K_{iFra}, and fractional integral order λ are tuned by GA optimization algorithm.

We can consider to build the PI^λ controller from the view of optimization.

The objective function selected to be optimized is: (1) sideslip angle discrepancy; (2) yaw velocity discrepancy.

To find: Design variables

to minimize: $F_t = \sqrt{\Psi}$

Where $\Psi = \rho_1\left(\frac{\overline{e}_\omega}{B_{\overline{e}_\omega}}\right)^2 + \rho_2\left(\frac{\overline{e}_\beta}{B_{\overline{e}_\beta}}\right)^2 + \rho_3\left(\frac{e_{\omega\max}}{B_{e_{\omega\max}}}\right)^2 + \rho_4\left(\frac{e_{\omega\min}}{B_{e_{\omega\min}}}\right)^2 + \rho_5\left(\frac{e_{\beta\max}}{B_{e_{\beta\max}}}\right)^2 + \rho_6\left(\frac{e_{\beta\min}}{B_{e_{\beta\min}}}\right)^2$

subject to:

$$3\sqrt{\frac{\sum_{i=1}^{N}(a_z)^2}{N}} + 2\max(a_z) \le 1m/s^2$$

$$a_y \le 7m/s^2$$

$$\phi \le 3\deg$$

where $B_{\overline{e}_\omega}$, $B_{\overline{e}_\beta}$, $B_{e_{\omega\max}}$, $B_{e_{\omega\min}}$, $B_{e_{\beta\max}}$ and $B_{e_{\beta\min}}$ are the maximum values of $\overline{e}_\omega$, $\overline{e}_\beta$, $e_{\omega\max}$, $e_{\omega\min}$, $e_{\beta\max}$ and $e_{\beta\min}$ respectively. $\rho_1 ... \rho_6$ are the corresponding weights; $\overline{e}_\beta$ is the mean value of difference between the real sideslip angle and set value; $e_{\beta\max}$ and $e_{\beta\min}$ are the maximum and minimum values, respectively, of the difference between the real sideslip angle and set value; $\overline{e}_\omega$ is the mean value of difference between the real and ideal yaw velocity; $e_{\omega\max}$ and $e_{\omega\min}$ are the maximum and minimum values, respectively, of the difference between the real and ideal yaw velocities; a_z is the body acceleration; a_y is the lateral acceleration; and ϕ represents the roll angle of bodywork.

The parameters for fractional-order PI^λ controller tuned by GA are given as follows:

$K_{pFra} = 0.378$, $K_{iFra} = 3.425$, $\lambda = 0.863$.

D. Integrated Chassis Control

The distributing controller is designed based on lateral acceleration in this section. There are two main control objectives for the integrated chassis control and they are the sideslip angle and yaw velocity control. The sideslip angle control strategy reduces the lateral motion and transportation of vehicle, while it improves handling maneuverability and reduces the delay of response of the vehicle; the yaw velocity control strategy minimizes the rotational motion of vehicle and leads the vehicle to lateral side tracking the desired trajectory.

According to the control authority in understeer and oversteer correction and the work simulation, some basic rules of integrated chassis control are obtained [6, 8, 9]:

(1) when lateral acceleration is small, such as running straight, the yaw moment demand should be implemented only through AFS to ensure the ride comfort performance.

(2) When lateral acceleration is medium, the yaw moment should be implemented through both AFS and VDC.

(3) When the lateral acceleration is large, like the vehicle is at the limit of adhesion, the yaw moment should be implemented only through VDC.

IV. Simulation Results and Discussion

Step steering experiment is adopted to validate the performance of integrated control of VDC and AFS. The vehicle velocity is 80km/h, and the driver input steering wheel angle is 30 degrees. Figs. 15 through 18 show the simulation results of integrated chassis control, only VDC control and without control with the same condition where the test is on an icy road (adhesion coefficient is 0.2). We can see that the yaw velocity and sideslip angle convergence rapidly, and the oscillation of the system reduces with the integrated control of VDC and AFS. The yaw velocity of integrated control system can also track the 2dof ideal reference model closely.

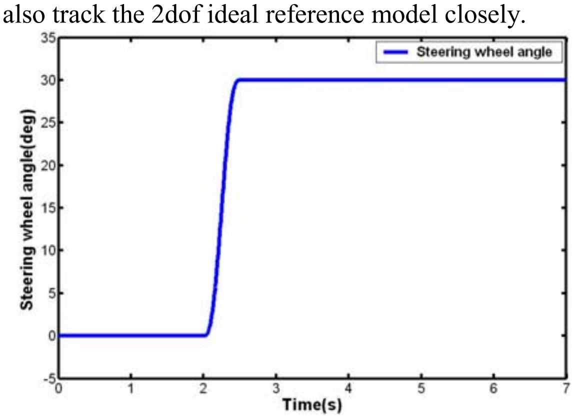

Fig. 15 Steering wheel angle

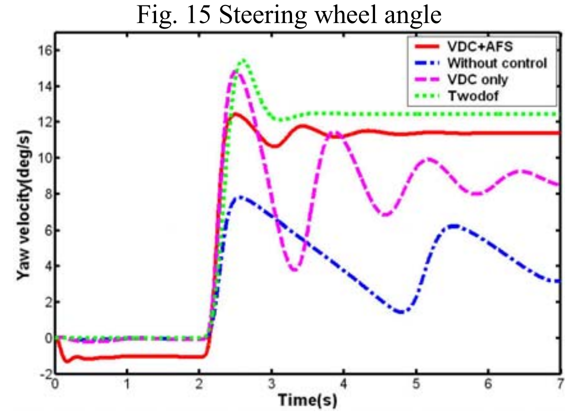

Fig. 16 Comparison of yaw velocity

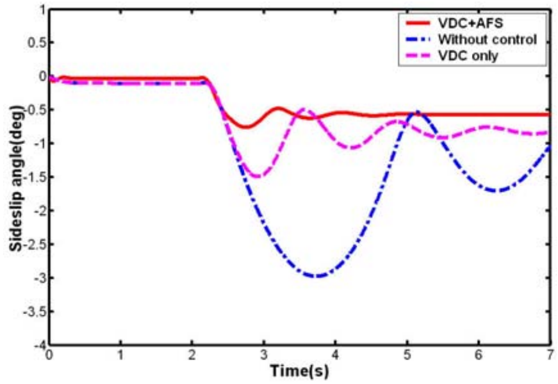

Fig. 17 Comparison of sideslip angle

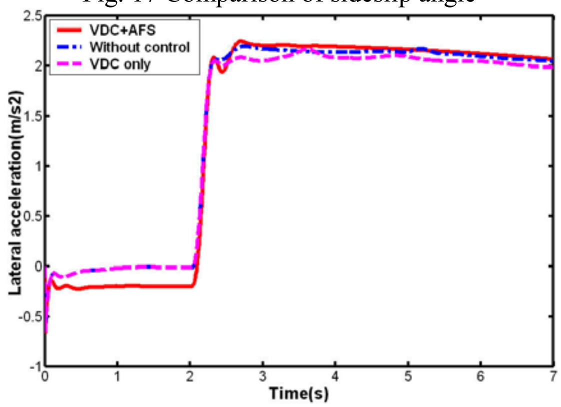

Fig. 18 Comparison of lateral acceleration

V. CONCLUSION

A simultaneous control technique based on the cooperation of ANFIS VDC controller and fractional-order $PI^{\lambda}D^{\mu}$ AFS controller has shown a good performance. The distribution of yaw moment is based on the control authority in understeer or oversteer correction. The training data of ANFIS is obtained from the simulation based on multibody vehicle dynamic model, and its performance would be better if the data comes from experiment. The parameters of the fractional-order $PI^{\lambda}D^{\mu}$ are tuned using GA optimization method, which ensure its good quality.

ACKNOWLEDGEMENTS

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