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Research Article

Vehicle Yaw Rate Control Based on Fuzzy PID Control Technology

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Abstract

In this paper, control of vehicle yaw stability system is studied by using a two vehicle models where the first one is a linear two degree of freedom vehicle model to design the controller and the other one is a planar motion model which represents a nonlinear vehicle model (actual vehicle). The strategy of vehicle yaw stability control based on the yaw rate control which adopts the fuzzy PID controller. Compared to traditional PID control, the fuzzy PID control can adjust and tune the proportional, integral and derivative parameters and make efficient the system responds. To make sure that Fuzzy PID controller works well, it will be tested at two cases of input steering angle of the vehicle front tires which are a step signal maneuver and a single-lane-change maneuver. Various of computer simulations and results show that the control system of vehicle yaw stability and using of fuzzy PID controller can improve the stability and handling of vehicle significantly.

Keywords: Yaw rate, Vehicle model, Fuzzy PID controller

Introduction

Stability of vehicles is the ability which vehicle could moves at a desired route by the driver without any rollover, sideslip and side skid [2]. Most control systems of the vehicle stability use the feedback control of the vehicles yaw rate to improve and achieve the proposed vehicle stability control. But in decrease of emergency case, such that when the vehicle moves at snowy road, the vehicle yaw rate must be controlled in order to decrease the vehicle over spin and side slip. Therefore, body sideslip angle and vehicle yaw rate is the parameter key of vehicle stability control. The yaw rate of the vehicle can be measured by gyro and some calculations depending on the differential equations of motion [3]. The linear and nonlinear vehicle models that described the behavior of lateral dynamic and yaw rate motion are explained for controller design and evaluation purpose. To achieve the control objectives, it is important to control the variables of both of yaw rate and sideslip angle to ensure the vehicle stable. It is required that the actual vehicle yaw rate and the vehicle sideslip angle have fast responses and good tracking capability in the desired responses [4].

Vehicle Model and Tire Model

Vehicle Model

Figure 1 shows a planar model of vehicle (3 degree of freedom model of vehicle) used in the work. This model represents a non-linear vehicle model and four-wheel drive 4WD, only consideration the planar movement: yaw, longitudinal, and lateral. The vehicle in this case is modeled and assumed as a rigid body with three degree of freedom 3DOF. The roll, bob, and pitch motions are neglected.

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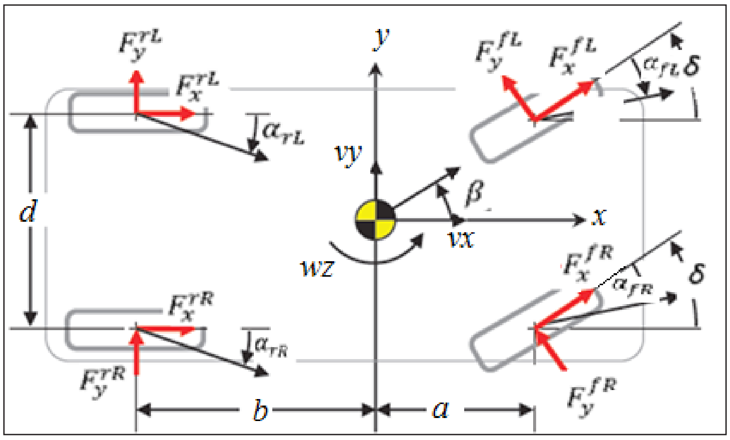

Figure 1.Planar model of the vehicle

The mathematical equations of motion of the vehicle can be represented as follows:

For yaw movement:

$$Iw_z = [a(F_x^{fR} + F_x^{fL}) \sin\delta + a(F_y^{fR} + F_y^{fL}) \cos\delta - b(F_y^{rL} + F_y^{rR}) + \frac{d}{2}(F_x^{fR} - F_x^{fL}) \cos\delta + \frac{d}{2}(F_x^{rR} - F_x^{rL}) + \frac{d}{2}(F_y^{fL} - F_y^{fR}) \sin\delta]$$

For longitudinal movement:

$$x_x - v_r w_z = \frac{1}{m}[(F_x^{fR} + F_x^{fL}) \cos\delta - (F_y^{fR} + F_y^{fL}) \sin\delta + F_x^{rL} + F_x^{rR}$$

For lateral movement:

$$v_y + v_x w_z = \frac{1}{m}[(F_y^{fL} + F_y^{fR}) \cos\delta + (F_x^{fL} + F_x^{fR}) \sin\delta + F_y^{rL} + F_y^{rR}$$

Where F_x^{fL}, F_y^{fL}, F_x^{fR}, F_y^{fR}, F_x^{rL}, F_y^{rL}, F_x^{rR}, F_y^{rR} are the components of forces for the front left tire, front right tire, rear left tire, and the rear right tire along *x* axis and *y* axis coordinates; *a* ,*b* are the displacement of the COG of the vehicle to both of front and rear axle; L_w is the displacement between left and right tires ; v_x, v_y are the car longitudinal and the car lateral velocity, w_z is the yaw rate of the vehicle, δ is the steering angle of front wheel, *m* is the vehicle total mass, *I* represents the moment inertia of the vehicle body about its yaw.

The input steering **δ** in this paper will set as a step signal which have an amplitude of two degrees (0.035 radians) as illustrated in the figure 2. Also, the input steering will set as a lane change maneuver with amplitude of front steering angle of 0.035 radians as obtained in figure 3.

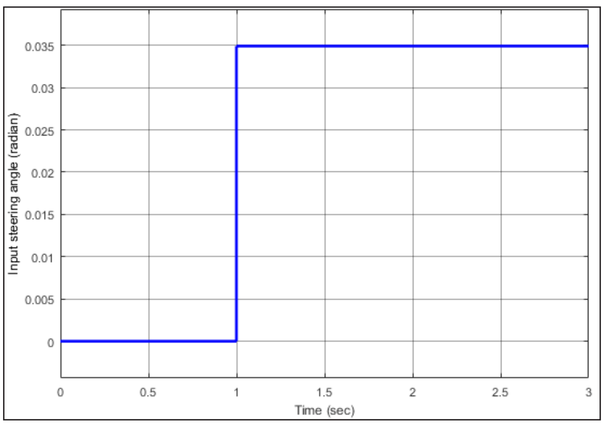

Figure 2.The steering input of vehicle maneuver

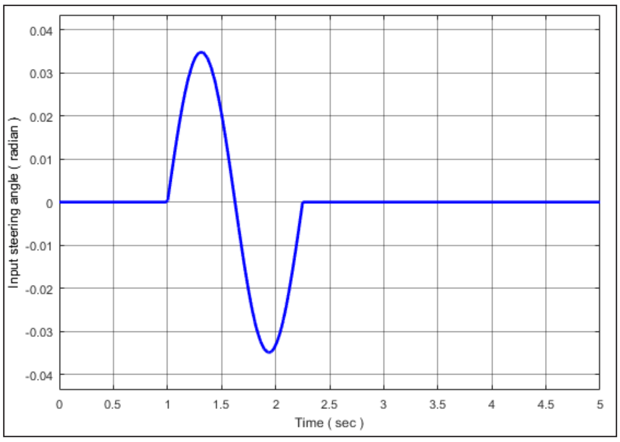

Figure 3.The steering input of vehicle with a lane change maneuver

The slip angle at each wheel is expressed and derived using the geometry of the vehicle and the vectors of wheel speed. If the velocity at each wheel road contact point is known then, it can easily derive the tire slip angle at each tire geometrically and can be expressed as follows:

$$\alpha_{fR} = \tan^{-1}\left[\frac{v_y + a.w_z}{v_x + \frac{d}{2}.w_z}\right] - \delta$$

$$\alpha_{fL} = \tan^{-1}\left[\frac{v_y + a.w_z}{v_x - \frac{d}{2}.w_z}\right] - \delta$$

$$\alpha_{rL} = \tan^{-1}\left[\frac{v_y - b.w_z}{v_x + \frac{d}{2}.w_z}\right]$$

$$\alpha_{rR} = \tan^{-1}\left[\frac{v_y + a.w_z}{v_x - \frac{d}{2}.w_z}\right]$$

Where α_{ij} is the slip angle at each individual tire.

Wheel Dynamics Model

A sketch diagram of a modeled tire is illustrated in the figure below. The moment of inertia of the wheel is I_w and the effective radius is R_w. The applied torque on each tire T_i and the longitudinal force of the tire F_{xi} is produced at the tire bottom. Each tire rotates with angular speed ω and travels at a longitudinal speed v_x. The summation of moments about the rotation axis of the tire generates the dynamic equation derived in the following equation:

$$T_1 - F_{x1}R_w = I_w\dot{w}_1$$

$$T_2 - F_{x2}R_w = I_w\dot{w}_2$$

$$T_3 - F_{x3}R_w = I_w\dot{w}_3$$

$$T_4 - F_{x4}R_w = I_w\dot{w}_4$$

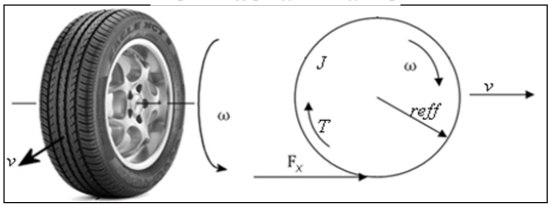

Figure 4.Wheel schematic diagram

Tire Model

Due to an extreme conditions of vehicle driving, the tire may rotate at a nonlinear behavior. The most common tire model which called the Dugoff's tire model is used in this paper. The Dugoff model permits an independent values of the tire stiffness in the lateral and longitudinal directions. Compared to Pacejka tire model (Magic Formula tire model), the Dugoff's tire model has the merit of being an analytically derived model which is developed from the calculations of force balance. Furthermore, the lateral and longitudinal tire forces are related to the tire-road friction coefficient directly in more transparent equations [3].

The Dugoff's tire model provides the forces calculation under combined generation of the longitudinal and lateral tire forces. The longitudinal tire force and the lateral tire force were derived as [3]:

$$F_x = C_1\frac{\sigma_x}{1+\rho_x}f(\lambda)$$

$$F_y = C_2\frac{\tan\alpha}{1+\rho_x}f(\lambda)$$

where C_1 is the longitudinal stiffness of the tire, C_2 is the tire cornering stiffness and λ is expressed by:

$$\lambda = \frac{\mu F_z(1+\rho_x)}{2\sqrt{[(C_1\rho_x)^2 + (C_2\,tan\,\alpha)^2]}}$$

and

$$f(\lambda) = (2-\lambda)\lambda \quad if\ \lambda < 1$$

$$f(\lambda) = 1 \quad if\ \lambda > 1$$

μ is the friction coefficient between the tire and road, F_z is the force acting on the tire in the vertical direction , while σ_x is the tire longitudinal slip ratio and can be derived in the accelerating case of as:

$$\sigma_x = \frac{v_x}{R_w w_i} - 1$$

Vehicle Reference Model

The bicycle model (reference vehicle model) as illustrated in figure 5 is an important part in the automotive engineering studies and vehicle dynamic field because it uses for controllers design and the analysis of yaw stability control prominently. According to some assumptions, it is possible to linearize the actual model of the vehicle (a nonlinear model), these assumptions are : the vehicle moves on plane surface or flat road (planar motion), the tires forces operate in a linear region, and the left and the right tires at the rear and front axle are placed in an unattached tire at the center line of a vehicle.

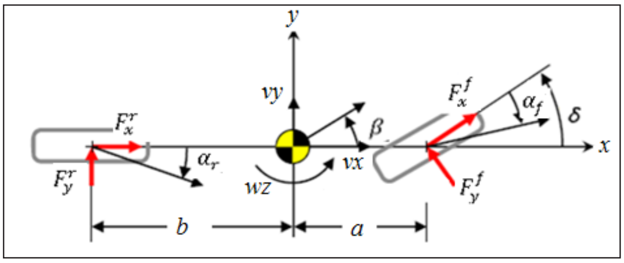

Figure 5.Bicycle (reference) model

The driver tries to control the vehicle's stability during normal and moderate cornering from the steer-ability point of view. The bicycle model gives a relationship between the stability factors and driver performance of the vehicle. Hence, the model is designed to generate a desired value of the yaw rate of the vehicle w_{zd} and the body sideslip angle β_d at each instance according to the driver's steering angle input and the vehicle forward velocity while considering of a constant speed.

The following equations represent the differential equations of yaw and lateral motions of the reference model [3]:

$$m\,(\dot{\beta}_d - w_{zd}) = (F_y^f + F_y^r) - w_{zd}$$

$$I_z \dot{w}_{zd} = a.F_y^f - b.F_y^r$$

$$F_y^f = C_{af}\,.\alpha_f$$

$$F_y^r = C_{ar}\,.\alpha_r$$

$$\alpha_f = \delta - \beta_d - \frac{a.w_{zd}}{v}$$

$$\alpha_r = -\beta_d + \frac{b.w_{zd}}{v}$$

Where:

w_{zd} and β_d are the desired yaw rate of the vehicle and desired vehicle sideslip angle. C_{af} and C_{ar} are the longitudinal and the lateral stiffness of front wheel and rear wheel. *a*, *b* are the isplacement of the COG of the vehicle to both of front and rear axle. The simulation results for the vehicle reference model at a step signal of steering angle and on a lane change maneuver which represent the desired vehicle yaw rate are obtained in figures 6 and 7.

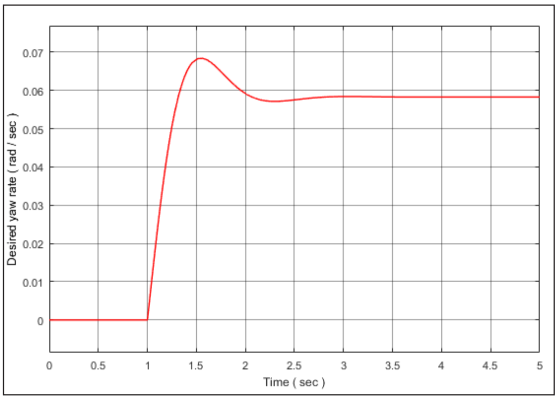

Figure 6.The desired vehicle yaw rate at a step signal of steering angle

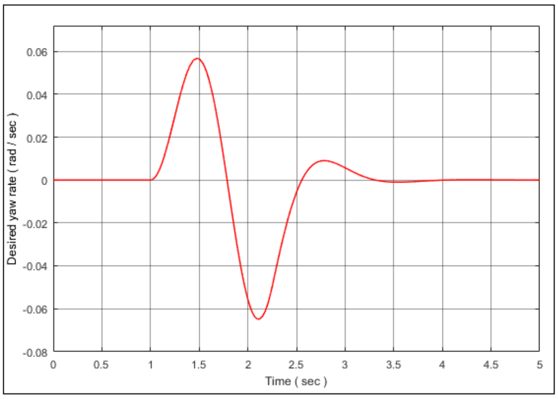

Figure 7.The desired vehicle yaw rate on a lane change maneuver

Control Strategy

In this paper, Fuzzy PID Controller FPIDC is designed to enhance and improve the yaw stability of four wheel drive vehicle. The fuzzy PID controller can be divided into equivalent proportional control PC, integral control IC and the derivative control DC parts. A control system is designed which includes three components: the Bicycle model reference vehicle model (reference vehicle model), planar model of the vehicle (A nonlinear model which represents actual vehicle), and the controller (Fuzzy PID Controller). The structure of the control system of vehicle model with controller is illustrated in Figure 8.

A method for determining the ideal values of vehicle yaw rate and sideslip angle can be done using reference model and based on the steering angle (δ) which can be derived through the driver action. The PID controller is commonly used control algorithm in various industries because of its simplicity, ready availability and faster processing time. The critical regulation of the PID parameters is a complex design task, while it is a very simple decision-making process in case of using fuzzy PID controller FPIDC [7].

In this study, the comparison between the sideslip angle β and the ideal sideslip βd is made and the ideal yaw rate wzd is also compared with the actual yaw rate wz. Then the calculation of the yaw rate error E and its change rate is done. The implementation of the fuzzy control is made as fellows: the first step is fuzzification of the input parameters such as yaw rate error E and its rate of change ΔE, followed by the fuzzification of the direct yaw moment Mz. Fuzzification of the input parameters transforms the actual input variables into fuzzy terms according to the selected member functions.

Simulation Results

The vehicle parameters which used in this paper are given in the table below (10).

The simulation results of planar vehicle model include comparison of the vehicle yaw rate performance and behavior in case of without control and in the other case due to using Fuzzy PID controller.

Figure 9 and figure 10 present the vehicle yaw rate at a step signal of steering angle and on lane change maneuver respectively.

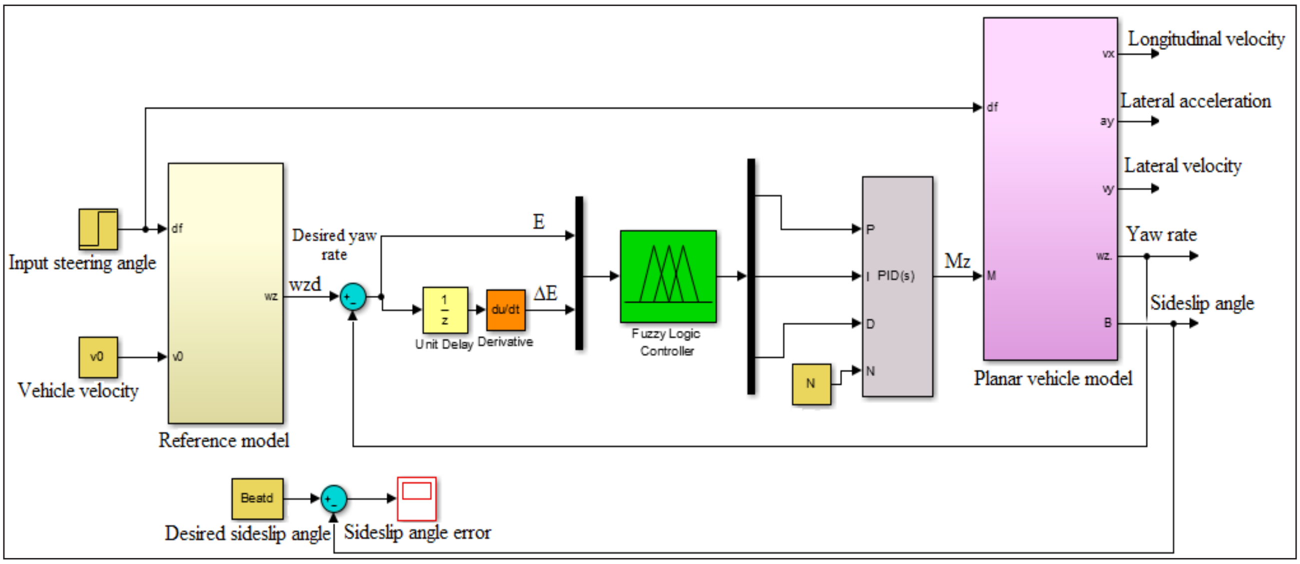

Figure 8.The structure of vehicle model with controller

Table 1.The physical parameters of the vehicle

Symbol	Unit	Value
m	Kg	1993
I	Kgm^2	2765
a	m	1.402
b	m	1.646
d	m	1.6
R_w	m	0.365
J	Kgm^2	4.07
μ	Non	1.0
C_1	N/rad	52526
C_2	N/rad	29000
g	m/s^2	9.81

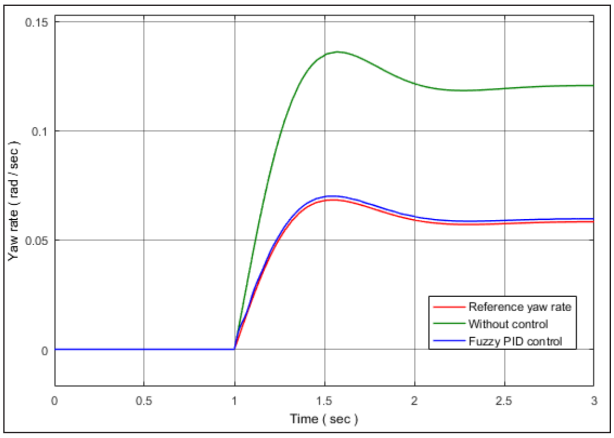

Figure 9.The vehicle yaw rate behavior with Fuzzy PID controller and without it at a step signal of steering angle

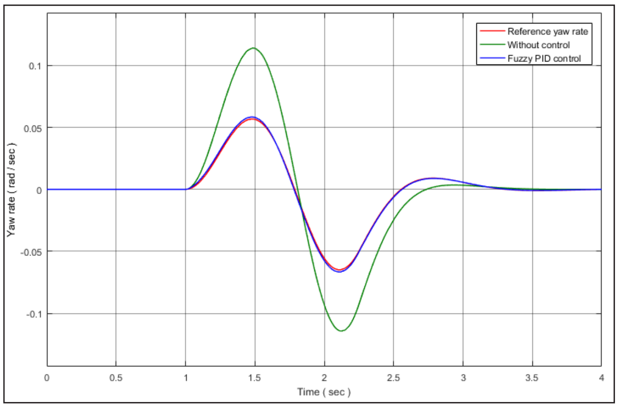

Figure 10.The vehicle yaw rate behavior with Fuzzy PID controller and without it on a lane change maneuver

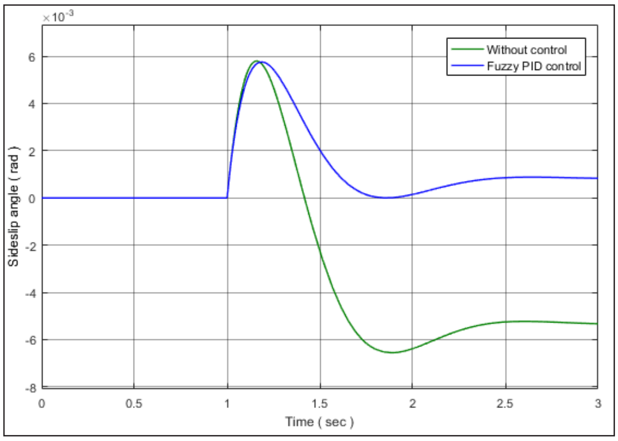

Figure 11.The vehicle sideslip angle with Fuzzy PID controller and without it at a step signal of steering angle

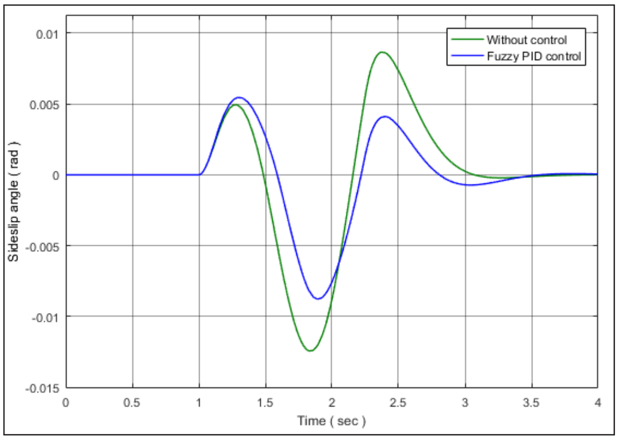

Figure 12.The vehicle sideslip angle with Fuzzy PID controller and without it on a lane change maneuver

Conclusion

Aim at improving the stability of vehicles and handling, the Matlab Simulink model of the vehicle is implemented using Fuzzy PID controller FPIDC to improve and ensure the vehicle yaw stability in this work. To make sure this controller works well it has been tested at two cases of input steering angle which are a step signal and a lane change maneuver. The plots and results show a significant differences between the vehicle yaw rate behavior in the case of when there is no control and the vehicle yaw rate behavior in the case with Fuzzy PID controller. It is found that, the yaw rate of the vehicle improved significantly, therefore, vehicle control system constructed with fuzzy PID controller can achieve and enhance the required stability and performance of the vehicle.

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