Fuzzy Logic Control of a Four Wheel Steering System

Evren Ozatay
Yeditepe University

Y. Samim Unlusoy
Middle East Technical University

Fuzzy Logic Control of a Four Wheel Steering System

Evren Ozatay, M. Sc.
Ford Otomotiv Sanayii, Kocaeli, Turkey

Prof. Samim Y. Unlusoy
Mechanical Engineering Department, Middle East Technical University, Ankara, Turkey

Murat A. Yildirim, Ph.D.
Ford Otomotiv Sanayii, Kocaeli, Turkey

ABSTRACT

Vehicle dynamics control systems may be used to integrate the driver's steering input together with the four-wheel steering system (4WS) in order to improve the vehicle's dynamic behavior with respect to yaw rate and body sideslip angle. The goal of this study is to develop a fuzzy logic controller for this purpose. In the first stage of the study, a three-degree of freedom nonlinear vehicle model including roll dynamics is developed. The Magic Formula is applied in order to formulate the nonlinear characteristics of the tires. Fuzzy logic controlled model is compared with front wheel steering vehicle and the vehicles having different control strategies that have previously been studied to actively control the rear wheel steering angle. The results of analysis indicate that the dynamic behavior of the fuzzy logic controlled vehicle is superior with respect to other strategies and front wheel steering vehicle in obtaining zero body sideslip angle even in transient motion and quick response of yaw rate during steady state cornering and lane change maneuvers.

Keywords: Four-Wheel Steering (4WS), Rear Wheel Steering, Fuzzy Logic Control (FLC), Fuzzy Logic

I. INTRODUCTION

There is an increasing trend toward sophisticated chassis control systems in vehicle design. The three main systems of chassis control are: lateral control, vertical control, and longitudinal control. These systems were developed independently to improve vehicle handling, ride comfort, and traction/braking performance as well as to relieve driver's workload. Among them, active four-wheel steering (4WS) systems enhance vehicle's cornering ability by steering the front and rear wheels in accordance with vehicle states. With such steering control systems, it becomes possible to improve the lateral stability and handling performance in a range where the vehicle dynamics could be described by linear models.

Four-wheel-steering (4WS) systems for passenger vehicles have been actively studied recently. The performance of these systems depends largely on how the rear wheels are controlled as functions of vehicle speed, steering angle, and/or states of the system. These rear steering controllers are usually designed to improve: (a) vehicle maneuverability at low speed, and (b) straight-line stability at high speed. Some 4WS vehicles operate by steering the rear wheels as a function of the front steer angle. Much of the research in 4WS has been in determining the degree of rear steering to complement the front steering for various steering maneuvers. It is well known among 4WS engineers that the magnitude and direction of rear steering - in-phase or out-phase - is a function of the vehicle's forward speed [1]. Steering of the rear wheels, in addition to steering of the front wheels, makes it possible to reduce the vehicle sideslip angle at the center of gravity of the vehicle to zero (Fig.1). This greatly improves maneuverability at low speed and stability at high speed.

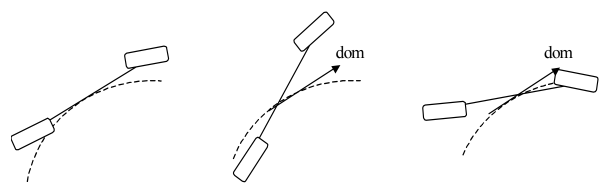

Fig.1. Schematic representation of the sign change of the sideslip angle.

In process industries FLC's (Fuzzy Logic Controllers) are becoming increasingly popular. They have been successfully used for a number of difficult processes; even sometimes they are proved to be more robust than conventional controllers. Fuzzy controllers are inherently nonlinear controllers, and hence fuzzy control technology can be viewed as a new, cost effective and practical way of developing nonlinear controllers. The major advantage of this technology over the traditional control technology is its capability of capturing and utilizing qualitative human experience and knowledge in a quantitative manner through the use of fuzzy sets, fuzzy rules and fuzzy logic. However, carrying out analytical analysis and design of fuzzy control systems is difficult not only because the explicit structure of fuzzy controllers is generally unknown, but also due to their inherent nonlinear and time varying nature.

The main goal of this study is to design and operate a controller that uses the basic functions of fuzzy logic. This controller is to improve the high-speed characteristics of the vehicle during cornering by controlling the rear wheel steering angle. Their research encompassed many aspects of vehicle dynamics and fuzzy logic. There are plenty of different control strategies that can be used in four-wheel steering systems. Sato et. al. [2] defined a yaw feedback and front steering angle controller. Sano [3] derived a controller, which yields zero sideslip angle at steady state. Whitehead [4] obtained a new controller by setting the rate of change of sideslip angle and sideslip angle to zero, which ultimately resulted in an uncoupled and uncontrolled system for the sideslip angle. Also, among them, only Will [5] and Szosland [6] used fuzzy logic approach in order to investigate the performance on controlling rear wheel angle.

The main motivation for investigating four-wheel steering systems for passenger cars is to increase the maneuverability during low speed related parking and stability especially for high-speed performance of the car; mainly to improve the handling behavior of a vehicle when steering is applied by the driver. This improvement will be achieved by reducing the sideslip angle to zero, as much as possible. At the same time, since the cornering performance is directly related with the yaw rate when the sideslip motion gets zero, in order not to loose the cornering ability of the vehicle, the controller should supply an optimum value for the yaw rate.

Initially, basic calculations and observations will be carried on simple Bicycle model. Since vehicle handling is the main purpose, longitudinal dynamics is taken as either zero or negligible. Lateral force interactions of tires and road surface will be analyzed by a nonlinear tire model. The variations on tire loads during cornering will be taken into account by considering a more realistic 3-dof model to obtain more reliable results. Overall study will be simulated numerically by using SIMULINK toolbox of MATLAB [7]. Under different situations and variable front wheel steering angles, the results are to be studied and compared with the controllers developed in the literature.

II. DYNAMIC MODEL OF THE VEHICLE

It is desirable to use a vehicle model that includes the important characteristics of vehicle motion without being unnecessarily complicated by nonlinear characteristics that are important only during maximum handling maneuvers and large angle deflections. So at the beginning of the modeling, simple "bicycle model" was used. Next, a three degree-of-freedom (3-dof) four-wheel steering vehicle model is used to develop and analyze fuzzy logic rear wheel controller. This model is used to investigate the effects of suspension design on vehicles handling and stability

A. Two Degree – Of – Freedom Vehicle Model – Bicycle Model

The essential features of car steering dynamics in horizontal plane are described by the "single-track model" (or "two wheel model", or "bicycle model"). The representation of slip angles for the front and rear axles, α_f and α_r, allows the combination of tire side forces for each side for n identical tires on each axle, i.e.,

$$F_f = nF_{11} = nF_{12} \tag{1}$$

$$F_r = nF_{21} = nF_{22} \tag{2}$$

Thereby, the car model is reduced to the one shown in Fig.2, and the coupling with roll, pitch, and bounce motions is not modeled.

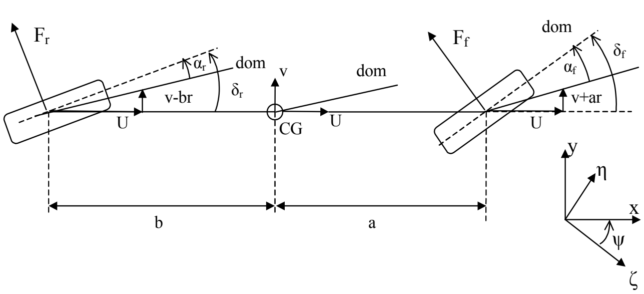

Fig.2. Kinematics variables of Bicycle model for car steering.

The distance between the center of gravity (CG) and the front axle (rear axle) is *a* (*b*) and together is the *L* = *a* + *b*, wheelbase. The angle between the vehicle's direction of motion and the longitudinal velocity, *U*, is called "vehicle sideslip angle", and denoted as β. In the horizontal plane of Fig.2, an inertially fixed coordinate system (ζ , η) is shown together with a vehicle fixed coordinate system (x , y) that is rotated by a "yaw angle", ψ. In the dynamic equations the yaw rate $\dot{\psi} = r$ will appear as a state variable.

Fig.3 shows a block diagram of the model. The side forces Ff and Fr are projected through the steering angles into chassis coordinates (x , y), where they appear as forces ΣFx, ΣFy and torque ΣM around z-axis which is pointing upward from the CG.

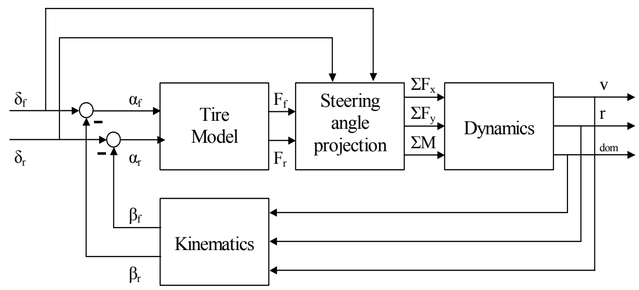

Fig.3. Block diagram of vehicle steering.

The forces transmitted from the road surface via the wheels to the vehicle chassis are represented in Fig.2 by the front axle side force, F_f, and rear axle side force, F_r. The forces in the longitudinal direction of the tires are assumed to be zero, i.e. the wheels are freely spinning. We do not model braking and the acceleration by the engine of the vehicle.

The equations of motions for the three degrees of freedom, i.e., longitudinal motion, lateral motion and yaw motion in the horizontal plane become:

$$\Sigma F_x = m \cdot a_x = m\left(\dot{U} - vr\right) \tag{3}$$

$$\Sigma F_y = m \cdot a_L = m\left(\dot{v} + Ur\right) \tag{4}$$

$$\Sigma M = J \cdot \alpha = J \cdot \dot{r} \tag{5}$$

In the next step, the sideslip angles, α_f and α_r, at the front and rear tires are obtained by a "kinematic model" from the steering angles δ_f, and δ_r. Fig.2 illustrates the vehicle motion around curvature having a radius of R.

The tire slip angles, α_f and α_r, can be defined by using Fig.2 as

$$\alpha_f = -\delta_f + \left(\frac{v + ar}{U}\right) \tag{6}$$

$$\alpha_r = -\delta_r + \left(\frac{v - br}{U}\right) \tag{7}$$

The feedback-structured model of Fig.3 is now completed by the nonlinear tire model

$$F_f = F_f\left(\alpha_f\right) \tag{8}$$

$$F_r = F_r\left(\alpha_r\right) \tag{9}$$

All four blocks in the diagram Fig.3 are nonlinear and can be linearized by additional assumptions, i to iv [8].

Solving above equations yields the linear state space model:

$$\begin{Bmatrix} \dot{v} \\ \dot{r} \end{Bmatrix} = \begin{bmatrix} \frac{C_f + C_r}{mU} & \frac{aC_f - bC_r}{mU} - U \\ \frac{aC_f - bC_r}{JU} & \frac{a^2C_f + b^2C_r}{JU} \end{bmatrix} \begin{Bmatrix} v \\ r \end{Bmatrix} + \begin{bmatrix} -\frac{C_f}{m} & -\frac{C_r}{m} \\ -\frac{aC_f}{J} & \frac{bC_r}{J} \end{bmatrix} \begin{Bmatrix} \delta_f \\ \delta_r \end{Bmatrix} \tag{10}$$

B. Three Degree-Of-Freedom Vehicle Model

The handling of the vehicle must also be checked using a three-degree of freedom (yaw, sideslip, and roll) model, as long as the suspension data are available. Such an analysis normally requires a mathematical model of the vehicle including the suspension system and the solution is normally too complicated for hand calculations [8].

An approach to include the roll motion into the simple vehicle handling model, still keeping the analysis sufficiently simple to obtain useful results by hand calculation is possible through the concepts of *roll centers* and *roll axes* [8]. As shown in Fig.4, for a vehicle, there are two roll centers, one for the front suspension and one for the rear suspension. In real applications, as a result of suspension deflection during roll, the positions of the roll centers are also variable. The basic assumption in the roll axis approach is that the roll center locations do not change. In spite of this severe assumption, the approach still gives an insight into the directional behavior of vehicles.

The vehicle model is derived by the Newtonian method. The sprung mass rotates counterclockwise as the unsprung masses subjected to lateral forces to the right. The lateral motion can be obtained by balancing the lateral forces. The yaw rotation can be obtained by taking moment balance of the unsprung masses along the Z_L direction as shown in Fig.4. The suspension roll dynamics is obtained by the moment balance of the sprung mass centered at the sprung mass center.

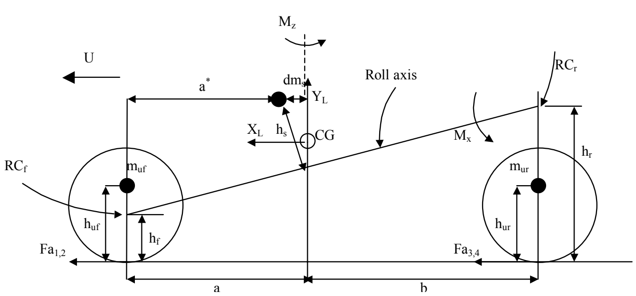

Fig.4. Vehicle model mass distribution in the x-z plane.

Equations of motion for three degree of freedom model take the form:

$$m(\dot{v}+Ur)-m_s h_s \ddot{\phi}\cos\phi = m_s g h_s \phi-(K_f+K_r)\phi-(C_f+C_r)\dot{\phi} \quad (11)$$

$$I_x\ddot{\phi}-m_s(\dot{v}+Ur)h_s\cos\phi = F_{b_1}+F_{b_2}+F_{b_3}+F_{b_4} \quad (12)$$

$$I_z\dot{r} = a(F_{b_1}+F_{b_2})-b(F_{b_3}+F_{b_4}) \quad (13)$$

III. Dynamic Model of the Tires

Within this study, the numerical/empirical tire model is used. This model commonly used in vehicle dynamics simulations was developed by H. Pacejka [9]. This is the first trigonometric model among many experimental (empirical) tire models found in literature. The Pacejka tire model calculates lateral force and aligning torque based on slip angle and longitudinal force based on percent longitudinal slip. The model parameters are dependent on the normal force, F_z, on the tire.

Standard SAE definitions are used for the wheel axis system, motion variables and forces. In addition, the camber angle, *γ,* was taken as zero for the tires and the effects of the self-aligning torque, M_z, were neglected in this study.

The Pacejka tire model is called the 'Magic Formula' because of its power to provide many important tire functions accurately. In its full form, the magic formula is given as:

$$Y = S_y + D\sin\left\{C\tan^{-1}\left[B\cdot(X-S_x)\cdot(1-E)+E\tan^{-1}(B\cdot(X-S_x))\right]\right\} \quad (14)$$

where *Y* represents any tire response quantity (lateral force, longitudinal force, self-aligning torque, etc.). The independent variable *X* may be lateral slip angle, longitudinal slip, or camber angle. Here, in this case, since the vehicle handling and cornering characteristics are considered *Y* will be taken as lateral force and *X* as lateral slip angle.

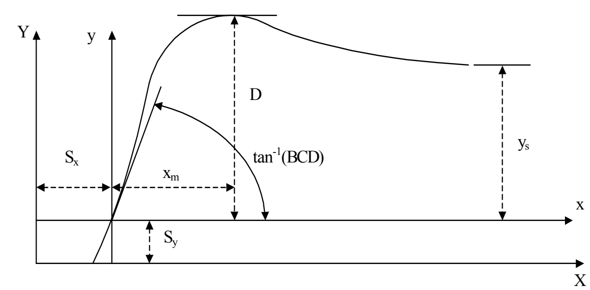

Fig. 5. Typical tire characteristics of Magic Formula

The coefficients in the "Magic Formula" must be identified from the experimental data using the nonlinear curve-fitting algorithms. For rapid execution of the iteration process during fitting and to ensure convergence, it is essential to generate initial estimates of the six coefficients that are close to their final values as shown in Fig.5.

The y(x)-curve passes through the origin x = y = 0 and is anti-symmetric with respect to the origin. However, in the real situation Y(X), the offsets S_x and S_y, appear due to the effects of a conicity, plysteer, and rolling resistance on F_y and α and also the camber angle on F_y and α. These offsets cause the deviation from the anti-symmetric shape. In this study the effect of camber angle is neglected in determining cornering forces. (Fig.6)

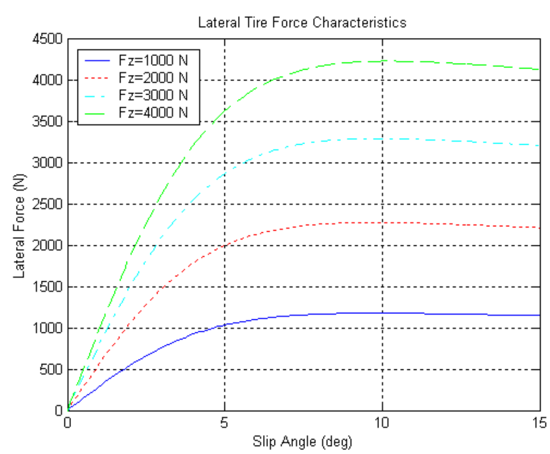

Fig. 6. Variation of cornering force (lateral force) with normal load and slip angle.

IV. CONTROLLER DESIGN

The structure of fuzzy controller can be examined for non-linear vehicle model used in this study relating handling characteristics of the vehicle during specified steering inputs are given.

A. Design Strategy For Fuzzy Controller

In designing a fuzzy controller, the following issues are required to be involved:

1) Select a reasonable structure for the fuzzy controller:

The first step for designing a fuzzy controller is to select and determine a reasonable structure for the fuzzy controller. Selecting a structure for the fuzzy controller means that the input and output variables of the fuzzy controller are required to be determined. In this study error signal "e" and error rate "Δe" are selected as input variables of the fuzzy controller and the change of controlled variable as output variable.

2) Select and extract fuzzy control rules:

The fuzzy control rules are the core of the fuzzy controller. To obtain a better fuzzy logic system, several important aspects related to the given performance requirements must be taken into account in deriving the fuzzy rules. In general, asset of fuzzy rules is derived considering the following four performance issues:

- Rapid response to large errors
- Prevent system overshoot
- Suppressing oscillation
- Zero steady state error

3) Determine strategies for fuzzification and defuzzification and define control table:

In building a fuzzy control system, at first all errors and error rates are needed to be transformed fuzzy inputs from crisp inputs. As we have seen, this process is called fuzzification. The fuzzy set U of control rates will then be transformed into the crisp output for the execution of the control. We also have known that this process is called defuzzification or fuzzy decision. In this study, 'center of area' method is used to realize defuzzification.

4) Determine parameters of the fuzzy controller:

The practical scope of the error and error rate in a control system is called basic universe of discourse of the quantized variables. During designing a specific fuzzy controller the universe of discourse for all inputs and output has to be determined. In this study the universe of discourses of input and output variables as one. However, scaling units are included to the structure of the fuzzy logic controller, which enables the controller to amplify the inputs and the output to a desired range.

B. Performance criteria

The steering system is a challenging control problem because the vehicle-tire dynamics are highly nonlinear with uncertain time-varying parameters. Intelligent controllers, such as fuzzy or neural, overcome these issues. Fuzzy controllers have the benefit of not requiring a mathematical model of the plant, while still being highly robust. Also, certain fuzzy control designs can be implemented that have the ability to learn or to adapt themselves to improve its performance. Because of these features, fuzzy controllers have been successfully implemented in the automotive field for controlling both wheel dynamic and vehicle dynamics.

A common control objective in a design of a control system in vehicle dynamics involves compensating for tire nonlinearity. Tire nonlinearity is the primary factor that is accounted for in our fuzzy modeling as well as in our fuzzy controller design. In this design approach, fuzzy modeling is used to specify our control objectives analytically. A design objective in developing a 4WS control system is to minimize the vehicle 's sideslip angle, β.

C. Design procedure

In this section, the 4WS control system design using a vehicle fuzzy model as a design model is presented. Just like conventional *nonfuzzy* control, which has two-term, and three-term control, the conventional fuzzy control also has two-term and three-term control.

1) Fuzzy Two-term Control

The fuzzy two-term control has two different types: one is Fuzzy-Proportional-Derivative (FZ-PD) type control which generates control output (u) from error (e) and change in error (Δe) and is a position type control; the other is Fuzzy-Proportional-Integral (FZ-PI) type control which generates incremental control output (Δu) from error (e) and change in error (Δe) and is a velocity type control.

The continuous-time algorithm of fuzzy two-term controllers, FZ-PI type and FZ-PD type control contains K_e and K_d, which are input scaling gains and K_1 and K_2, which are the output scaling gains, as shown in Fig.7. The fuzzy input parts are the same for both FZ-PI type and FZ-PD type control. The main differences lie in the rule base and fuzzy output processing. The FZ-PI type control contains an integral action in the output. The output of FZ-PI type control is $\dot{u}$, which can be called as the velocity type output. The output of FZ-PD type control, on the other hand, is u.

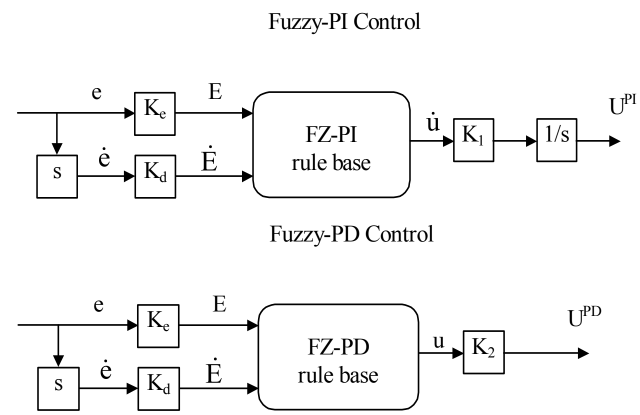

Fig.7. The continuous-time structure of fuzzy two-term control.

2) Fuzzy Three-term Control

The fuzzy three-term control is Fuzzy-Proportional-Integral-Derivative (FZ-PID) type control. There are too many different types of Fuzzy PID controller structures found in the literature. The earliest type of FZ-PID controller generates incremental control output from error (e), change in error (Δe) and acceleration error ($\Delta^2 e$). On the other hand, the acceleration error term has little influence on the performance because of limited measurement and computer resolution. Mann [87] illustrates the basic 6 types of Fuzzy PID controllers. Among them, two-input fuzzy PID type is selected for the simulations in this study. The rule base used is two dimensional instead of three-dimensional one. By combining both PI and PD actions as shown in Fig.8, a two-input fuzzy PID controller can be formed. This type of PID controller shares a common rule base for both FZ-PI and FZ-PD parts. The rule base structure is identical to Mamdani-type fuzzy PI controller as PI control is normally more important for steady state performance.

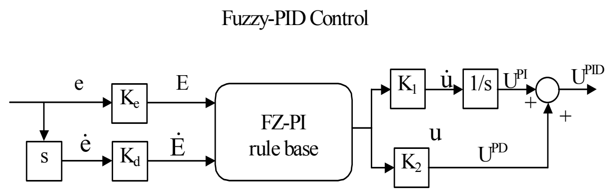

Fig.8. The general structure of two-input FZ-PID control.

Moreover, Li and Gatland [10] defined a 'fuzzy PI+D' (FZ-PI+D) type controller, which brings a derivative effect on the fuzzy PI type controller.

FZ-PI type control is known to be more practical than FZ-PD type because it is difficult for the FZ-PD to remove steady state error. The FZ-PI type control is, however, known to give poor performance in transient response for higher order process due to the internal integration operation. To improve the performance of FZ-PI type control, a method with resetting capability has been proposed [11]. However, the method needs another rule-base for resetting capability. Theoretically, FZ-PID type control should enhance the performance a lot. However, the existing FZ-PID type control needs three inputs, which will expand the rule-base greatly and make the design more difficult. Although some approximations on acceleration error (Δ^2e) can reduce the difficulties, the performance is not improved much over FZ-PI because of the small influence of acceleration error in general.

V. Performance Analysis

Throughout the study, the fuzzy toolbox of MATLAB [12] is used. The mamdani fuzzy inference method is implemented in the fuzzy toolbox of MATLAB. The fuzzy controller uses the '*and*' operator and implication are realized by '*min*' function. The defuzzification is performed by the 'centroid method'.

For simulation, the performance of the designed FZ-PID type controller will be compared with some of the strategies used to control the rear wheel steering angle during cornering maneuvers with the. One is the control strategy (ST2), which ensures zero sideslip angle during steady state cornering [3]. To obtain the expression for this control strategy, one simply sets v = 0 in the steady state equations of motion and solves for rear wheel steering angle, δ_r, in terms of yaw rate, r, and front wheel steering angle, δ_f. Next control strategy (ST1) used in the simulations composed of the yaw velocity feedback in addition to the proportional feedforward of δ_f. It will give zero vehicle sideslip angle even in the transient part of cornering. Last controller type (ST3) uses only yaw feedback in determining the rear wheel angle [2]. The interesting property of this control strategy is that it gives neutral steer characteristics to the 4WS vehicle when applied, irrespective of the forward speed and the handling characteristics of the original FWS vehicle. In simulations, also the performance of the two wheel steering vehicle is implemented in order to compare the appropriateness of the FLC designed in this study.

Two case studies are presented here. In the first one steady state cornering having 1 degree of front wheel steering input at steady state is used. The second one is again uses the similar front wheel steering input, however, the steady state value and amplitudes for the fuzzy logic controller is 1.75 degrees instead of 1 degree, which is the corresponding value that results in same amount of yaw rate with 2WS and FLC vehicle. This improvement is utilized in order to investigate the advantages of fuzzy logic controller over front wheel steered vehicles.

Case1: Steady state cornering

Notice that the starting point of this study is to make the sideslip angle zero whatever condition takes place. According to the Fig. 9, ST2 first increases to 0.3 degree and reaches a zero steady state sideslip angle at time becomes 2 seconds. This may be applicable with this respect but when considering yaw rate response, its poor behavior results in undesirable huge radius of curvature during cornering. This disadvantage is also observable in lateral acceleration curves. Again a low value of lateral acceleration, dominates the gain that ST2 provides in sideslip angle performance.

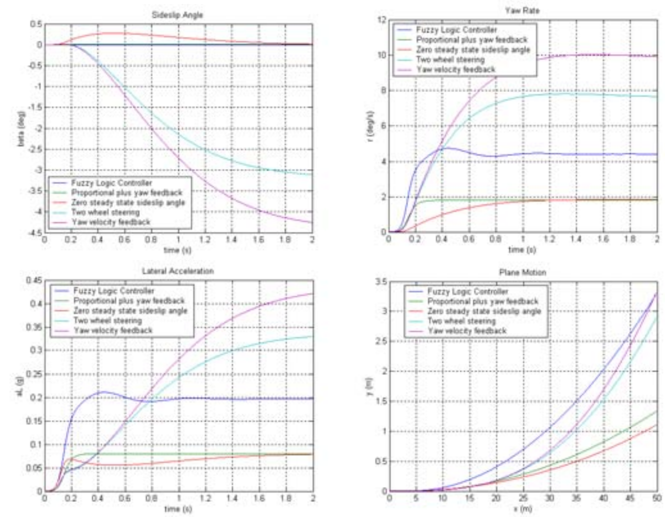

Fig.9. Comparison of sideslip angles, yaw velocities, lateral accelerations and lateral displacements in x-y plane for case 1.

The main property of ST1 is giving a zero sideslip angle even in the transient period. A couple of words are worth to say about the performance of yaw response of ST1. Its steady state value is as low as ST2 and is not desirable when compared with other strategies-even with 2WS-, but the response time is quite high as that of the ST2. There is nearly 1 second between the times they reach the final value of yaw rate. When one compares ST1 and ST2, although their sideslip angle and yaw rate responses are similar, there is a big difference in the angle of rotation of rear wheel. ST1 turns 0.75 degrees under the effect of 1 degree of front wheel steering angle, however, the angle of rotation of rear wheel in ST2 is approximately 0.05 degrees.

ST3 is also en extreme case, which provides relatively fast but a huge amount of yaw velocity response since it only considers yaw motion and feedbacks the yaw rate without taking the sideslip angle into account. Hence, a poor response of sideslip angle is achieved when compared with the others. Another interesting feature is that this controller rotates the rear wheel in the opposite direction with that of the front. It is expected and is the reason of that much yaw velocity steady state value.

Case2: Steady state cornering - improved

Fuzzy Logic Controller that is designed in this study should bring a worth-to-design alternative over 2WS vehicles. Actually, just obtaining a zero sideslip angle for all the times, including transient period, of simulation and advantages of gaining a good stability and controllability by this way, make the FLC controller be a good idea to implement on 2WS vehicles in order to improve by also actively controlling the rear wheels, anyhow.

Nevertheless, as stated in the previous cases, the FLC provides a better response characteristic of yaw rate such as fast response behavior when compared with the other strategies and 2WS. This feature will be developed in this case by giving a relatively small increase, from 1 degree to 1.75 degrees, to the steering wheel of the car to turn the vehicles front wheels. Normally, steering gear ratios is designed to be about 15-20. In order to get 0.75 degrees of enhancement in the front wheel steering angle, the driver should turn the steering wheel approximately 12 degrees more. This case 3, therefore, is worth to investigate the additional performance benefit that may be gained from the FLC.

Fig.10 illustrates that a slight increase in the front wheel steering angle improves the sideslip, yaw rate, lateral acceleration and yaw angle responses of the vehicle. This improved response reaches steady state value faster, is less oscillatory and has less overshoot. When one compares 2WS with the four-wheel steering system vehicle having a fuzzy logic controller, it should be noticed that the lateral accelerations do not differ much. However, the time to reach the final value is reduced by 1.5 seconds, which is relatively high value as the vehicles 90-km/h forward speed is taken into consideration.

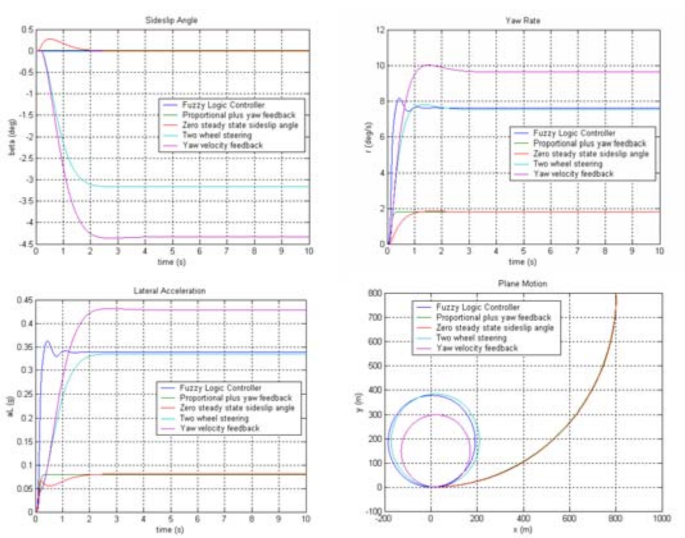

Fig.10. Comparison of sideslip angles, yaw velocities, lateral accelerations and lateral displacements in x-y plane for case 2.

In obtaining lateral displacements in x-y plane, the simulation is run for 50 seconds to get the complete rotation of FLC, 2WS and ST3. ST1 and ST2 could only make 90 degrees of yaw angle on the x-y plane. They took a distance of 800 meters in y-direction as it shifts 800 meters in x-direction. These distances are a little bit higher than 200 meters for 2WS and lower than 200 meters for FLC. ST3, on the other hand, travels 175 meters both in x and y directions to complete a quarter of a circle.

VI. Conclusions

In this study, the FLC is proposed as a design model to construct a four-wheel steering control system for an understeered standard medium passenger car. Here, a method, controlling the rear wheel steering angle based on fuzzy theory, has been derived. The control method feeds back the sideslip angle. It is, however, made clear that a four-wheel steering system that yields a zero sideslip angle is not the only outcome from this study. The effect of increasing the response time of yaw motion is worth to mention here.

The simulations given in the previous section show the sideslip angle can be significantly reduced. It means that vehicle stability and controllability can be improved, and driving performances become better. Besides, the simulation results proved that the FLC type active rear wheel steering system greatly improves the response time of yaw rate, and hence the cornering ability.

References

[1] Shibahata, Y., Irie, N., Itoh, H., and Nakamura, K., “The Development of an Experimental Four-Wheel Steering Vehicle”, **Society of Automotive Engineers Transactions**, No. 860623, pp. 862-869, 1985.

[2] Sato, H., Hirota, A., Yanagisawa, H. and Fukushima, T., “Dynamic Characteristic of a Whole Wheel Steering Vehicle with Yaw Velocity Feedback Rear Wheel Steering”, **I. Mech. E.** 124/83, Road Vehicle Handling, pp. 147-156, 1983.

[3] Sano, S., Furukawa, Y. and Shiraishi, S., “Four Wheel Steering System with Rear Wheel Steer Angle Controlled as a Function of Steering Wheel Angle”, **Society of Automotive Engineers Transactions**, Vol. 95. No. 860625, pp. 880-893, 1985.

[4] Whitehead, John, C., “Four-Wheel Steering: Maneuverability and High Speed Stabilization”, **Society of Automotive Engineers Transactions**, Vol. 97, No. 880642, pp.1-12, 1988.

[5] Will, Anthony B., Intelligent Vehicle Braking and Steering Control Systems, **Master’s Thesis**, Purdue University, 1997.

[6] Szosland, A., “Fuzzy Logic Approach to Four-Wheel Steering of Motor Vehicle”, **Int. J. of Vehicle Design**, Vol. 24, No. 4, pp.350-359, 2000.

[7] MATLAB by Mathworks, Version 5.1.0.450 Release 12.1.

[8] Unlusoy, Y. S. **ME 513 Vehicle Dynamics Lecture Notes**, METU, 2001.

[9] H. Pacejka, E., Bakker, L., and Nyborg, “Tire modeling for use in vehicle dynamics studies”, **Society of Automotive Engineers Transactions**, No. 870432, 1987.

[10] Li, H. X., and Gatland, H. B., “Enhanced Methods of Fuzzy Logic Control”, **IEEE Trans.Syst., Man, Cybern.**, pp. 331–336, 1995.

[11] Lee, J., “On Methods for Improving Performance of PI-Type Fuzzy Logic Controllers**”, IEEE Transactions on Fuzzy Systems**, vol. 1, no. 4, pp. 298-301, 1993.

[12] Fuzzy Logic Toolbox of MATLAB, Version 5.1.0.450 Release 12.1.