Adjustable Robustness Method for Fuzzy Logic Integrated Control of Active Steer Angle and Direct Yaw Moment

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Abstract

In this paper, a modified fuzzy logic control for integrated vehicle dynamics control is presented. This modification is derived from incorporating data exponential forgetting technique into the standard fuzzy controller. The proposed controller is capable of adjusting the robustness of the fuzzy logic controller based on operating conditions, considering uncertainties and nonlinear characteristics of vehicle dynamics. The controller uses yaw rate and side slip angle errors and their integrals as inputs in order to compute the total yaw moment and lateral force required for desired responses. The subsystems considered in the integrated control system are active steering and direct yaw moment controls. To show the efficiency of the proposed controller and also to investigate the benefits of the modified controller over the standard fuzzy logic control schemes simulation during a severe lane change maneuver are executed. Simulation results reveal that the proposed controller can effectively enhance the vehicle stability and handling performances, and also show the advantages of modified controller compared to the standard form.

Keywords: *Data exponential forgetting, fuzzy logic control, integrated control, vehicle dynamics control*

1. Introduction

In recent years by development of electronic control devices, the vehicle handling and passenger safety systems have been enhanced efficiently. Active Front Steering (AFS), Active Rear Steering (ARS), and Direct Yaw moment Control (DYC) are some of the systems which are developed through fast extension of electric and electronic technologies. In line with this progress, electronic systems have been employed successfully for vehicle control [1, 2-3]. The most prevalent and the main features that the vehicle stability and handling involve are the side slip stability and yaw rate tracking. In this regard, generally, two techniques have been implemented to influence these aspects. These are Active Steering Control (ASC) and direct yaw moment control.

The vehicle dynamics is a challenging control problem because tire characteristics and vehicle brake dynamics are highly nonlinear with uncertain time-varying parameters. Therefore, the control algorithm used for vehicle dynamics should be highly robust.

Intelligent controllers, such as fuzzy, can 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 controller designs improve the performance by learning and adaption features. Because of these characteristics, fuzzy controllers have been successfully implemented in the automotive field for controlling both wheel dynamics and vehicle dynamics [4].

In this paper an innovative controller based on integration of data exponential forgetting technique and fuzzy mode control scheme has been introduced. The proposed controller is designed in such a way that makes the vehicle follow the desired trajectory obtained from the driver's command. To this end, the yaw rate and side slip angle are computed based on a simplified two degrees of freedom vehicle (known as the bicycle model), then these two parameters and their integrals are fed into controller in order to compute the total yaw moment and lateral force required to guide the vehicle.

The paper is organized as follows: first, vehicle dynamics control containing two separate standard fuzzy logic controllers, one for side slip angle and the other for yaw rate control are presented. Then the controllers are modified to provide adjusting ability for robustness of the fuzzy logic controller, in the presence of uncertainties and severe nonlinear vehicle dynamics. Lastly, by applying the proposed integrated control scheme to a 9DOF nonlinear vehicle model, effectiveness of the control system is demonstrated through simulation using MATLAB and SIMULINK.

2. Vehicle and Tire Model

2.1. 9DOF non-linear vehicle simulation model

The non-linear model used for simulation of vehicle behavior is a two-mass system. In Figure 1, m_s and m_{us} denote the mass centers of the sprung and the unsprung masses respectively. The body fixed coordinate system (x, y, z) is attached to the mass centre of the unsprung mass. Position of the sprung mass in this coordinate is given by $(\bar{x}_s, \bar{y}_s, \bar{z}_s)$.

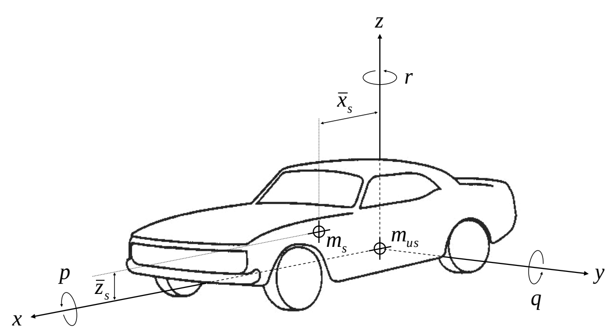

Figure 1. Vehicle coordinate system

The 9DOF model includes body longitudinal and lateral motions in the x and y directions, roll, pitch and yaw motions relative to the x, y, and z axes, and the rotational motion of four wheels. It is assumed:

1) Positional variations of the roll and pitch axes are negligible. Also these axes are parallel to the road plane.
2) Vertical displacements of the vehicle are small enough to be ignored.
3) Aerodynamic forces are not considered.

Thus vehicle dynamic equations are given as

Longitudinal motion:

$$\sum F_x = m(\dot{u} - vr) - m_s \bar{x}_s (q^2 - r^2) + m_s \bar{z}_s (pr + \dot{q}), \tag{1}$$

Lateral Motion:

$$\sum F_y = m(\dot{v} + ur) + m_s \bar{z}_s (qr - \dot{p}) + m_s \bar{x}_s (pq + \dot{r}), \tag{2}$$

Roll motion:

$$\begin{aligned} \sum M_x = {} & I_{xs}\dot{p} - (I_{ys} - I_{zs})qr - I_{xzs}(pq + \dot{r}) \\ & - m_s \bar{z}_s (\dot{v} + ur), \end{aligned} \tag{3}$$

Pitch motion:

$$\begin{aligned} \sum M_y = {} & I_{ys}\dot{q} - (I_{zs} - I_{xs})pr + I_{xzs}(p^2 - r^2) \\ & + m_s \bar{z}_s (\dot{u} - vr) - m_s \bar{x}_s (-uq + vp), \end{aligned} \tag{4}$$

Yaw Motion:

$$\begin{aligned} \sum M_z = {} & I_{z(total)}\dot{r} - (I_{xs} - I_{ys})pq + I_{xzs}(qr - \dot{p}) \\ & + m_s \bar{x}_s (\dot{v} + ur), \end{aligned} \tag{5}$$

where u and v refer to the velocities of the vehicle in the longitudinal and lateral directions, p, q, and r denote the body roll, pitch, and yaw velocities respectively, the elements of the tensor I_{ijs} $(i, j = x, y, z)$ stand for the inertia tensor of sprung mass with respect to the (x, y, z) axes, $I_{z(total)}$ represents the total inertia of the vehicle including the sprung and unsprung masses about the z axis, and m is the total mass of the vehicle. $\sum F_x$,$\sum F_y$,$\sum M_x$,$\sum M_y$,$\sum M_z$ are the sum of the external forces acting on the vehicle.

Referring to the Figure 2 equations of motion are given by:

$$\begin{aligned} \sum F_x = {} & F_{x1}\cos\delta_1 + F_{x2}\cos\delta_2 - F_{y1}\sin\delta_1 - F_{y2}\sin\delta_2 \\ & + F_{x3}\cos\delta_3 + F_{x4}\cos\delta_4 - F_{y3}\sin\delta_3 - F_{y4}\sin\delta_4 , \end{aligned}$$

$$\begin{aligned} \sum F_y = {} & F_{x1}\sin\delta_1 + F_{x2}\sin\delta_2 + F_{y1}\cos\delta_1 + F_{y2}\cos\delta_2 \\ & + F_{x3}\sin\delta_3 + F_{x4}\sin\delta_4 + F_{y3}\cos\delta_3 + F_{y4}\cos\delta_4 , \end{aligned}$$

$$\sum M_x = m_s \bar{z}_s g\phi - (k_{\phi f}\phi + c_{\phi f}\dot{\phi}) - (k_{\phi r}\phi + c_{\phi r}\dot{\phi}),$$

$$\sum M_y = m_s \bar{z}_s g\theta - (k_{\phi f}\theta + c_{\phi f}\dot{\theta}) - (k_{\phi r}\theta + c_{\phi r}\dot{\theta}),$$

$$\tag{6}$$

$$\begin{aligned}\sum M_z &= L_f\left(F_{x1}\sin\delta_1 + F_{x2}\sin\delta_2 + F_{y1}\cos\delta_1 + F_{y2}\cos\delta_2\right)\\ &- L_r\left(F_{x3}\sin\delta_3 + F_{x4}\sin\delta_4 + F_{y3}\cos\delta_3 + F_{y4}\cos\delta_4\right)\\ &+ \frac{d}{2}\left[\sum_{i=1}^{4}(-1)^i F_{xi}\cos\delta_i + \sum_{i=1}^{4}(-1)^{i-1}F_{yi}\sin\delta_i\right]\\ &- m_s g\bar{x}_s\phi,\end{aligned}$$

In the above equations, F_{xi} and F_{yi} denote the longitudinal and lateral forces of the ith tire, δ_i is the steering angle of the ith tire, ϕ and θ are the roll and pitch angles of the vehicle body, $k_{\phi f}$ and $k_{\phi r}$ indicate the stiffness of front and rear suspensions, $c_{\phi f}$ and $c_{\phi r}$ represent the front and rear damping coefficients of the suspension, L_f and L_r are the mass center distances from the front and rear axles respectively, and d is the vehicle width. $\sum F_x$,$\sum F_y$ are the sum of the longitudinal forces and the sum of the lateral forces in the vehicle body fixed coordinate system, representing all tire forces. Since these forces are given in the wheel-fixed coordinate system, the lateral and longitudinal tire forces for each wheel must be transformed to vehicle body fixed coordinate.

2.2. Tire model and wheel dynamics

The accuracy of the simulation results is predominantly determined by the accuracy of the tire model. The tire model must be able to represent the nonlinearity and coupling effects of longitudinal and lateral forces of the tire because the DYC and ASC deals with the vehicle dynamics in cornering and braking. Therefore, we formulated the tire behavior using Dugoff's tire model, where the longitudinal and lateral forces are given by [5]:

For longitudinal forces:

$$F_{xi} = C_\sigma \frac{\sigma_i}{1+\sigma_i} f(\lambda_i), \qquad i = 1,\ldots,4 \tag{7}$$

And for lateral forces:

$$F_{yi} = C_\alpha \frac{\tan(\alpha_i)}{1+\sigma_i} f(\lambda_i), \qquad i = 1,\ldots,4 \tag{8}$$

In which:

$$f(\lambda_i) = \begin{cases}(2-\lambda_i)\lambda_i, & \lambda_i < 1\\ 1, & \lambda_i \geq 1\end{cases}$$

$$\lambda_i = \frac{\mu_i F_{zi}(1+\sigma_i)}{2\left\{(C_\sigma\sigma_i)^2 + (C_\alpha \tan(\alpha_i))^2\right\}^{0.5}}, \qquad i = 1,\ldots,4 \tag{9}$$

In the above equations σ_i is the wheel slip, α_i is slip angle also, C_σ and C_α are the positive longitudinal and cornering stiffness of tires respectively.

Another input quantity for the tire force calculations is the normal force on each tire. According to the quasi-static longitudinal and lateral load transfers, F_{zi} is a function of the static vertical load, longitudinal acceleration a_x, lateral acceleration a_y, and roll stiffness. The normal forces acting on each tire are given by [6]:

$$F_{z1} = \frac{mgL_r}{2L} - \frac{m_s a_x \bar{z}_s}{2L} - k_f \frac{m_s a_y \bar{z}_s}{d},$$

$$F_{z2} = \frac{mgl_r}{2L} - \frac{m_s a_x \bar{z}_s}{2L} + k_f \frac{m_s a_y \bar{z}_s}{d},$$

$$F_{z3} = \frac{mgL_f}{2L} + \frac{m_s a_x \bar{z}_s}{2L} - k_r \frac{m_s a_y \bar{z}_s}{d},$$

$$F_{z4} = \frac{mgL_f}{2L} + \frac{m_s a_x \bar{z}_s}{2L} + k_r \frac{m_s a_y \bar{z}_s}{d}, \quad (10)$$

Where L is the wheelbase, d is the track width, and k_f, k_r are front and rear suspension roll stiffness, respectively.

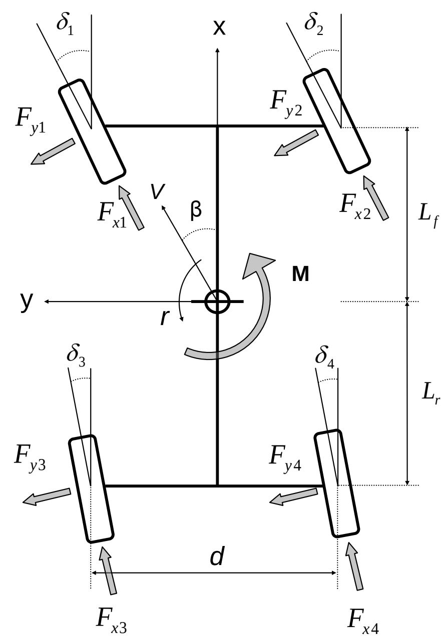

Figure 2. Schematic top view diagram of the vehicle dynamic model

The rotational dynamics of four wheels are given by (Figure 3)

$$I_w \dot{\omega}_i = -RF_{xi} + T_{bi}, \qquad (i = 1,...4) \quad (11)$$

Where R is the radius of each wheel, I_w is the moment of inertia of each wheel about its rotational axis, ω_i is the angular velocity, and T_{bi} is the applied torque at the centre of the ith wheel.

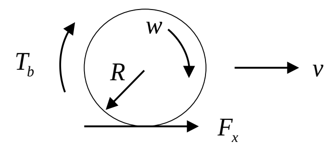

Figure 3. The rotational dynamics of each wheel

3. Controller Design

In this section we introduce the proposed fuzzy control system for vehicle system dynamics. The fuzzy logic is a linguistic control strategy that is usually based on the control engineer's knowledge. These controllers have the benefit of not requiring a mathematical model of the plant, while still being highly robust. Therefore the main advantages of fuzzy control systems are simplicity of their implementation and their robustness to disturbances and uncertainties of the system model. On the other hand, they can successfully control nonlinear systems. Due to complicated non-linear dynamics of vehicle system and unknown characteristics of tires forces, the exact modelling of the vehicle system is not always feasible, and these features cause difficulties in developing a classical controller system. Thus, in such conditions where the uncertainties are significant a fuzzy controller can be effective.

3.1. Reference generator

Here, the purpose of fuzzy control system is to make the vehicle's yaw rate and side slip angle follow their desired values. The desired values of yaw rate and side slip angles are analytically calculated on the basis of the driver's steering input, δ_f , and the vehicle forward speed, V, using a linear two degree of freedom vehicle model , as given by [7].

$$\frac{\beta_d}{\delta_f}(s) = G_B \frac{1+T_B s}{1+\frac{Q}{P}s+\frac{1}{P}s^2}, \tag{12}$$

$$\frac{r_d}{\delta_f}(s) = G_R \frac{1}{1+T_e s}, \tag{13}$$

Where

$$P = \frac{4L^2 C_f C_r}{m I_z V^2}\left(1+AV^2\right),$$

$$Q = \frac{2}{m V I_z}\left[m\left(L_f^2 C_f + L_r^2 C_r\right) + I_z\left(C_f + C_r\right)\right],$$

$$T_B = \frac{I_z V}{2 L L_r C_r}\frac{1}{1-(m/2L)\left(L_f / L_r C_r\right)V^2},$$

$$G_R = \frac{1}{1+AV^2}\frac{V}{L},$$

$$G_B = \frac{1-(m/2L)(L_f/L_rC_r)V^2}{1+AV^2}\frac{L_r}{L},$$

$$A = \frac{m}{2L^2}\frac{L_rC_r - L_fC_f}{C_fC_r},$$

Where C_f, C_r are combined cornering stiffness of front and rear tires.

3.2. Yaw-rate and side-slip angle control

Although to design a fuzzy controller, developing the mathematical model is not required, but a proper knowledge of plant system dynamics is unavoidable. In other words, understanding of relation between system variables and dynamics system behaviour by control engineers plays a key role in design of a fuzzy controller. Since, a 2DOF linear vehicle model, with constant speed is capable of demonestrating a number of important basic facts of vehicle dynamics behaviour, it is choosen in this paper to accomplish the task of control design. Using the schematic model of Figure 4 the basic equations of motion for this model are given as [7]:

$$I_z\dot{r} = M, \tag{14}$$

$$mV\dot{\beta} + mVr = F_y, \tag{15}$$

Where m denotes the total mass of the vehicle, I_z the yaw moment of inertia, and V the vehicle velocity. β and r denote the actual vehicle side slip angle and the yaw rate, respectively. M and F_y are sum of external forces and moments in lateral and yaw direction acting on the vehicle.

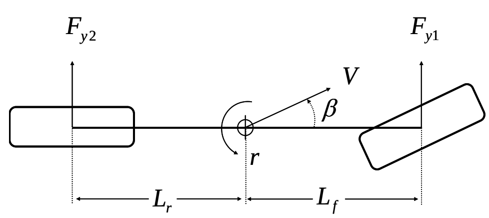

Figure 4. 2DOF linear vehicle schematic model

These equations show that, the side-slip and yaw rate dynamics are directly proportional to the lateral forces and total yaw moment acting on the vehicle, respectively. On the other hand, the dynamic of these two parameters is not coupled, *i.e.*,

$$\dot{r} \propto M, \tag{16}$$

$$\dot{\beta} \propto F_y, \tag{17}$$

Accordingly, two separate fuzzy controls are proposed, one for yaw rate control and the other for side slip angle control. The former uses the linguistic yaw rate error, $e_r = r - r_d$, and its integral, $\int e_r dt$, to get the yaw moment, M, and the later employs the linguistic side slip angle error, $e_\beta = \beta - \beta_d$, and its integral, $\int e_\beta dt$, to compute the lateral force, F_y. Where r and β are the actual yaw rate and side slip angle measured by the corresponding sensors on the vehicle and r_d and β_d are the desired yaw rate and side slip angle estimated by the reference model described in Section 3.1.

As mentioned before, one of the linguistic input signals considered is error integral. The advantage of this integration is twofold. It removes the steady state error, and also improves the robustness of the controller system by filtering out high frequency noises arising from the parameter estimation or data acquisition. Figure 5 shows the block diagram of the control structure.

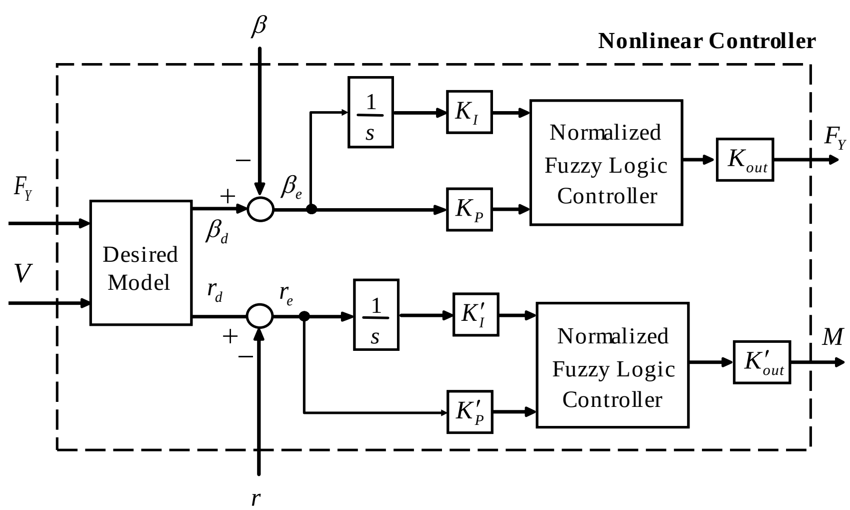

Figure 5. Schematic of control structure

As shown in Figure 6 and 7, we consider three membership functions for the input variables, *i.e.*, r_e and β_e, and five membership functions for the output variables, *i.e.*, M and F_y. These inputs and outputs of the fuzzy logic controller vary in the range (-1.0 to 1.0). Therefore, the input signals should be normalized and also the normalized control signals should be multiplied by the maximum allowable yaw moment and lateral forces values to get the actual results. Additional membership functions in the controller outputs are designed to give good system resolution for the rule table to determine the output signals.

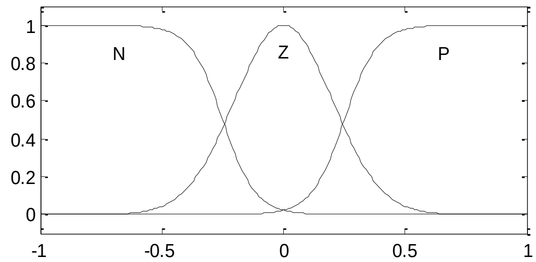

Figure 6. The membership function of error and integral of error (side slip angle and yaw rate)

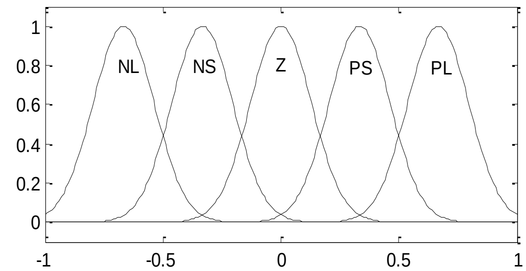

Figure 7. Membership function for the outputs (F_{fuzzy} and M_{fuzzy})

The fuzzy inference rules are given in Table 1. These elements are entered based on the expert knowledge of the designer. The input variables are fuzzified, and then according to the fuzzy computation the corresponding control rules are evaluated from the rule table. Finally, the control rules are defuzzified to derive the control signal. Figure 8 shows a three-dimensional graph representing the input/output surface of the controller. The following are the interpretation of the fuzzy labels used:

NL (Negative Large), NS (Negative Small), N (Negative), Z (Zero), PS (Positive Small), P (Positive), PL (Positive Large).

Table 1. Fuzzy rules for the Yaw-Rate and Side-Slip angle controllers

		Integral of Error		
		N	*Z*	*P*
Error	*N*	*NL*	*NL*	*NS*
	Z	*NS*	*Z*	*PS*
	P	*PS*	*PL*	*PL*

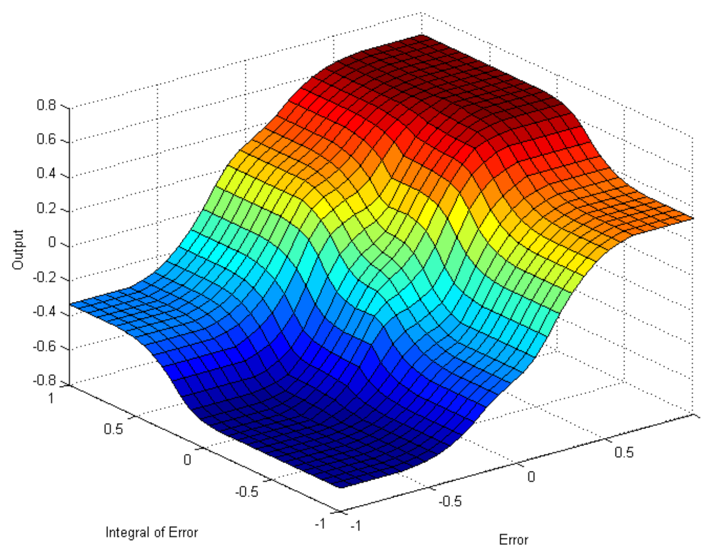

Figure 8. Normalized Input/output surface for the controller

3.3. Modification of the vehicle dynamics controller

Even though the error integral as an input control has some advantages, which was described in the previous section, the summation of all errors leads to reduction of the control sensitivity to new errors. Therefore, the intuitive motivation of this section is that past data are generated by past errors, and thus should be discounted when being used for the inference of the fuzzy logic controller. Exponential forgetting of data is a very useful technique in dealing with this issue.

In this section, first, the general formulation of this method with time-varying forgetting factor is described, and then a simplified form, employed in the simulation section is introduced.

The general formulation is as follows.

$$\int_{s_0}^{s_t} \exp\left(-\int_{s}^{t} \eta(n) dn \right) \; e_{\beta,r} ds, \tag{18}$$

In which $\eta(t) \geq 0$ is a design parameter called the time-varying forgetting factor and $e_{\beta,r}$ is the side slip or yaw rate error. The exponential term in the integral represents the weighting for the errors. In fact this method reduces the effect of old errors and increases the contribution of new errors, i.e. the errors near to and at the present time, by passing errors through a first order filter. Thus, at time t, instead of the integral of errors, the exponential forgetting of errors is used in the fuzzy logic controller. We can interpret this method by a moving window which the error summation inside of this window is fed to the controller. The window width is defined by the time-varying forgetting factor. This concept is depicted in Figure 9.

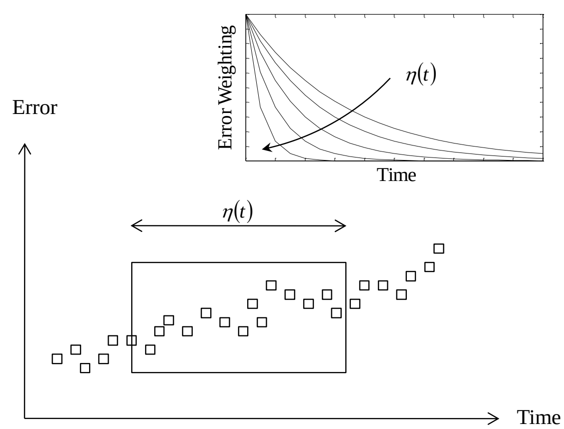

Figure 9. Data exponential forgetting technique with time-varying forgetting factor

A zero forgetting factor leads to reducing the window width, *i.e.*, vanishes the integration of the errors and thus turns it into the instant error fed to the controller. It implies that the disturbance and noise in the prediction error may lead to violent oscillations of the controller signals. On the other hand, choosing a large forgetting factor increases the window width and includes all past errors, resulting in the standard fuzzy controller method described in Section 3.2. It means that the noise and disturbance are filtered out and the system is highly robust. As it is mentioned earlier the main flaw of this case is that the summation of all errors leads to reduction of the control sensitivity to new errors. Therefore, a time-varying forgetting factor provides the opportunity to adapt the robustness of the controller according to the system operation condition (Figure 10). In other word, it is desirable to tune the forgetting factor variation so that errors forgetting are activated when low robustness is needed, and suspended when high robustness is intended. Thus, caution has to be exercised in choosing the forgetting factor.

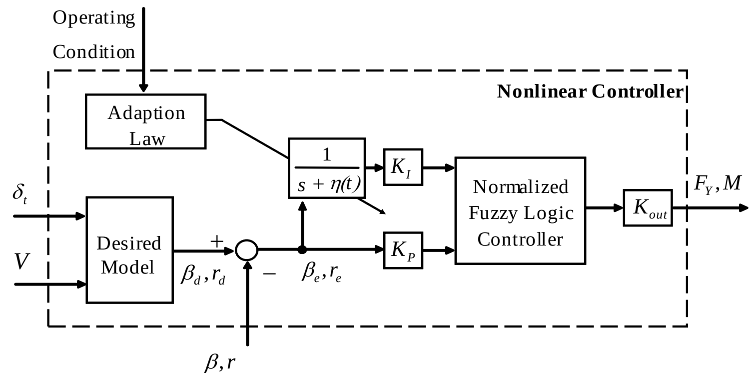

Figure 10. Control schematic structure-Adjustable robustness approach

The focus of this paper is to demonstrate the concept of incorporating exponential forgetting of data into fuzzy logic inference system, thus the adaptation or finding the optimum value of the forgetting factor is not of concern at this moment. Thus for the sake of simplicity, a constant value determined by trial and error is used for simulation purpose.

The simplified formulation is given by,

$$\int_{s_0}^{s_t} e^{-\eta(t-s)} e_{\beta,r} ds, \tag{19}$$

4. Actuatator Redundency Management

When discussing vehicle dynamics, often two output variables, such as the yaw rate and the sideslip angle of the vehicle body, must be controlled for desired responses. Theoretically two control inputs are needed to control two output variables to realize desirable responsiveness. Therefore, the proposed controller used in this paper is the integration of ASC and DYC subsystems.

There are different combinations of inputs which result in the total lateral force and yaw moment demanded. In other words, a unique set of eight unknown control inputs (longitudinal and lateral force for each corner of the vehicle), to provide integration of controller subsystems and satisfying two control outputs as a solution, cannot be easily determined. On the other hand, an inappropriate assigning wheel slips to the wheels reduces the chance of satisfying the control objectives, *i.e.*, tracking the desired yaw rate and side-slip angle, as a result of tire saturation limit. Thus, a distributing method adjusting the magnitude of the tire forces according to their capacity has significant importance. In this regard, AODF method, proposed by the authors in [8] is employed in the paper to find an optimal distribution of the control effort among a redundant set of actuators.

4.1. Objective function subject to inequality constraints

Using DYC in all ranges of the tire friction circle may lead to uncomfortable driving conditions, because of a reduction in the vehicle's total speed. However, AODF activates DYC only when it is required. This is the main advantage of this method.

The objective function consists of the linear weighting of two terms. The first term is the summation of the squared normalized resultant tire forces (the workload of tires), generated at each corner of the vehicle, and the second term is the summation of squared normalized longitudinal tire forces relative to tire capacities. The first term minimizes the workload of tires and the second part is intended to adjust the contribution of the DYC subsystem to an integrated vehicle control system according to

$$f = \sum_{i=1}^{4} \left\{ A_i \frac{\overbrace{\left(F_{xi}^{\ 2} + F_{yi}^{\ 2}\right)}^{F_i^2}}{\left(\mu_i F_{zi}\right)^2} + B_i \left(\frac{F_{xi}}{\mu_i F_{zi}} \right)^2 \right\} = \sum_{i=1}^{4} C_i \left(D_i F_{xi}^{\ 2} + F_{yi}^{\ 2} \right), \tag{20}$$

Where

$$C_i = \frac{A_i}{\left(\mu_i F_{zi}\right)^2},$$

$$D_i = 1 + \frac{B_i}{A_i},$$

In which i denotes the wheel number, F_{xi} and F_{yi} are the longitudinal and lateral tire forces respectively in the fixed coordinate system of the vehicle body, F_{zi} is the vertical load, μ_i is the tire friction coefficient, and A_i and B_i are the weighting coefficients. If only the first term of the objective function given by (20) (*i.e.*, $B_i = 0$) is considered, an optimal solution is obtained, which is referred to as the ODF method [8-9]. In that case, DYC will be active in all driving conditions, which is not desirable.

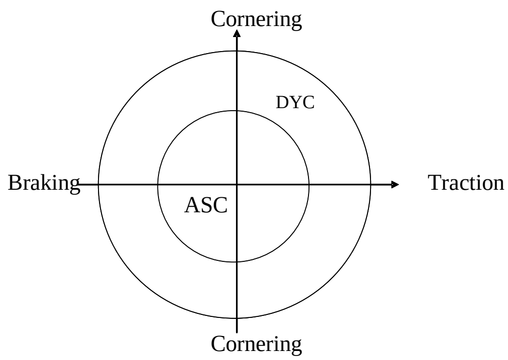

Figure 11. Vehicle coordinate system

However, the weighting coefficients can be updated by means of an adaptation mechanism to overcome the shortcomings of the ODF. In this approach the contribution of each subsystem in the integrated vehicle dynamics control scheme is adjusted according to the friction circle notion, which is a consequence of the inherent saturation property of tire forces. Thus in normal situations, where the lateral tire force is approximately proportional to the corresponding side-slip angle, by increasing the contribution of the DYC subsystem in the cost function, the ASC system becomes more effective in controlling the vehicle, and the handling performance of the vehicle will be enhanced. When the vehicle approaches the handling limit and the lateral tire force is close to or even beyond the saturation point, by reducing the contribution of the DYC subsystem to the cost function, this subsystem is used to stabilize the vehicle dynamics. This is the approach of AODF mentioned above.

The weighting coefficient B_i can be described as a function of the workload of the tires for a constant coefficient A_i ; this is given by

$$B_i(u) = \begin{cases} 1, & u < a \\ \dfrac{b-u}{b-a}, & a \le u \le b \\ 0, & u > b \end{cases} \tag{21}$$

Where

$$u = \max(F_{Nj}), \qquad j = 1,\ldots,4$$

In (21), F_{Nj} is the normalized resultant force of the ith tire. The saturation range of each tire is determined with a and b. The values of weighting coefficient B_i determine how much each longitudinal tire forces participate in the objective function. Thus, as shown in Figure 12, when the tire lateral force approaches its saturation value, the participation of the DYC becomes large. Figure 11 depicts the activation area of each subsystem.

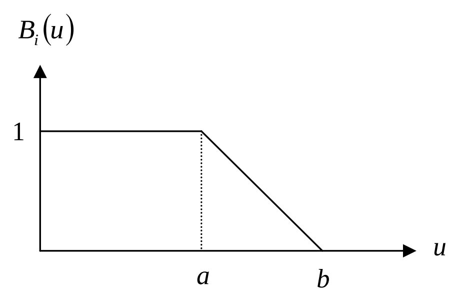

Figure 12. Functionality of weighting coefficient B_i versus normalized resultant tire forces

The sum of the lateral forces and yaw moments acting on the vehicle by each tire force should be equal to the required total lateral force Fy and yaw moment M computed on the basis of fuzzy control theory. Therefore, all the variables in the objective function must satisfy the two equality constraints

$$F_y = \sum_{i=1}^{4} F_{yi}, \tag{22}$$

$$M = \sum_{i=1}^{2}\left(L_f F_{yi} - L_r F_{y(i+2)}\right) + \\ + \frac{d}{2}\sum_{i=1}^{2}\left(F_{x(2i)} - F_{x(2i-1)}\right), \tag{23}$$

and the four inequality constraints

$$g_i = F_{xi} \le 0, \qquad i = 1,\ldots,4 \tag{24}$$

The Inequality Constraints (24) ensure the assumption of yaw moment generation, using only braking forces. This consideration restricts the algorithm to an available control system that is widely used in vehicles, *i.e.*, vehicles equipped with ABS systems, where just braking torque is applied and the driveline torque is not feasible. Considering only braking torques causes inequality constraints in the optimization problem and makes it more difficult.

Using two equality constraints, any two of the variables can be eliminated in the objective function. Then, the optimization problem can be replaced with minimization of a new objective function, with six variables that are only subjected to four inequality constraints.

Since the objective function and constraint equations are convex (in terms of 'convex programming problems class' definitions), the Kuhn– Tucker conditions are necessary and sufficient for a global minimum [10].

Substituting (22)-(23) and the set of Inequality Constraints (24) into the Kuhn–Tucker conditions with proper simplification, two set of linear equations can be reached:

For the condition $\lambda_1 \ge 0$,

$$\underset{5\times5}{[\mathbf{A}^{(1)}]}\ \underset{5\times1}{\mathbf{x}^{(1)}} = \underset{5\times1}{\mathbf{b}^{(1)}}, \tag{25}$$

Where

$$\mathbf{x}^{(1)} = [\lambda_1 \quad F_{x2} \quad F_{y1} \quad F_{y2} \quad F_{y3}]^T,$$

And for the condition $\lambda_2 \ge 0$,

$$\underset{5\times5}{[\mathbf{A}^{(2)}]}\ \underset{5\times1}{\mathbf{x}^{(2)}} = \underset{5\times1}{\mathbf{b}^{(2)}}, \tag{26}$$

Where

$$\mathbf{x}^{(2)} = [\lambda_2 \quad F_{x1} \quad F_{y1} \quad F_{y2} \quad F_{y3}]^T,$$

The elements of the matrices $[\mathbf{A}^{(n)}]$ and of $\mathbf{b}^{(n)}$, $n = 1,2$, are functions of A_i, B_i, μ_i, F_{zi}, d, L_f, L_r, F_y and M. In addition, the weighting coefficients B_i are updated with an adaptation mechanism (Figure 12), as stated before. Also, the vertical loads F_{zi} are estimated by considering lateral and longitudinal load transfers.

According to the signs of λ_1 and λ_1, one of the linear systems of (25)-(26) is solved at each time step to find the distribution of braking and lateral tire forces in optimal form. Then, using the inverse of the tire model, the active steering angle can be determined as

$$\delta_i \approx \begin{cases} \beta + \dfrac{L_f r}{v_x} - \alpha_i, & i = 1,2 \\ \beta - \dfrac{L_r r}{v_x} - \alpha_i, & i = 3,4 \end{cases} \tag{27}$$

where α_i is the side-slip angle of the ith tire and is obtained as

$$\alpha_i \approx -\frac{F_{yi}}{C_i}, \qquad i = 1,\ldots,4 \tag{28}$$

In which C_i denotes the cornering stiffness of the *i*th tire. In addition, it is assumed that the brake torque required for the DYC system is equal to the wheel radius multiplied by the brake force and thus the wheel dynamics is neglected here.

Table 2. Vehicle parameters (Small passenger car)

Parameter	Value	Unit
m	1070	kg
m_s	900	kg
I_{zz}	2100	kg·m^2
I_{xx}	500	kg·m^2
I_{xz}	47.5	kg·m^2
I_w	0.9	kg·m^2
R	0.2836	m
h	0.6	m
L_f	1.1	m
L_r	1.3	m
d	1.4	m
$C_{f,r}$	90624	N/rad
$k_ø$	65590	N·m/rad
$c_ø$	2100	N·m·s/rad

5. Simulation Results

In order to compare vehicle responses, employing the introduced fuzzy logic controller and the modified form of the controller, simulations are performed. A 9DOF nonlinear vehicle model comprising Dugoff’s tire model is used for the simulation purpose. The vehicle parameters are those used in [8].

To investigate the differences between standard fuzzy control and the modified fuzzy control schemes we consider a severe lane change maneuver. The vehicle runs with an initial velocity of 120 km/h for 0.5 s followed by a 60 degrees sine wave steering wheel command, with the frequency of 0.5 Hz (Figure 13), on dry road surface (i.e. $\mu_i = 1$), also steering mechanism gear ratio is equal to 18. The maneuver is performed in absence of braking command.

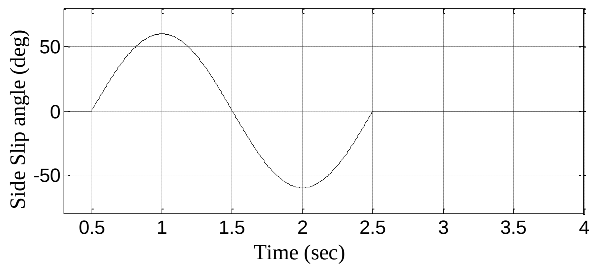

Figure 13. Steering wheel command

Comparison of the vehicle sideslip angle and yaw rate implementing integrated control using standard fuzzy control and modified form, are shown in Figure 14 through 16. The same results for the vehicle without controller are also provided. In all figures the dotted lines stand for the target response. As depicted, vehicle without controller cannot track the desired yaw rate and side slip angle.

On the other hand, both integrated vehicle control systems, i.e., standard fuzzy control and the modified fuzzy control can stabilize the vehicle motion and also make its response characteristics close to those achieved by the model reference.

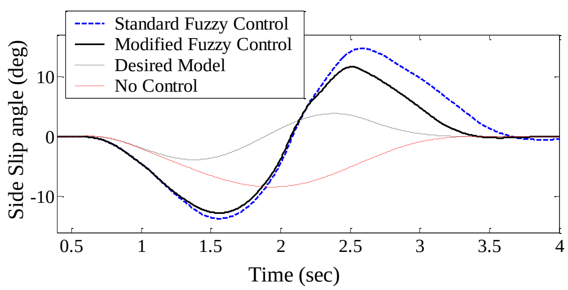

Figure 14. Side slip angle response

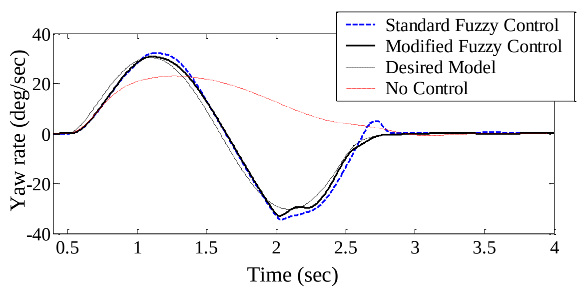

Figure 15. Yaw rate response

In Figure14, the second half of vehicle manoeuvre using modified controller demonstrates much better tracking performance compared to standard fuzzy mode control. This is due to the exponential forgetting of data employed in the modified form of the controller in order to decrease the effect of old errors and increase the effects of new errors. This improvement in handling and stability is more evident in the side slip angle, and thus lead to faster response in lateral acceleration as shown in Figure 17; the smaller sideslip angle, the more stable the vehicle.

The vehicle trajectories together with their attitudes are compared in Figure 18, where the integrated vehicle control with the standard fuzzy control and modified fuzzy control can successfully perform the lane change maneuver, whereas the system without controller cannot perform the task.

For comprehensive investigation, this maneuver has been done in three other different speeds and results are summarized in Figure 19 by the ratios of trough to peak value of the side slip response to the same value of the yaw rate response. This diagram can be used as the performance stability index, where a vehicle with a closer locus to the desired model is assumed to be more stable. As a result, from this figure higher performance of the proposed controller is obvious.

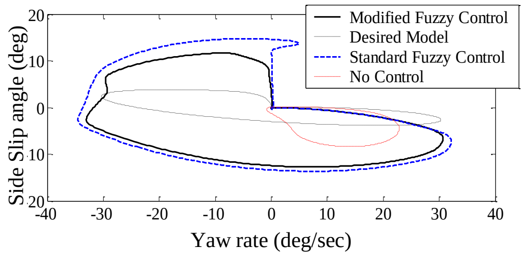

Figure 16. Phase plane response

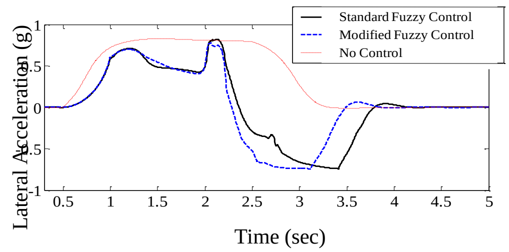

Figure 17. Lateral acceleration response

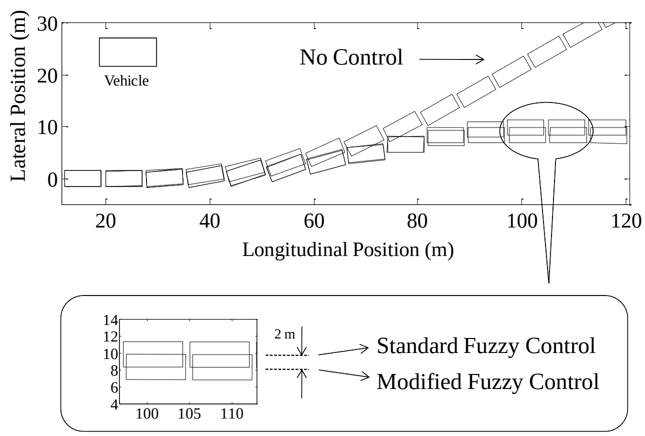

Figure 18. Vehicle trajectory

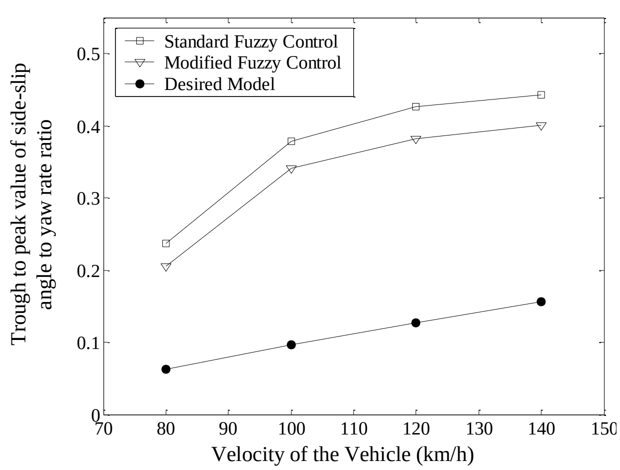

Figure 19. Side slip angle-yaw rate ratio versus vehicle speed

6. Conclusion

In this study, by incorporating data exponential forgetting technique into the standard fuzzy controller a modified fuzzy controller for integrated vehicle dynamics control is proposed.

Performance of this integrated vehicle dynamics controller and its modified form of the control scheme on stabilizing a passenger car during severe lane change maneuvers has been examined through digital simulations.

Results show satisfactory performance of the proposed control systems and indicate that the modified fuzzy controller manages integrated vehicle control systems more effectively compared to standard fuzzy logic controller.

Finding a time-varying forgetting factor as well as an optimum value of this factor according to the system's operating conditions is the topic of the ongoing research.

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