

Design of an Integrated Yaw-Roll Moment and Active Front Steering Controller using Fuzzy Logic Control

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ABSTRACT

This paper presents an advanced control system, which integrates three fuzzy logic controllers namely; Direct Yaw-moment Control (DYC), Active Roll-moment Control (ARC) and Active Front Steering (AFS) to enhance vehicle cornering and overturning stability. Based on a well-developed and validated fourteen degree of freedom (DOF) full vehicle model with non-linear tire characteristics, a reference 3-DOF yaw-roll plane vehicle model is introduced to control yaw rate, sideslip angle, and roll angle of the vehicle body. The control actions of both direct yaw and active roll moments are performed by generating differential braking moments across the front wheels, while the control action of the active steering is performed by modifying the steering wheel angle. Different standard cornering tests are conducted in MATLAB / Simulink environment such as J-turn, fishhook and lane change maneuvers. The simulation results show a substantial improvement in vehicle stability by employing the proposed integrated controller compared to that of the individual controllers and the conventional vehicle without control.

KEY WORDS: vehicle dynamics, integrated chassis control, fuzzy logic control

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INTRODUCTION

Over the years, different control systems have been proposed to improve the performance of various vehicle subsystems. It has been concluded that, regulating tire forces in longitudinal, lateral and vertical directions is a milestone in modern automotive control systems such as Acceleration Slip Regulation (ASR) [1, 2], Electronic Stability Program (ESP) [3, 4], and Anti-lock Braking System (ABS) [5, 6]. These control systems are widely implemented and have gained a substantial performance improvement during emergency braking, high speed cornering and driving on slippery roads with offering the possibility of differential distribution of braking forces without exceeding its permissible limits. Additionally, Active Front Steering (AFS) has been proposed to regulate tire slip angles and therefore the lateral tire forces in order to improve both vehicle handling and yaw stability [7].

Recently, the arising challenge is to integrate different control systems for simultaneous improvement of traction performance, braking safety, cornering stability and passenger comfort [8, 9, 10, 11, 12]. Numerous advanced control theories have been proposed to actively integrate vehicle chassis control systems such as Lyapunov based control technique [13], Theory of Sliding Mode Control (SMC) [12], Gain-scheduled Linear Parameter Varying control (LPV) [14]

and Optimal Control Theory [15]. Additionally, many researchers have implemented Fuzzy Logic Control (FLC) to integrate both AFS and DYC in order to improve vehicle lateral dynamics and rollover stability [1, 2, 3, 16]. FLC is a knowledge-based control method, which imitates the human skills for controlling systems. It is widely applied because of the convenience it offers in the control of dynamic systems with high nonlinearities.

Based on initial works carried out by the authors [17–18], this paper integrates different advanced automotive control systems such as AFS, ARC and DYC in order to enhance both the vehicle cornering and over-turning stability. The implementation of Fuzzy Logic Control theory offers the advantage of incorporating highly non-linear 14-DOF full vehicle simulation environment by which vehicle subsystems are accurately represented. The paper is organized in five main sections as follow: Second section describes the mathematical formulation of the vehicle model adopted for control system design as well as its validation results. Both the architecture and design procedures of the proposed integrated chassis control system are outlined in third section. Simulation results during different high-speed standard cornering test maneuvers are discussed in fourth section, while the main findings and conclusions are drawn in the last section.

FULL VEHICLE NONLINEAR MODEL

The mathematical formulation of the vehicle sub-systems is based on a 14-DOF full vehicle model, which is developed by Sharaf [19]. This model is generic and appropriate for simulating a wide variety of vehicle dynamics in longitudinal, lateral and vertical directions as shown in Figures 1-a and 1-b. The vehicle is divided into five masses including its body, which is located above four unsprung masses via conventional spring and damper system. Additionally, all aspects of rotational inertial dynamics, friction, damping and stiffness of a 4x4 drive train are included. The model is relatively complex with offering the advantage of writing an in-house code of a specific vehicle subsystem. Furthermore, particular emphasis is considered for the modularity and user-friendly interface.

Sprung Mass Dynamics

The vehicle body has 6-DOF, which includes three translations (x, y, z) and three angular rotations (ϕ, θ, ψ). Based on Newton-Euler formulation [20], the equations of motion of the vehicle sprung mass (body) are formulated as follow:

$$\begin{aligned} \Sigma F_x = & m_b \cdot (\ddot{x} - \dot{y} \cdot \dot{\psi} + \dot{z} \cdot \dot{\theta}) - \\ & m_b \cdot \left[x_G \cdot (\dot{\theta}^2 + \dot{\psi}^2) - y_G \cdot (\dot{\phi} \cdot \dot{\theta} - \ddot{\psi}) - z_G \cdot (\dot{\phi} \cdot \dot{\psi} + \ddot{\theta}) \right] \end{aligned} \quad (1)$$

$$\begin{aligned} \Sigma F_y = & m_b \cdot (\ddot{y} - \dot{z} \cdot \dot{\phi} + \dot{x} \cdot \dot{\psi}) - \\ & m_b \cdot \left[y_G \cdot (\dot{\psi}^2 + \dot{\phi}^2) - z_G \cdot (\dot{\theta} \cdot \dot{\psi} - \ddot{\phi}) - x_G \cdot (\dot{\phi} \cdot \dot{\theta} + \ddot{\psi}) \right] \end{aligned} \quad (2)$$

$$\begin{aligned} \Sigma F_z = & m_b \cdot (\ddot{z} - \dot{x} \cdot \dot{\theta} + \dot{y} \cdot \dot{\phi}) - \\ & m_b \cdot \left[z_G \cdot (\dot{\phi}^2 + \dot{\theta}^2) - x_G \cdot (\dot{\phi} \cdot \dot{\psi} - \ddot{\theta}) - y_G \cdot (\dot{\theta} \cdot \dot{\psi} + \ddot{\phi}) \right] \end{aligned} \quad (3)$$

$$\begin{aligned} \Sigma M_x = & I_{xx} \cdot \ddot{\phi} - (I_{yy} - I_{zz}) \cdot \dot{\theta} \cdot \dot{\psi} + I_{yz} \cdot (\dot{\psi}^2 - \dot{\theta}^2) - \\ & I_{zx} \cdot (\dot{\phi} \cdot \dot{\theta} + \ddot{\psi}) + I_{xy} \cdot (\dot{\phi} \cdot \dot{\psi} - \ddot{\theta}) + \\ & m_b \cdot y_G \cdot (\ddot{z} - \dot{x} \cdot \dot{\theta} + \dot{y} \cdot \dot{\phi}) - m_b \cdot z_G \cdot (\ddot{y} - \dot{z} \cdot \dot{\phi} + \dot{x} \cdot \dot{\psi}) \end{aligned} \quad (4)$$

$$\begin{aligned} \Sigma M_y = & I_{yy} \cdot \ddot{\theta} - (I_{zz} - I_{xx}) \cdot \dot{\phi} \cdot \dot{\psi} + \\ & I_{xz} \cdot (\dot{\phi}^2 - \dot{\psi}^2) - I_{xy} \cdot (\dot{\theta} \cdot \dot{\psi} + \ddot{\phi}) + \\ & I_{yz} \cdot (\dot{\theta} \cdot \dot{\phi} - \ddot{\psi}) + m_b \cdot z_G \cdot (\ddot{x} - \dot{y} \cdot \dot{\psi} + \dot{z} \cdot \dot{\theta}) \\ & - m_b \cdot x_G \cdot (\ddot{z} - \dot{x} \cdot \dot{\theta} + \dot{y} \cdot \dot{\phi}) \end{aligned} \quad (5)$$

$$\begin{aligned} \Sigma M_z = & I_{zz} \cdot \ddot{\psi} - (I_{xx} - I_{yy}) \cdot \dot{\phi} \cdot \dot{\theta} + \\ & I_{xy} \cdot (\dot{\theta}^2 - \dot{\phi}^2) - I_{yz} \cdot (\dot{\psi} \cdot \dot{\phi} + \ddot{\theta}) + I_{zx} \cdot (\dot{\psi} \cdot \dot{\theta} - \ddot{\phi}) \\ & + m_b \cdot x_G \cdot (\ddot{y} - \dot{z} \cdot \dot{\phi} + \dot{x} \cdot \dot{\psi}) - \\ & m_b \cdot y_G \cdot (\ddot{x} - \dot{y} \cdot \dot{\psi} + \dot{z} \cdot \dot{\theta}) \end{aligned} \quad (6)$$

The vehicle body translational dynamics are calculated due to the effect of net external forces ($\Sigma F_x, \Sigma F_y, \Sigma F_z$) in the longitudinal lateral and vertical directions which arising from tire forces, air and grade resistances. The effects of total external moments about vehicle body coordinates ($\Sigma M_x, \Sigma M_y, \Sigma M_z$) are also considered for the estimation of vehicle body angular dynamics. According to SAE Recommended Practice J670e, six coordinates systems are considered to observe the kinematics of the vehicle motion namely, earth-fixed axis system, vehicle axis system and wheel axis system at each wheel center.

Unsprung Mass Dynamics

The unsprung mass dynamics is represented by 2-DOF for each wheel, one for the vertical movement and the other for wheel spin as shown in Figures 1-c and 1-d as provided at the end of this paper. The equations of motion of the unsprung mass dynamics are formulated as follows, for wheel (i) from 1 to 4:

$$\begin{aligned} m_{w_i} \cdot \ddot{z}_{w_i} = & m_{w_i} \cdot g + \\ & \underbrace{C_i \cdot (\dot{z}_{b_i} - \dot{z}_{w_i}) + K_i \cdot (z_{b_i} - z_{w_i})}_{\text{Suspension Force (F}_{si})} + F_{z_i}(z) \end{aligned} \quad (7)$$

$$I_{w_i} \cdot \dot{\omega}_i = M_{Di} - M_{Ui} - M_{Bi} - (F_{xi} \cdot r_{di}) \quad (8)$$

Tire Forces and Moments

Magic Formula MF tire-model is employed to estimate tire forces and moments during different maneuvers such as traction, braking and cornering as follows [21]:

$$y(x) = D \sin \left[C \arctan \left(Bx - E (Bx - \arctan Bx) \right) \right] \quad (9)$$

Where ($y(x)$) denotes tire longitudinal force, lateral force, or self-aligning torque which are calculated based on tire longitudinal slip or skid (x) and cornering slip angle (α). The magic formula's coefficients (B, C, D, E) are obtained based on the typical measured forces and moment's data. For each tire, the vertical force is calculated from the static weight and the dynamic load due to forward and lateral accelerations (a_x, a_y) as follow:

$$\left. \begin{aligned} F_{z1} &= \frac{b}{2L} m_t \cdot g - \frac{h}{2L} m_t \cdot a_x - \frac{bh}{Lt_{rf}} m_t \cdot a_y \\ F_{z2} &= \frac{b}{2L} m_t \cdot g - \frac{h}{2L} m_t \cdot a_x + \frac{bh}{Lt_{rf}} m_t \cdot a_y \\ F_{z3} &= \frac{a}{2L} m_t \cdot g + \frac{h}{2L} m_t \cdot a_x + \frac{ah}{Lt_{rr}} m_t \cdot a_y \\ F_{z4} &= \frac{a}{2L} m_t \cdot g + \frac{h}{2L} m_t \cdot a_x - \frac{ah}{Lt_{rr}} m_t \cdot a_y \end{aligned} \right\} \quad (10)$$

In order to ensure an accurate representation of the derived model, the simulation output of the 14-DOF vehicle model is compared to that of well-known commercial packages such as highly complex multi body dynamics models created in Adams-Car and medium sophisticated models created in CarSim. For this purpose, identical technical parameters of a given vehicle (see [Appendix](#)) are assumed during similar driving conditions of standard double lane change maneuver. Consequently, the main cornering stability indicators such as lateral acceleration, yaw rate and roll angle of the vehicle body for all simulation environments are then compared and found to be in good agreement as shown in [Figure 2](#) as provided at the end of this paper.

INTEGRATED CHASSIS CONTROLLER DESIGN

In order to control both cornering and overturning stability of the vehicle, a reference linear model with 3-DOF is adopted by considering the lateral, yaw and roll dynamics as given by Equations (11–13). The reference model estimates the steady-state desired yaw rate (r_{des}), desired sideslip angle (β_{des}) and desired roll angle (ϕ_{des}) relative to the change of driver's steering angle (δ) and vehicle forward velocity (U) as indicated by Equations (14–16) [22]:

$$\begin{aligned} m_t \cdot \underbrace{(\ddot{y} + \dot{x} \cdot \dot{\psi})}_{\text{Lateral Acceleration } a_y} &= C_{\alpha_f} \underbrace{\left(\delta - \frac{\dot{y} + a\dot{\psi}}{\dot{x}} \right)}_{\substack{\text{Front Slip Angle } \alpha_f \\ \text{Front Cornering Force } F_{yf}}} \\ &+ C_{\alpha_r} \underbrace{\left(-\frac{\dot{y} - b\dot{\psi}}{\dot{x}} \right)}_{\substack{\text{Rear Slip Angle } \alpha_r \\ \text{Rear Cornering Force } F_{yr}}} \end{aligned} \quad (11)$$

$$\begin{aligned} I_{zz} \cdot \ddot{\psi} &= a \cdot C_{\alpha_f} \underbrace{\left(\delta - \frac{\dot{y} + a\dot{\psi}}{\dot{x}} \right)}_{\text{Front Cornering Force } F_{yf}} \\ &- b \cdot C_{\alpha_r} \underbrace{\left(-\frac{\dot{y} - b\dot{\psi}}{\dot{x}} \right)}_{\text{Rear Cornering Force } F_{yr}} \end{aligned} \quad (12)$$

$$(I_{xx} + m_b \cdot h^2) \cdot \ddot{\phi} = \underbrace{(-K_{\phi} + m_b \cdot g \cdot h_1)}_{\text{Total Resisting Roll Moments } M_x} \cdot \phi - C_{\phi} \cdot \dot{\phi} \quad (13)$$

$$r_{des} = \frac{U}{L + \underbrace{\left(\frac{m_t \cdot (b \cdot c_{\alpha_r} - a \cdot c_{\alpha_f})}{2 \cdot c_{\alpha_f} \cdot c_{\alpha_r} \cdot L} \right)}_{\text{understeer coefficient / g}}} \cdot \delta \quad (14)$$

$$\beta_{des} = \frac{b - \left(\frac{a \cdot m_t}{2 \cdot c_{\alpha_r} \cdot L} \right) \cdot U^2}{L + \underbrace{\left(\frac{m_t \cdot (b \cdot c_{\alpha_r} - a \cdot c_{\alpha_f})}{2 \cdot c_{\alpha_f} \cdot c_{\alpha_r} \cdot L} \right)}_{\text{understeer coefficient / g}}} \cdot \delta \quad (15)$$

$$\phi_{des} = 0 \quad (16)$$

The developed controller constantly monitors the output of the 14-DOF vehicle non-linear model in particular vehicle yaw rate (r), roll angle (ϕ) and side slip angle (β) as illustrated in [Figure 3](#) as provided at the end of this paper. Consequently, the instantaneous yaw rate error ($e(r)$), roll angle error ($e(\phi)$) and side slip angle error ($e(\beta)$) are calculated as follows:

$$\left. \begin{aligned} \text{Yaw Rate Error :} \quad e(r) &= r - r_{des} \\ \text{Roll Angle Error :} \quad e(\phi) &= \phi - \phi_{des} \\ \text{Side Slip Angle Error :} \quad e(\beta) &= \beta - \beta_{des} \end{aligned} \right\} \quad (17)$$

The integrated vehicle chassis closed loop control system is designed to eliminate the aforementioned gains errors. The control actions are performed through the accurate calculations of the direct yaw moment (M_{yaw}), the active roll moment (M_{ϕ}) and the corrective steering wheel angle (δ_{corr}) as follows:

$$\text{Direct Yaw Moment :} \quad M_{yaw} = \frac{M_{Bl}}{r_{di}} \cdot \frac{t_{rf}}{2} \quad (18)$$

$$\text{Active Roll Moment : } M_{\phi} = M_{\phi ff} + M_{\phi fb} \quad (19)$$

$$\text{Corrective Steering Angle : } \delta_{corr} = \delta - \delta_f \quad (20)$$

Direct Yaw-Moment Controller (DYC)

It is widely known that, stimulating the vehicle body yaw rate by altering the individual wheel torque balance is usually addressed as 'Direct Yaw-moment Control'. In this work, direct yaw moment is created due to the variation of the braking torque between the vehicle front left and right wheels. Sign of control moment specifies which wheel will be employing brakes to generate the desired yaw control. Fuzzy Logic Control (FLC) is used to estimate the direct yaw moment (M_{yaw}) based on both the yaw rate error ($e(r)$) and side slip angle error ($e(\beta)$).

Five membership functions are selected for each input to FLC (Figures 4-a, 4-b) as provided at the end of this paper including two trapezoidal and three triangle membership functions with the following variables: high negative (HN), low negative (LN), zero (ZO), low positive (LP), high positive (HP). The control output (direct yaw moment) is obtained with a scaling factor by considering eleven membership functions including two trapezoidal and nine triangle membership functions as illustrated in Figure 4-c. The eleven variables for the DYC are; N5, N4, N3, N2, N1, ZO, P1, P2, P3, P4, and P5. The rule base of the DYC FLC is given in Table 1.

Table 1. Rule base for DYC and AFS fuzzy logic controllers

		Side Slip Angle Error				
		HN	LN	ZO	LP	HP
Yaw Rate Error	HN	N1	N1	ZO	P1	P1
	LN	N2	N2	ZO	P2	P2
	ZO	N3	N3	ZO	P3	P3
	LP	N4	N4	ZO	P4	P4
	HP	N5	N5	ZO	P5	P5

Active Roll-Moment Controller (ARC)

The proposed active roll control strategy consists of feedforward and feedback FLC as presented in Figure 3 at the end of this paper. For the feedforward fuzzy logic controller, the inputs are the driver steering wheel angle (δ) and vehicle longitudinal velocity (U), while the output is feedforward counter roll moment ($M_{\phi ff}$).

For the feedback FLC, both the roll angle error ($e(\phi)$) and roll angle error rate ($\dot{e}(\phi)$) are the inputs, while the feedback counter roll moment ($M_{\phi fb}$) is the output. The resultant active roll moment (M_{ϕ}) is the summation of the both feedforward ($M_{\phi ff}$) and feedback ($M_{\phi fb}$) moments. Gaussian membership functions are chosen to represent the inputs and output variables for both controllers due to their smooth mapping property for control. Feedforward FLC considers seven Gaussian memberships for each input of steering angle (δ) (Figures

5-a) and feedforward moment ($M_{\phi ff}$) (Figures 5-c) with symbols: high negative (HN), medium negative (MN), low negative (LN), zero (ZO), low positive (LP), medium positive (MP), and high positive (HP). Additionally, five Gaussian memberships' functions are considered for forward speed (U) (Figures 5-b) with symbols (x_1, x_2, x_3, x_4, x_5) as provided at the end of this paper. The rule base between the inputs and the output is tabulated in Table 2.

Table 2. Rule base for feedforward fuzzy logic controller

		Steering Wheel Angle						
		HN	MN	LN	ZO	LP	MP	HP
Forward Speed	X ₁	ZO	ZO	ZO	ZO	ZO	ZO	ZO
	X ₂	MN	MN	MN	ZO	MP	MP	MP
	X ₃	LN	MN	MN	ZO	MP	MP	HP
	X ₄	LN	LN	LN	ZO	HP	HP	HP
	X ₅	LN	LN	LN	ZO	HP	HP	HP

Table 3. Rule base for feedback fuzzy logic controller

		Roll Error Rate				
		MN	LN	ZO	LP	MP
Roll Error	MN	MN	MN	MN	MP	MP
	LN	MN	LN	LN	LP	MP
	ZO	MN	LN	ZO	LP	MP
	LP	MN	LN	LP	LP	MP
	MP	MN	MN	MP	MP	MP

Active Front Steering (AFS) Controller

In Active Front Steering (AFS), the actual steering wheel angle (δ_p) is the difference between the input steering angle by the driver (δ) and the corrective steer angle generated by the fuzzy controller (δ_{corr}). AFS fuzzy controller computes this corrective steer angle with a scaling factor based on yaw rate error ($e(r)$) and driver steering angle (δ). Accordingly, five membership functions are set for each input (Figures 7-a, 7-b) including two trapezoidal and three triangle membership functions with symbols HN, LN, ZO, LP and HP. On the other hand, eleven membership functions are selected to represent the corrective steering angle as an output of the controller (Figures 7-c) including two trapezoidal, and nine triangle membership functions with symbols N5, N4, N3, N2, N1, ZO, P1, P2, P3, P4, and P5 as provided at the end of this paper. The universe of discourse for the inputs is chosen based on their operating range. The rule base of the AFS Fuzzy controller is similar to DYC as given previously in Table 1.

Anti-lock Braking System (ABS) controller

ABS is introduced as a supervisory controller to regulate the braking moment at each wheel which is primarily resulting from the direct yaw and/or roll moment to avoid excessive wheel skidding. For this purpose, a simple bang-bang (on-off) controller is successfully employed to regulate the braking torque and therefore the wheel skid