

A New Intelligent Strategy to Integrated Control of AFS/DYC Based on Fuzzy Logic

R. Karbalaei, A. Ghaffari, R. Kazemi, and S. H. Tabatabaei

Abstract—An integrated vehicle dynamics control system is developed in this paper by a combination of active front steering (AFS) and direct yaw-moment control (DYC) based on fuzzy logic control. The control system has a hierarchical structure consisting of two layers. A fuzzy logic controller is used in the upper layer (yaw rate controller) to keep the yaw rate in its desired value. The yaw rate error and its rate of change are applied to the upper controlling layer as inputs, where the direct yaw moment control signal and the steering angle correction of the front wheels are the outputs. In the lower layer (fuzzy integrator), a fuzzy logic controller is designed based on the working region of the lateral tire forces. Depending on the directions of the lateral forces at the front wheels, a switching function is activated to adjust the scaling factor of the fuzzy logic controller. Using a nonlinear seven degrees of freedom vehicle model, the simulation results illustrate considerable improvements which are achieved in vehicle handling through the integrated AFS/DYC control system in comparison with the individual AFS or DYC controllers.

Keywords—Intelligent strategy, integrated control, fuzzy logic, AFS/DYC.

I. INTRODUCTION

A trend in modern vehicles is the application of active safety systems to improve vehicle handling, stability & comfort. In order to maintain vehicles safe handling characteristics, several active safety systems such as active braking, active steering and active suspension are able to improve the lateral dynamics of a vehicle. An active suspension system by wheel load controlling system can be used to improve the lateral dynamics. Also, by controlling the steering angles of the wheels (both front and rear wheels), an active steering system has great influence on the lateral dynamic behavior of a vehicle. Ultimately, an active braking system like Direct Yaw-moment Control (DYC, also known as VDC or ESP) is very effective in increasing the lateral stability of the vehicles [1, 2].

Based on the above discussion, there are two main methods to control yaw moment in order to maintain safe handling characteristics of vehicles. The first one is DYC which by

developing a difference in the longitudinal forces on two sides of the vehicle; an external stabilizing yaw moment is produced in the vehicle. The other method is active steering which works based on lateral tire forces control, through control of steering angle. The potential of active steering will be easily usable once Steer-by-wire (SBW) technology is established. Nowadays, the most practical approach to steering control is Active Front Steering (AFS) which is added a correction steering angle to the driver's steering input in this method.

Due to the inherent nonlinear characteristics of Pneumatic tire lateral forces, AFS performance is limited within linear vehicle handling region (low to mid-range lateral accelerations) [3]. On the other hand, in spite of good performance of DYC in both linear and nonlinear vehicle handling regions, continuing DYC actuation could lead to uncomfortable driving conditions. Furthermore, in emergency situations, when right and left sides of the vehicle are on different surfaces (μ -split condition); DYC would partially release the brakes on one side of the vehicle resulting a longer stopping distance.

From a controlling design view point, an integration of chassis control systems could be very useful. As a result, an intelligent combined strategy to integrated control of AFS/DYC ought to be used to allow AFS to perform in its effective range while providing the assistance of DYC in those situations where it is needed.

In Ref. [4], linear model matching theory and LQR control theory were applied to the design of the integrated control system. Wang et al. [5] presented a full-vehicle active suspension system to simultaneously improve vehicle ride comfort and steady-state handling performance. Alleyne [6] investigated the integration of various subsystems of an automobile's chassis. The specific focus of this research was the integration of active suspension components with Anti-lock Braking System (ABS) mechanisms. In Ref. [7] combining, coordinating steering and individual wheel braking have been investigated. In this study, a model regulator based on yaw stability controller was designed. March and Shim [8] developed an integrated control system of active front steering and normal force control by using fuzzy logic strategy.

Several approaches [9, 10] have been proposed under the names like integrated or global chassis control and integrated vehicle dynamics control. Among them, vehicle dynamics management (VDM) is the Bosch approach for coordinating vehicle dynamics functions with integrated control of active chassis systems [11].

In this paper, a new approach to integrated control of AFS/DYC is developed. The proposed approach is based on

R. Karbalaei is with the Department of Mechanical Eng., Azad University South Branch of Tehran, Jamalzadeh Ave., Tehran, Iran (e-mail: rezakma@yahoo.com).

A. Ghaffari is a professor and R. Kazemi is an associate professor at the Mechanical Engineering Department of K.N. Toosi University of Technology, Vanak Sq., Tehran, Iran (e-mail: {ghaffari, kazemi}@kntu.ac.ir).

S. H. Tabatabaei, is with the Department of Mechanical Eng., Azad University South Branch of Tehran, Jamalzadeh Ave., Tehran, Iran (e-mail: hamedtaba@yahoo.com).

the determination of two controlling cases in the active front steering system for generating a corrective yaw moment. Hence, a switching function is implemented in the lower layer to adjust the input scaling factor of the fuzzy logic controller. On the other hand, a rule-based integration scheme is proposed to optimize the overall vehicle performance by minimizing interactions between the two controllers and extending functionalities of individual AFS or DYC controllers. The results of simulation show that the proposed approach could provide appropriate composition of DYC and AFS control efforts to achieve new levels of performance in tracking of the desired vehicle yaw rate and handling stability in an intelligent manner.

The paper organization is as follows. Section II describes the structure of integrated control system. Section III is dedicated to the yaw rate controller and the fuzzy integrator. A switching function is proposed to adjust the input scaling factor of the fuzzy logic controller at the fuzzy integrator in Section IV; the control system performance is evaluated in Section V by computer simulation. Conclusion is finally made in Section VI.

II. STRUCTURE OF INTEGRATED CONTROL SYSTEM

The structure of proposed integrated control system is shown in Fig. 1. The control system has a hierarchical structure consisting of two controlling layers. The upper layer is the yaw rate controller which is designed based on fuzzy logic strategy. The outputs of yaw rate controller are the direct yaw moment control signal M_z and the steering angle correction of the front wheels δ_{cor} which are applied as the controlling inputs.

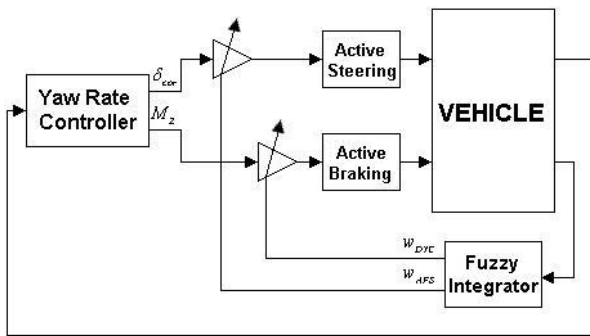


Fig. 1 Block diagram of control system

A fuzzy logic controller acting in the lower layer is named fuzzy integrator to manage the controlling inputs. Fuzzy integrator as an active supervisory module of the upper layer at the control system provides appropriate composition of the controlling inputs in order to enhance the amount of handling safety with fewer changes in behavior relation in order to make changes in driving conditions.

The direct yaw moment control is generated by the differential longitudinal forces (in this research, only the braking force) on two sides of the vehicle. On the other hand, the correction steering angle is added to driver's steering input

δ_{driver} , this task simply is accomplished if the vehicle will be equipped with a Steer-by-wire (SBW) system [3,13].

III. CONTROL SYSTEM DESIGN

A. Yaw Rate Controller

A fuzzy logic controller is designed in the upper layer of the control system to keep the yaw rate in its desired value. A desired yaw rate can be dynamically calculated based on the driver's steering input and longitudinal vehicle velocity [10] as follows:

$$r_d = \frac{v_x \delta_{driver}}{l + k_u v_x^2} \quad (1)$$

Where, v_x is the longitudinal vehicle velocity, l is the wheel base and k_u is the understeer gradient.

Fig. 2 shows the block diagram of the yaw rate controller. The yaw rate error is defined as:

$$e_r = r_d - r \quad (2)$$

Where r the yaw rate is. The yaw rate error and its rate of change are applied to fuzzy logic controller as inputs. As a consequent, by proper selecting gains of the k_{DYC} and the k_{AFS} as the outputs scaling factors, M_z and δ_{cor} are made as controlling inputs.

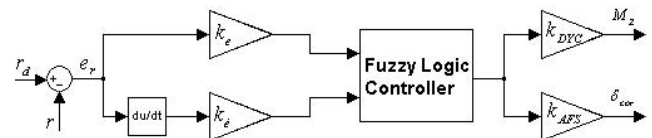


Fig. 2 Block diagram of yaw rate controller

The normalized membership functions for fuzzification of the controller inputs and defuzzification of the controller output are depicted in Fig. 3. Also, the rule base for the fuzzy logic yaw rate controller is shown in Table I.

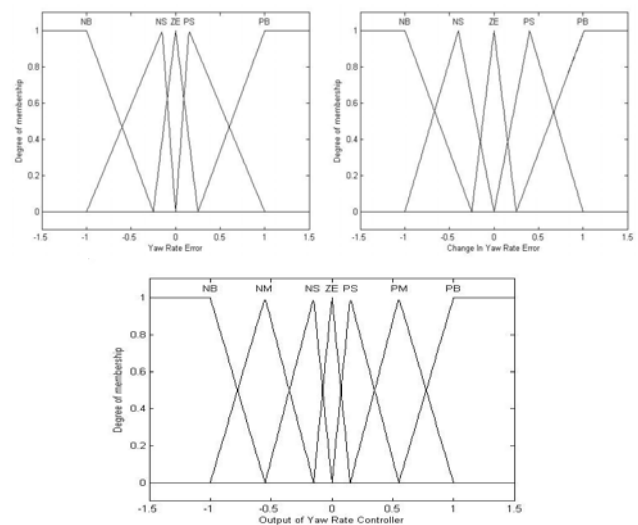


Fig. 3 Membership functions for the yaw rate controller

TABLE I
 FUZZY RULE-BASED FOR THE YAW RATE CONTROLLER

Output of Fuzzy Logic Controller		Change In Yaw Rate Error				
		NB	NM	ZE	PM	PB
Yaw Rate Error	NB	NB	NB	NB	NM	NS
	NM	NB	NB	NM	NS	ZE
	ZE	NM	NS	ZE	PS	PM
	PM	ZE	PS	PM	PB	PB
	PB	PS	PM	PB	PB	PB

B. Fuzzy Integrator

In the upper layer of the control system, only the potential of controlling is produced. In order to achieve a new level of performance in handling stability; another controller is needed to manage upper layer controlling. Therefore, a fuzzy logic controller with variable input scaling factor is designed in the lower layer. This controller according to the vehicle conditions determines the percentage of usage of controlling inputs actively. The layout of the fuzzy integrator is shown in Fig. 4.

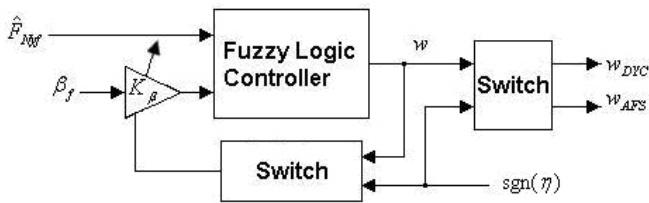


Fig. 4 Fuzzy integrator layout

w_{DYC} and w_{AFS} controlling weights which moderate AFS and DYC controlling inputs, are made as the outputs of fuzzy integrator. The fuzzy logic controller applied in fuzzy integrator is designed based on the working region of the lateral tire forces.

Because of the particular shape of the tire lateral force characteristics, two distinct kinds of vehicle handling behavior exist. The first, linear, is a kind of behavior occurring where tire-cornering characteristics remain within the linear range of operation. The second, nonlinear handling behavior, takes place when the tire forces do not increase proportionally to the slip angles. At this point, the tire slip angles may increase quite rapidly without corresponding increase in lateral forces [14]. On the other hand, the excessive at the tire slip angle can diminish control braking forces. Thus, degrades the performance of the DYC system. Therefore, an active front steering system can be used to maintain the sideslip angle of front wheels at or very near the peak of lateral force.

Based on the above discussion, in this research, the working region of the lateral tire forces is divided into three areas: linear area, nonlinear area and saturation area which are depicted in Fig. 5. Due to this division, a two-dimensional rule table is created based on the lateral forces and on the sideslip angle of tire, which is shown in Table II.

This rule table as a rule base is applied in the fuzzy logic controller. The front axle slip angle β_f and the estimated sum

of the lateral forces at the front axle, in the normalized rule \hat{F}_{Ny} , are applied to fuzzy logic controller as inputs. On the basis of these inputs, the control weight w , ($0 < w < 1$), is calculated as the output. The membership functions for the controller inputs and the controller output are shown in Fig. 6.

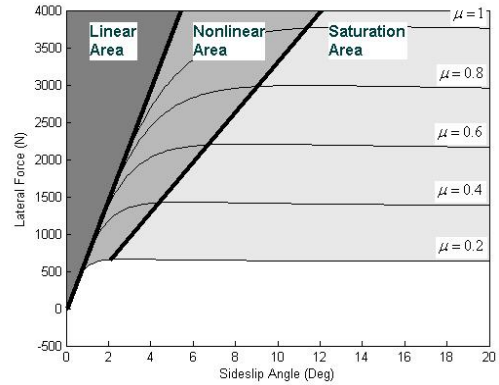


Fig. 5 Tire lateral force characteristics

TABLE II
 FUZZY RULE-BASED FOR THE FUZZY INTEGRATOR

$\frac{F_y}{W}$	Fy1	Fy2	Fy3	Fy4	Fy5	Fy6	Fy7	Fy8	Fy9
B1	W5	W5	W5	W5	W5	W5	W5	W5	W5
B2	W3	W4	W5	W5	W5	W5	W5	W5	W5
B3	W1	W3	W4	W5	W5	W5	W5	W5	W5
B4	W1	W1	W3	W4	W4	W5	W5	W5	W5
B5	W1	W1	W1	W3	W3	W4	W4	W5	W5
B6	W1	W1	W1	W1	W2	W3	W3	W4	W4
B7	W1	W1	W1	W1	W1	W2	W2	W3	W3
B8	W1	W1	W1	W1	W1	W1	W1	W1	W2
B9	W1	W1	W1	W1	W1	W1	W1	W1	W1

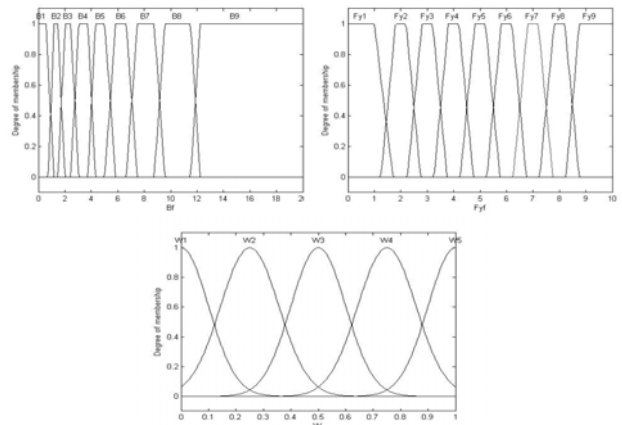


Fig. 6 Membership functions for the fuzzy integrator

Consequently, based on the front vehicle dynamics condition, the front tire working region and the vehicle maneuver's severity, the appropriate control is selected in the forms functioning only AFS, simultaneously AFS, DYC or DYC only through calculating of control weight w . Table III shows the control logics in the different working cases.

TABLE III
 CONTROL LOGICS

Working Area	Weight Control w	Action
Linear	w_5	Only AFS
Nonlinear	w_4, w_3, w_2	simultaneously AFS, DYC
Saturation	w_1	Only DYC

The front axle slip angle β_f is calculated based on 2-DOF vehicle model as follows:

$$\beta_f = \arctan\left(\frac{v_y + l_f r}{v_x}\right) - \delta \quad (3)$$

where v_y is the lateral vehicle velocity, l_f is the distance of the front axle to the center of gravity and δ is the front wheels steering angle. Sum of the lateral forces at the front axle can be estimated based on 2-DOF as follows:

The basic equation of motion for this model can be derived as follows:

$$ma_y = F_{yf} + F_{yr} \quad (4)$$

$$I_z \dot{r} = F_{yf} l_f - F_{yr} l_r + M_z \quad (5)$$

In the above equations, m denotes mass of the vehicle; I_z is the yaw moment of inertia of the vehicle; l_r is the distance of the rear axle to the center of gravity; F_{yf} is the lateral force of the two front tires; F_{yr} is the lateral force of the two rear tires, and M_z is the yaw moment generated by the longitudinal forces. From equations (4) & (5), the sum of the lateral forces at the front axle can be estimated as:

$$\hat{F}_{yf} = \frac{ma_y l_r + I_z \dot{r} - M_z}{2l} \quad (6)$$

\hat{F}_{yf} is normalized based on per unit of front mass as:

$$\hat{F}_{N_{yf}} = \frac{\hat{F}_{yf}}{m_f} \quad (7)$$

where $m_f = ml_r / l$. Finally, $\hat{F}_{N_{yf}}$ is inputted to fuzzy controller.

IV. SWITCHING FUNCTION

In this section, by introducing a switching function, a new methodology is proposed for the active front steering design and consequently, a new approach can be derived to the integrated control of AFS/DYC.

Depending on the directions of the lateral forces at the front wheels, there are two ways to generate a corrective yaw moment on a vehicle using AFS controller; either increasing lateral forces by increasing the sideslip angle to generate a desired yaw moment or decreasing lateral forces by reducing the sideslip angle to mitigate the undesired yaw moment arising from lateral forces at the front axle. Hence, two controlling cases are considered as follows.

Case -1. The first case of operation for the active front steering in different conditions is shown in Fig. 7(a). In this

case, a desired yaw moment is generated with increasing the lateral forces at the front wheels. On the other hand, the front wheels sideslip angle is increased by adding the correction steering angle to driver's steering input. Therefore, in this case, a corrective yaw moment can be generated via increasing the sideslip angle to peak value of the lateral forces.

Case -2. The second case of operation for the active front steering in different conditions is shown in Fig. 7(b). In this case, a corrective yaw moment can be applied to the vehicle by reducing the lateral forces at the front wheels. Also, the front wheels sideslip angle is decreased by adding the correction steering angle to driver's steering input. Hence, in this case, a compensating yaw moment can be generated with decreasing the sideslip angle to counter a potential undesired yaw moment.

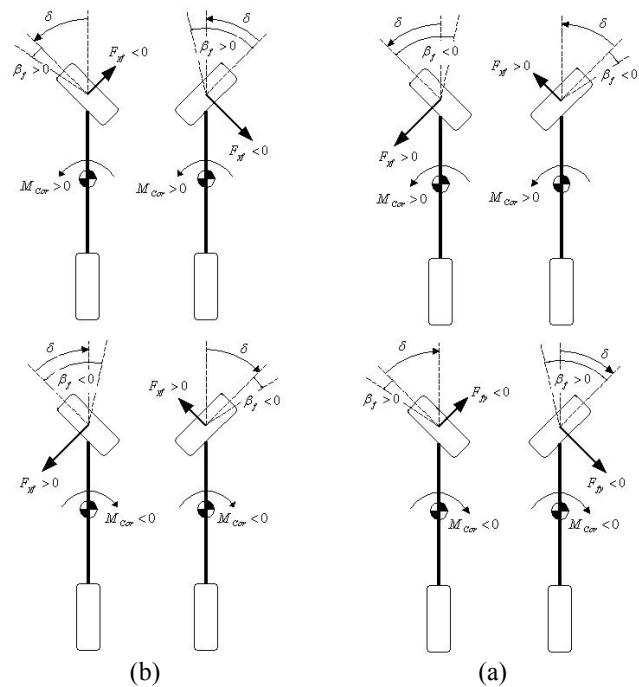


Fig. 7 Turn directions of the front wheels during the operation of active front steering in various conditions: a) case_1, b) case_2.

Using the above statements, for identification and distinctness of these two cases, depending on the direction of the lateral forces at the front axle a switching function is defined as:

$$\eta = M_z \cdot \beta_f \quad (8)$$

where M_z is the direct yaw moment control signal and β_f is the front axle side slip angle.

A. Control Procedure

Based on equation (8), the sign of the switching function, $\text{sgn}(\eta)$, is used in the fuzzy integrator to switch between two controlling cases. This task is accomplished due to change in the input scaling factor β_f which is shown by k_β (Fig. 4). As

a result, controlling weights w_{DYC} and w_{AFS} are calculated as follows:

Case -1. When $\eta < 0$, in this case $k_\beta = 1$ and the control logics in Table III are applied to moderate controlling inputs. In this case, controlling weights are selected as below:

$$\begin{cases} w_{AFS} = w \\ w_{DYC} = 1 - w \end{cases} \quad (9)$$

Case -2. When $\eta > 0$, in this case, the control logics in Table III can not be applied to moderate controlling inputs. In this case for reducing undesired yaw moment due to the lateral forces at the front axle, corresponding to conditions, the maximum allowable steering correction can be applied to driver's steering input for decreasing sideslip angle. So, through selecting, $k_\beta = w$ the steering correction is allowed to add to driver's steering input slowly. However, the steering correction is limited because of actuator limits. On the other hand, through adopting control weights correspond to Eq. (15), the sideslip angle at the front wheels reduces to a sufficient value.

$$\begin{cases} w_{AFS} = w \\ w_{DYC} = w \end{cases} \quad (10)$$

B. Distribution of Braking Torques

The direct yaw moment control is generated by applying braking forces only at either left or right side of the vehicle. In the conventional strategies, the most part of the braking torques (main braking) is applied on the individual wheels based on detection of the understeer or oversteer driving situations [2]. In this research, taking into consideration the direction of the lateral forces at each of wheels, a different strategy is proposed for distributing braking torques.

Fig. 8 shows the applied forces on the vehicle when the control system is activated ($M_z > 0$, $M_z < 0$). In this figure, the lateral force at each of the front wheels are exerted to vehicle in the opposite (contra) direction of the yaw moment control (M_z). Likewise, the lateral force at each of the rear wheels are exerted to vehicle in the same (pro) direction of the yaw moment control (M_z). Therefore, a switching function η is defined for each wheel as:

$$\eta_i = \text{sgn}(\beta_i M_z) \quad (i=1, \dots, 4) \quad (11)$$

Where β_i is the sideslip angle of the i th wheel and M_z is the direct yaw moment control.

Based on the above considerations, the most part of the braking torques (main braking) is applied on the wheel which its lateral forces are in the opposite direction of the yaw moment control (contra- M_z). Table IV shows the direction of the lateral force for each wheel relative to the yaw moment control.

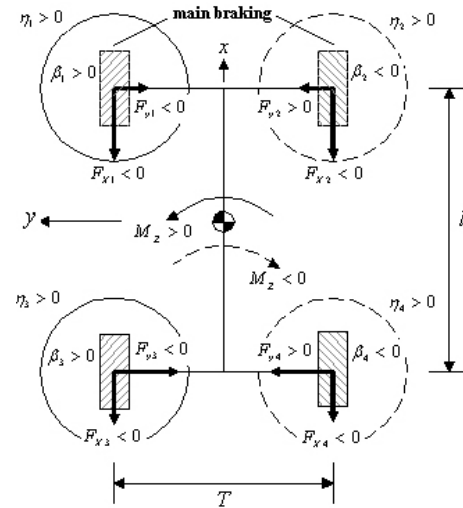


Fig. 8 Applying forces during activated control system

TABLE IV
 DIRECTION OF LATERAL FORCES IN RELATION TO YAW MOMENT CONTROL

η	Front Wheels	Rear Wheels
$\eta_i > 0$	contra- M_z (main braking)	pro- M_z
$\eta_i < 0$	pro- M_z	contra- M_z (main braking)

V. SIMULATION RESULTS AND ANALYSES

A nonlinear seven degrees of freedom vehicle model [15] with nonlinear Pacejka tire model [12] in combined slip condition is used for the simulation purposes. Three state variables: longitudinal velocity, lateral velocity and yaw rate, which determine vehicle motion in the yaw plane, are the main vehicle degrees of freedom. The remaining four degrees of freedom corresponds to rotational velocity of each wheel.

The simulation involves tracking a desired yaw rate for an emergency lane change maneuver. In this maneuver, vehicle runs at the initial velocity 80km/h on a snowy road with friction coefficient $\mu = 0.2$. The results are shown in Fig. 9.

Fig. 9(a) shows the performance of individual AFS, individual DYC and integrated AFS/DYC in tracking desired yaw rate. It is seen that the integrated AFS/DYC tracks the desired yaw rate more accurately than individual systems AFS or DYC. In Fig. 9(a), it is clearly seen that individual AFS using proposed approach, approximately in the second half of the maneuver, has been able to stabilize the vehicle in comparison with non controlled vehicle. This improvement in vehicle handling arises as result of reducing undesired yaw moment on the vehicle. It is also observed that in the beginning of the second half of the maneuver, the performance of the individual DYC in tracking of the desired yaw rate becomes inaccurate. It is due to diminishing the capability of the braking forces in generation of yaw moment control in the slippery roads. Fig. 9(b) shows the vehicle

trajectories for individual systems AFS, DYC and integrated system AFS/DYC. A comparison between front axle sideslip angle for two cases, the actual value and the fuzzy controller input at fuzzy integrator is shown in Fig. 9(c). Another comparison between lateral acceleration and the normalized sum of the lateral forces at the front axle is shown in Fig. 9(d). As shown in Fig. 9(c), at second one and second two, k_{β} is switched between two controlling cases and actual sideslip angle is decreased. Also, it is observed from Fig. 9(d) that sum of the lateral forces at the front axle and lateral acceleration are decreased in correspondent to decreasing front axle sideslip angle. Therefore, by reducing undesired yaw moment on the vehicle, a corrective yaw moment is applied to the vehicle. Fig. 9(e) and Fig. 9(f) shows the wheels braking torque command and the front wheels steering angle, respectively.

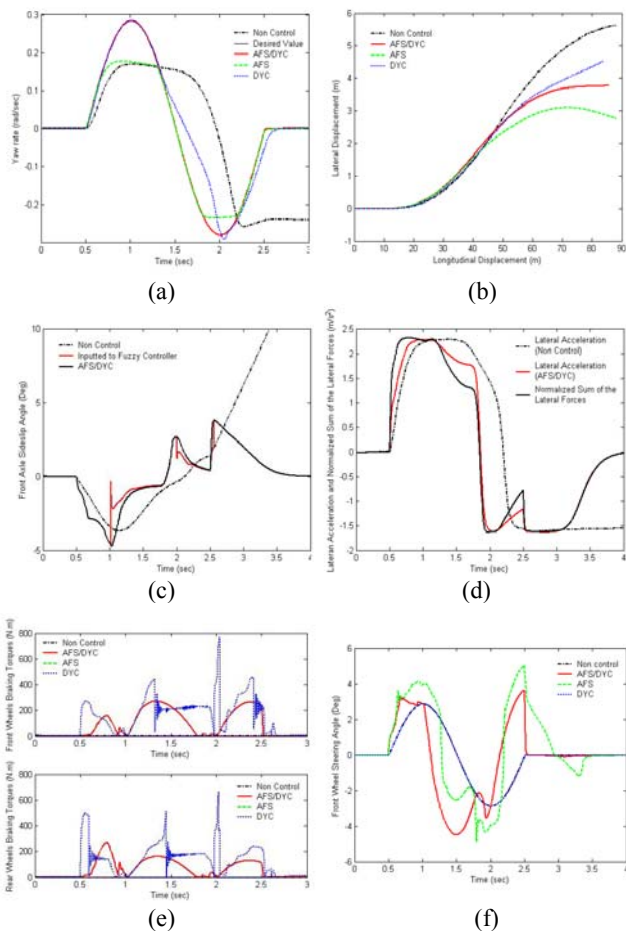


Fig. 9 Simulation results of lane change maneuver: a) Yaw rate, b) Vehicle trajectory, c) Front axle sideslip angle, d) Lateral acceleration and normalized sum of the lateral forces, e) Wheel braking torque, f) Front wheel steering angle

VI. CONCLUSION

In this paper, an integrated vehicle dynamics control system which aims to improve vehicle handling and stability has been designed. The proposed control system includes a central yaw rate controller in the upper layer and a controller

that coordinates the action of active front steering and direct yaw moment control in the lower layer. The proposed algorithm is based on the identification of two controlling cases in the active front steering system and a rule based on integration scheme. As the simulation results demonstrate, in the slippery roads that the capability of the braking forces in generation of yaw moment control are decreased, the proposed control system is able to increase the tracking performance of the desired vehicle yaw rate by reducing undesired yaw moment on the vehicle through a steering correction. Other advantages of the proposed control system are to improve the driving dynamics and as well as the reduction of the stopping distance in braking μ -split condition which could also be investigated through extension of the simulations.

REFERENCES

- [1] A. T. Van Zanten, "Evolution of Electronic control systems for improving the vehicle dynamics behavior", Proceedings International Symposium on Advanced Vehicle Control, vol. 20 024481, 2002, pp. 7-15.
- [2] A. T. Van Zanten, R. Erhardt, K. Landesfeind, and G. Pfaff, "VDC systems development and perspective," SAE Technical Paper, 980235, 1998.
- [3] Y. Shibahata, "Progress and future direction of chassis control technology," Annual Reviews in Control 29 (2005) 151-158.
- [4] M. Nagai, M. Shino, and F. Gao, "Study on Integrated Control of Active Front Steer Angle and Direct Yaw Moment", JSAE Review 23 (2002) 309-315.
- [5] J. Wang, D. A. Wilson, W. Xu, and D. A. Crolla, "Active suspension control to improve vehicle ride and steady-state handling," Proceedings of the 44th IEEE Conference on Decision and Control, and the European Control Conference 2005 Seville, Spain, December 12-15, 2005.
- [6] A. Alleyne, "Improved vehicle performance using combined suspension and braking forces," Proceedings of the American Control Conference, Washington, June 1995.
- [7] B. A. Güvenc, et al., "Coordination of steering and individual wheel braking actuated vehicle yaw stability control," IEE Conf., 2003.
- [8] C. March, and T. Shim, "Integrated control of suspension and front steering to enhance vehicle handling," Journal Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, Issue Volume 221, Number 4, 2007.
- [9] M. A. Selby, M. D. Brown, W. J. Manning, and D. A. Crolla, "A Coordination Approach for DYC and Active Front Steering" SAE Technical Paper, 2001-01-1275, 2001.
- [10] Y. A. Ghoneim, W. C. Lin, D. M. Sidlosky, H. H. Chen, Y. Chin, and M. J. Tedrake, "Integrated chassis control system to enhance vehicle stability" International Journal of Vehicle Design 2000 - Vol. 23, No.1/2 pp. 124-144.
- [11] A. Trachtler, "Integrated vehicle dynamics control using active brake, steering and suspension systems," International Journal of Vehicle Design 2004 - Vol. 36, No.1 pp. 1 - 12.
- [12] H. Pacejka, "Tire and vehicle dynamics," SAE, Warrendale, 2002.
- [13] A. Hac, "Control of brake- and steer-by-wire systems during brake actuator failure," SAE Technical Paper, 2006-01-0923, 2006.
- [14] A. Hac, "Influence of active chassis systems on vehicle propensity to maneuver-induced rollovers," SAE Technical Paper, 2002-01-0967.
- [15] T. Pilutti, et al., "Vehicle steering intervention through differential braking," Trans. of ASME, Vol. 129, pp. 314-321, Sept. 1998.