

# Design of an Integrated Active Front Steering and Active Rear Differential Controller using Fuzzy Logic Control

Samaneh Arabi, Mohammad Behroozi

**Abstract** — An integrated vehicle dynamic control system for a vehicle with an active front steering (AFS) and active rear differential (ARD), based on fuzzy logic control, is developed to improve the vehicle stability and its handling performance. The controller structure is consisted of two layers in a hierarchical arrangement. A fuzzy logic controller is used in the upper layer to keep the yaw rate in its desired value. The yaw rate error, lateral acceleration and the side slip angle are applied to the upper controlling layer as the inputs, where the desired traction torque transfer ratio and the steering angle correction of the front wheels are the outputs. However, the ideal control effectors could not be directly the control inputs for the rear differential. Therefore, in the lower control loop, one should map the ideal control effectors to the physical control inputs for the rear differential by optimum dynamic traction force distribution. A nonlinear eight degree-of-freedom (DOF) vehicle model with the traction force distribution being utilized by a PI controller is considered. The simulation results illustrate considerable improvements have been achieved for the vehicle stability and handling performance through the integrated AFS/ARD control system.

**Index Terms:** Active differential, Active front steering, Fuzzy logic control, Vehicle stability control.

## I. INTRODUCTION

Recent researches introduce yaw rate control as one of the most important methods in vehicle stability in extreme maneuvers and also this strategy can avoid dangerous and undesirable behaviors in vehicles. Several references have addressed this strategy, which makes direct changes in the yaw moment to adjust yaw rate [1-3]. Most of these approaches are based on braking forces in wheels, but provide serious restrictions in the vehicle performance [4, 5].

To tackle this problem, active differentials have the capability of usage in yaw rate control systems to avoid decrease in vehicle performance during different maneuvers. The system works on the basis of varying

traction forces on wheels, and therefore, will not have destructive effect on longitudinal response of vehicle. Intelligent differentials are much in demand with interest of public in vehicle safety. The control torque distribution on each wheel of vehicle system with one or two active differentials for safety driving have discussed by the numerous authors [6-8].

Another method to control the yaw moment is active steering which works on the base of lateral tire-force control through control of steering angle [9, 10]. The potential of active steering would easily be usable where the steer-by-wire (SBW) technology is established. Recently, a most practical approach to steering control is AFS in which a correction of steering angle is added to the driver's steering input. More recently, active steering has been combined with the yaw rate feedback to robustly decouple the yaw and lateral motions [11]. Several projects have been conducted on control of steering angle accompanied by the braking force control as a way to improve the vehicle handling and stability [12, 13].

To enhance the ability in control of vehicle lateral dynamic performance, the usage of an integrated vehicle dynamics control system for vehicle by a combination of active front steering (AFS) and active rear differential (ARD) is proposed. In this study, a new approach to integrated control of AFS/ARD is developed. The control system has a hierarchical structure consisting of two layers. A fuzzy logic controller is used in the upper layer to keep the yaw rate in its desired value. The desired traction torque transfer ratio and the steering angle correction of the front wheels are the outputs. However, the ideal control effectors could not be directly the control inputs for rear differential. Therefore, in the lower control loop, one should map the ideal control effectors to the physical control inputs for rear differential by optimum dynamic traction force distribution. It would finally cause a sensible enhancement in tracking the desired path and increasing rear-wheel drive (RWD) vehicle stability.

The structure of this study consists of three major parts. First, a mathematical modeling of the rear-wheel drive vehicle with intelligent rear differential is performed in accordance to the limited slip differential. Then, the structure of control system is applied to ensure the stability of the yaw rate based on a fuzzy logic control (FLC) and PI controller. These mitigate the oversteering or understeering tendencies of the vehicle when it is encountering dangerous cornering situation. Finally,

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computer simulation has been performed on an 8-DOF vehicle dynamic model to show the enhancements in the vehicle dynamics responses with the controllers.

## II. MODEL DESCRIPTION

### A. Eight DOF vehicle dynamic model

Since this research is focused on investigating how to improve the vehicle stability and performance by dynamically distribution of the torque on rear wheels and control of steering angle, mathematical analysis and modeling of RWD vehicle with active differential are necessary. A comprehensive eight DOF vehicle dynamic model is illustrated in Fig. 1 [14]. These include the longitudinal velocity  $u$ , lateral velocity  $v$ , yaw rate  $r$ , roll rate  $p$ , and the rotational speed of wheels  $w_i$ .

The model includes nonlinearities of the system, such as the nonlinear behavior of tires, nonlinearities exist in the longitudinal and the lateral tire normal load transfers, the roll steer effects, and the camber angle changes due to the vehicle roll. The model has been primarily developed as a tool for the vehicle dynamic computer simulation.

The four governing equations of motion are organized for the longitudinal and lateral motions along the X and Y axes, respectively, and the corresponding rotational motions of the yaw and roll about the Z and X axes.

$$ma_x = \sum_{i=1}^4 F_{xi} - mg \sin \gamma - C_d A_f \rho_a u^2 / 2 \quad (1)$$

$$ma_{yu} + m_s h' \dot{p} = \sum_{i=1}^4 F_{yi} \quad (2)$$

$$I_{zz} \dot{r} = a(F_{y1} + F_{y3}) - b(F_{y2} + F_{y4}) + (F_{x1} + F_{x2})T / 2 - (F_{x3} + F_{x4})T / 2 \quad (3)$$

$$I_{xxs} \dot{p} + m_s h' a_{yu} = m_s g h' \sin(\phi) - K_\phi \phi - C_\phi \dot{\phi} \quad (4)$$

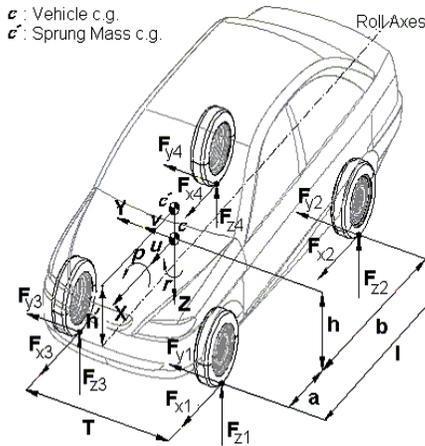


Fig.1. the 8-DOF vehicle handling model [14]

Where  $\phi$  is the roll angle of the sprung mass about the roll axis and  $a_x$  and  $a_{yu}$  are the longitudinal acceleration of the vehicle and the lateral acceleration of the unsprung mass.

$$a_x = \dot{u} - vr, \quad a_{yu} = \dot{v} + ur \quad (5)$$

The terms  $F_{xi}$  and  $F_{yi}$  are the respective tire forces in the X and Y directions, which could be expressed as functions of the traction and lateral tire forces with  $\delta_{Ti}$  being the steering angle.

$$F_{xi} = F_{Ti} \cos \delta_{Ti} - F_{si} \sin \delta_{Ti} \quad i = 1, \dots, 4 \quad (6)$$

$$F_{yi} = F_{Ti} \sin \delta_{Ti} + F_{si} \cos \delta_{Ti} \quad i = 1, \dots, 4 \quad (7)$$

$$I_{wi} \dot{\omega}_i = T_{hi} - T_{bi} - F_{zi} d \pm F_{xi} R \quad i = 1, \dots, 4 \quad (8)$$

The normal loads on wheels are mostly used in the modeling process of tires. Using the quasi-static lateral and longitudinal load transfer, the normal load expression for each wheel could be written as

$$\begin{cases} F_{z1} = \frac{W}{2} \left[ \frac{b}{l} - \frac{a_x}{g} \left( \frac{h}{l} \right) + K_R \left[ \frac{a_{ys}}{g} \left( \frac{h}{T} \right) - \left( \frac{m_s}{m} \right) \left( \frac{h'}{T} \right) \sin \phi \right] \right] \\ F_{z2} = \frac{W}{2} \left[ \frac{a}{l} + \frac{a_x}{g} \left( \frac{h}{l} \right) + (1 - K_R) \left[ \frac{a_{ys}}{g} \left( \frac{h}{T} \right) - \left( \frac{m_s}{m} \right) \left( \frac{h'}{T} \right) \sin \phi \right] \right] \\ F_{z3} = \frac{W}{2} \left[ \frac{b}{l} - \frac{a_x}{g} \left( \frac{h}{l} \right) - K_R \left[ \frac{a_{ys}}{g} \left( \frac{h}{T} \right) - \left( \frac{m_s}{m} \right) \left( \frac{h'}{T} \right) \sin \phi \right] \right] \\ F_{z4} = \frac{W}{2} \left[ \frac{a}{l} + \frac{a_x}{g} \left( \frac{h}{l} \right) - (1 - K_R) \left[ \frac{a_{ys}}{g} \left( \frac{h}{T} \right) - \left( \frac{m_s}{m} \right) \left( \frac{h'}{T} \right) \sin \phi \right] \right] \end{cases} \quad (9)$$

Where  $K_R$  is the front roll stiffness ratio that determines the front/rear distribution of the total lateral load transfer, and  $a_{ys}$  is the lateral acceleration of the sprung mass as

$$a_{ys} = \dot{v} + ur + (m_s / m) h' \dot{p} \quad (10)$$

The expression for the slip angle of each wheel, as a function of the variables, can be written as

$$\alpha_1 = \delta_{T1} - \arctan\left( \frac{v + ar}{u + \frac{1}{2}Tr} \right) \quad (11)$$

$$\alpha_2 = \delta_{T2} + \arctan\left( \frac{br - v}{u + \frac{1}{2}Tr} \right) \quad (12)$$

$$\alpha_3 = \delta_{T3} - \arctan\left( \frac{ar + v}{u - \frac{1}{2}Tr} \right) \quad (13)$$

$$\alpha_4 = \delta_{T4} + \arctan\left( \frac{br - v}{u - \frac{1}{2}Tr} \right) \quad (14)$$

Furthermore, the vehicle side slip angle is defined as

$$\beta = -\arctan(v / u) \quad (15)$$

The tire model which has been used in this handling model is known as "Combined Slip Magic Formula" in the non-linear tire model, [15]. The tire model calculates the traction and lateral tire forces of each wheel based on its slip angle, the normal force, and the longitudinal slip.

### B. Rear Limited Slip Differential

A schematic diagram of two clutches which represent electric limited slip differential is shown in Fig.2. Let us define  $T_{LSD-R}$  and  $T_{LSD-L}$  to be the torque transferred through the right clutch and left clutch,  $T_{diff}$  is the torque that is equally distributed to the left and right axle and transferred through the differential gear. If we assume efficiency of the torque is 100 % and differential has no mass inertia, we will have

$$(T_{LSD-R})_{rear} + (T_{LSD-L})_{rear} + (T_{diff})_{rear} = T_r \beta_{dr} \quad (16)$$

Traction torque to each wheels are given by

$$T_{rr} = (T_{LSD-R})_{rear} + \frac{(T_{diff})_{rear}}{2} = \frac{1}{2}[T_r \beta_{dr} + (T_{LSD-R} - T_{LSD-L})_{rear}] \quad (17)$$

$$T_{rl} = (T_{LSD-L})_{rear} + \frac{(T_{diff})_{rear}}{2} = \frac{1}{2}[T_r \beta_{dr} - (T_{LSD-R} - T_{LSD-L})_{rear}] \quad (18)$$

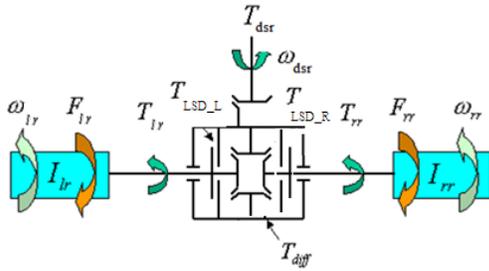


Fig.2. Front and Rear Limited Slip Differential

Distribution of traction torque to the right and left axle is defined to the left-right ratio parameter ( $\lambda_{L-R}$ ) and according above equation it can be obtained as function of  $T_{LSD}$  and  $T_r$ .

$$\lambda_{L-R} = \frac{T_{rr}}{T_{rr} + T_{rl}} \quad (19)$$

$$(\lambda_{L-R})_{rear} = \frac{1}{2} \left[ 1 - \frac{(T_{LSD-L})_{rear}}{T_r \beta_{dr}} + \frac{(T_{LSD-R})_{rear}}{T_r \beta_{dr}} \right] \quad (20)$$

Assuming zero inertia, clutch in electric limited slip differential can be modeled as a spring-damper torsional element [6].

$$(T_{LSD-R})_{rear} = c \cdot [(\omega_{diff})_{rear} - \omega_{rr}] + \int k \cdot [(\omega_{diff})_{rear} - \omega_{rr}] dt \quad (21)$$

$$(T_{LSD-L})_{rear} = c \cdot [(\omega_{diff})_{rear} - \omega_{rl}] + \int k \cdot [(\omega_{diff})_{rear} - \omega_{rl}] dt \quad (22)$$

### III. CONTROL SYSTEM STRUCTURE

The structure of proposed integrated control system is shown in Fig.3. The control system has a hierarchical structure consisting of two controlling layers. The upper layer is the yaw rate controller which is designed on the bases of fuzzy logic strategy. The outputs of the upper controller are the desired traction torque transfer ratio ( $\lambda_{L-R}^*$ ) and the steering angle correction of the front wheels ( $\Delta\delta_{Cor}$ ) which are applied as the controlling inputs.

In this method, by applying independent traction torques on rear wheels through rear differential, the required yaw moment is generated to correct the vehicle dynamic behaviour. This control method will be effectively useful under extreme cornering manoeuvres. An FLC is therefore designed as an upper layer in hierarchical controller to come up vehicle guidance strategy in dangerous maneuvers. The controller structure is indicated in Fig.3.

This upper control loop is used to calculate the ideal control effectors ( $\lambda_{L-R}^*$ ) which is required for realizing the steady and dynamic characteristics of the control system. However, the ideal control effectors could not be directly the control inputs for the rear differential.

Therefore, in the lower control loop, one should map the ideal control effectors to the physical control inputs for the rear differential by optimum dynamic traction force distribution. The traction force distribution is implemented by a PI controller.

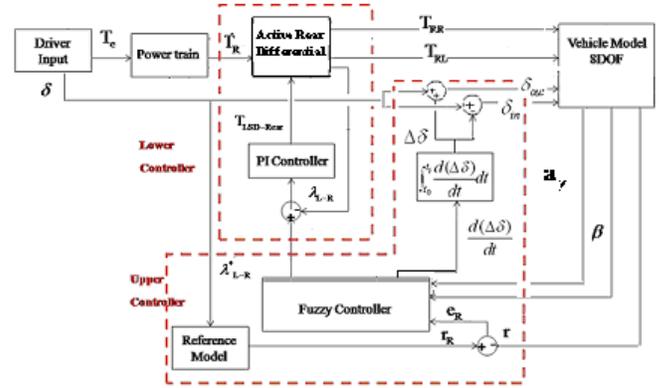


Fig.3. Block diagram of the proposed control system

It would be assumed that the vehicle control system is a rear wheel drive vehicle with a limited slip differential and can be used to transfer torque from the left to the right wheel as well as reverse direction. On the other hand, the correction steering angle is added to the driver's steering input  $\delta$ , and this task is simply accomplished if the vehicle is equipped with a steer-by-wire (SBW) system.

In this study, the inputs of the fuzzy controller are side-slip angle, lateral acceleration and the difference between actual yaw rate value (yaw rate ( $r$ ) of 8-DOF vehicle model) and the target value  $r_R$ , i.e.,  $e_r$  which are described as

$$e_r = r_R - r \quad (23)$$

Upper controller outputs are the desired traction torque transfer ratio ( $\lambda_{L-R}^*$ ) and the steering angle correction of the front wheels  $\Delta\delta_{Cor}$ . After integration of  $d(\Delta\delta_{Cor})/dt$ , the inner and the outer steering angles are finally calculated

$$\delta_{in} = \delta - \int_{t_0}^{t_i} \frac{d(\Delta\delta_{Cor})}{dt} dt \quad (24)$$

$$\delta_{out} = \delta + \int_{t_0}^{t_i} \frac{d(\Delta\delta_{Cor})}{dt} dt \quad (25)$$

A possible choice of the membership functions for the six mentioned variables of the system represented by a fuzzy set is as follows:

For side slip angle: HN (High Negative), LN (Low Negative), LP (Low Positive), HP (High Positive); For lateral acceleration: L (low), Mid (Mid), H (High), For yaw rate error: N (Negative), Z (Zero), PL (Positive Low), PH (Positive High); steering angle correction of the front wheels ( $\Delta\delta_{Cor}$ ): (Negative High, Negative Low, Zero, Positive Low and Positive High); and desired traction torque transfer ratio for rear differential  $\lambda_{L-R}^*$ : LH (Left High), LL (Left Low), M (Mid), RL (Right Low), RH (Right High) as shown in Fig.4 and 5.

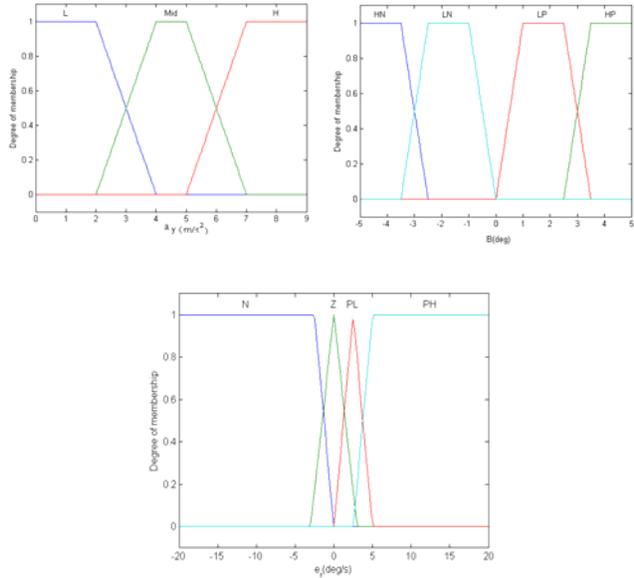


Fig. 4. Membership functions of fuzzy controller's inputs.

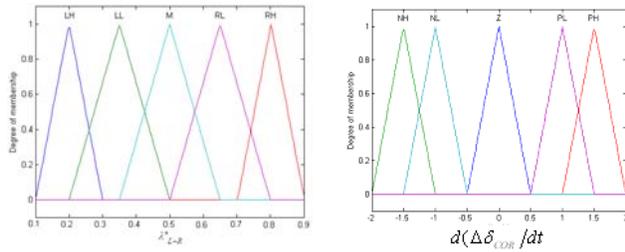


Fig. 5. Membership functions for fuzzy controller's output.

By using the fuzzy control theory, the outputs are calculated based on appropriate fuzzy rules. The rule base used in the system can be represented by the following table with fuzzy terms derived by modeling the designer's knowledge and experience. The linguistic control rules of the fuzzy logic controller obtained from the Tables 1 and 2 used in general form of:

$$R_i : IF(\beta = A_i) AND(a_y = B_i) AND(e_r = C_i) THEN(\lambda_{L-R}^* = D_i, d(\delta_{COR}/dt) = E_i)$$

Where  $A_i, B_i, C_i, D_i$  and  $E_i$  are labels of fuzzy sets representing the linguistic values of  $\beta, a_y, e_r, \lambda_{L-R}^*$  and  $d\delta_{COR}/dt$  respectively, which are characterized by their membership functions.

Table 1. Fuzzy inference rules for  $\lambda_{L-R}^*$

$\beta$ (degree)	$a_y$	$e_r$			
		N	Z	PM	PH
High Negative	Low	LL	LL	LL	LL
	Mid	LH	LH	LL	LL
	High	LH	LH	LH	LH
Low Negative	Low	M	M	RL	RH
	Mid	LL	M	RH	RH
	High	LL	M	RH	RH
Low Positive	Low	M	M	LL	LH
	Mid	RL	M	LH	LH
	High	RL	M	LH	LH
High positive	Low	RL	RL	RL	RL
	Mid	RH	RH	RL	RL
	High	RH	RH	RH	RH

Table 2. Fuzzy inference rules for  $d(\delta_{COR}/dt)$

$\beta$ (degree)	$a_y$	$e_r$ (degree/sec)			
		N	Z	PL	PH
Low	Low	NL	Z	PL	PH
	Mid	NL	Z	PL	PH
	High	NH	Z	PL	PH
High	Low	NH	NH	NL	NL
	Mid	NH	NH	NH	NH
	High	NH	NH	NH	NH

In the event that side-slip angle is provided in the desired region, the controller tracks the reference model yaw rate, and in case the side-slip angle exceeds the boundary of the desired region, the FLC forces the vehicle to move to safe and desired region of yaw rate. The control strategy is based on the following facts:

- When vehicle is cornering in high lateral acceleration and yaw rate error is negative, the vehicle is oversteered. In this case, FLC will order to transfer traction torque toward wheel located inside corner by rear differential, and also increase the steering angle of the inner wheel and decrease the steering angle of the outer wheel, which can decrease the front axle's total lateral force and leads to more stability.
- In severe understeering condition, the yaw rate error is positive when the side slip angle  $\beta$  is still in the safe region. To improve the vehicle handling and the yaw rate following characteristic, by increasing steering angle of the outer wheel and decreasing the steering angle of the inner wheel, one could increase the total lateral force applied on the front axle. The traction torque is transferred to the outside corner wheel to generate more lateral force in the event that yaw rate error is positive, the vehicle is in less understeer situation than reference model and side-slip angle is in the desired region.
- In the situation that the side-slip angle is not in the desired region, the control objective is just set to decrease the side-slip angle and consequently, the traction torque of rear inside corner wheel and inner front wheel steering angle increase.

In this study the Mamdani algorithm is employed for the fuzzy logic operation and the Gravity algorithm is utilized for the calculation of the defuzzification.

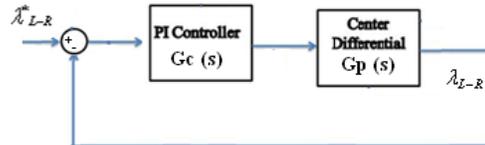


Fig. 6. Lower control loop

The lower controller attempts to achieve the desired distribution of the traction torque to the front and rear axles through central differential using a PI controller. In the lower control loop, shown in Fig.6, a PI controller is proposed, in which the PI controller integrates the error between the traction torque transfer ratio and the desired current to generate the control torque.

This value is then fed into the 8-DOF model of RWD vehicle. The equation of the PI controller can be expressed as

$$\lambda_{L-R} = K_p \Delta + K_I \int \Delta dt \quad (26)$$

Where  $K_p$  is the P gain,  $K_I$  gain the I gain, and  $\Delta$  is given as

$$\Delta = \lambda_{L-R}^* - \lambda_{L-R} \quad (27)$$

The traction torque of each rear wheels can be expressed from equations (19) – (22).

#### IV. NUMERICAL SIMULATION AND ANALYSIS

Computer simulation of the system is performed to evaluate the performance of the integrated control of the AFS/ARD as an intelligent control system.

An increase in the traction force on the rear inner wheel and steering angle of front inner wheel, as can be seen from Fig.8, would generate the turning moment in the counter clock-wise (CCW) direction. As a result of generating a CCW turning moment, the stability tendency of the vehicle moves toward the understeered case and the side slip angle is decreased.

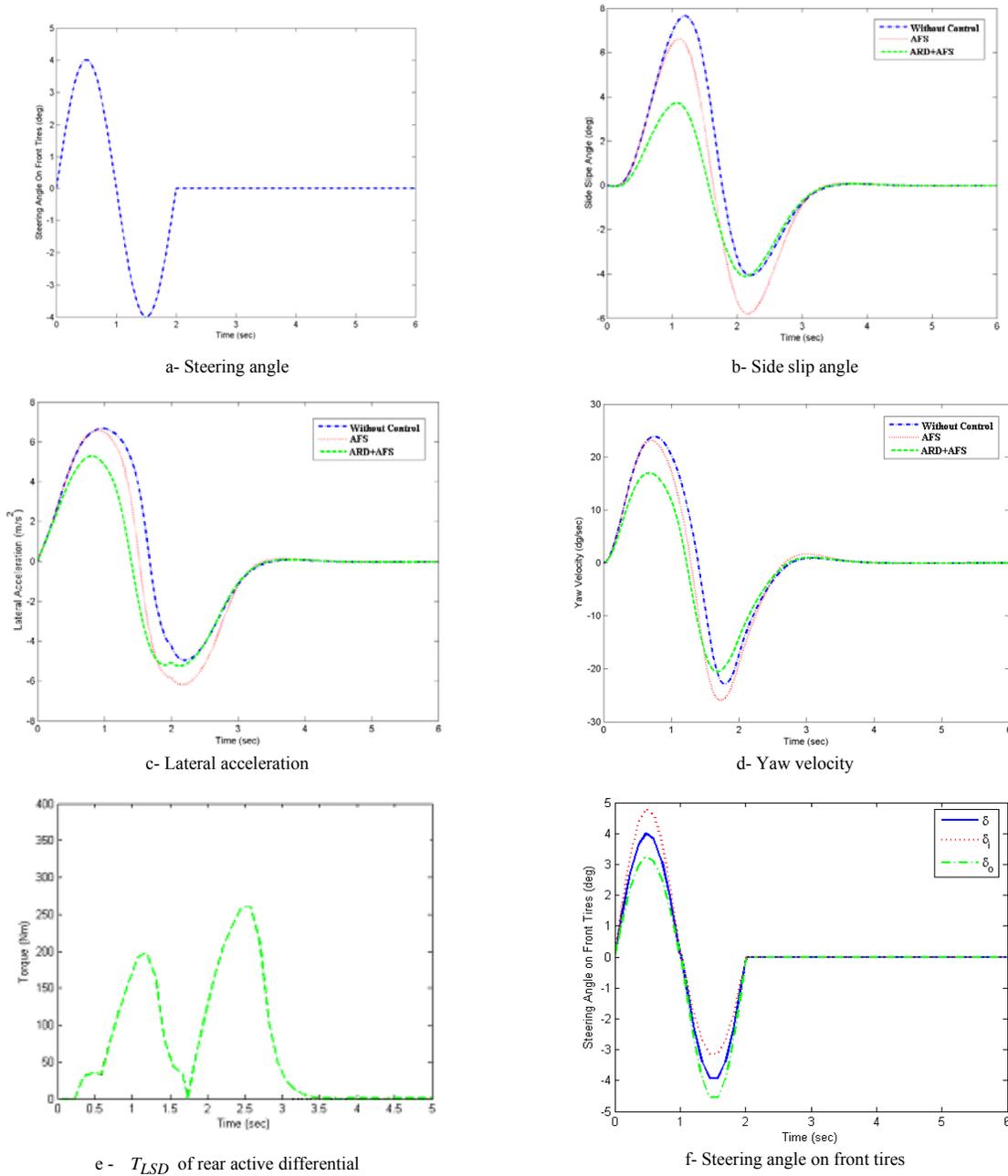


Fig.7. Simulation results for Single lane change maneuver on a high -  $\mu$  surface

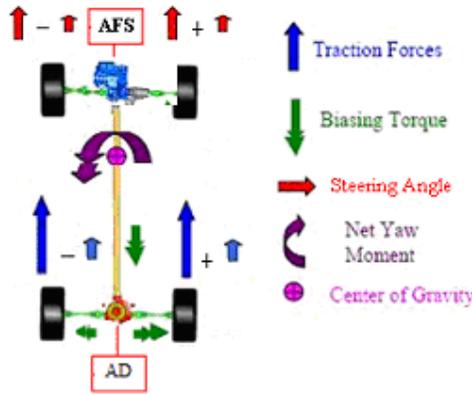


Fig.8. Yaw rate control.

The simulation case study consists of a high lateral acceleration lane change maneuver at speed of 30 m/s on a road with friction coefficient  $\mu_{Road} = 0.9$ . In simulation, the lane change maneuver is completed in 2 seconds with two pulses ( $\delta = \pm 4$  deg) as it is shown in Fig.7-a. The vehicle is considered to be inherently oversteered and the simulation results are illustrated in Fig.7. It is evident from Fig.7-b, that the side slip angle of the uncontrolled vehicle grows rapidly and the vehicle shows a strong oversteered behavior. However, in the case of the controlled vehicle, the side slip angle has been effectively controlled well below the acceptable limit to intelligently control of the steering correction angles and the distribution of traction torque. It can be seen that integrated AFS/ARD has better performance and decreases the value of side-slip angle to 3.5deg by transferring 90% of the traction torque to the rear inner wheel; and also increases the front inner steering angle and decreases the front outer steering angle. In this case, the side slip control is the first priority of the control system.

## V. CONCLUSION

In this paper, an integrated dynamic control system AFS/ARD which aims to improve vehicle handling and stability without interfering in vehicle performance has been designed. The proposed control system includes two layers. A fuzzy logic controller is used in the upper layer to keep the yaw rate in its desired value and the lower control loop maps the ideal control effectors to physical control inputs for rear differential by optimum dynamic traction torque distribution.

In accordance with the simulation results, the integrated controller (AFS/ARD) could intelligently control the steering angle while active differential device employs appropriate traction force combinations in various conditions. It can provide the best possible yaw rate following performance while keeping the side slip angle as low as possible to simultaneously guarantee the safety and stability of the vehicle. This mitigates the vehicle's oversteering or understeering tendencies while the vehicle is undergoing with dangerous cornering situation.

It also compares the feasibility and effectiveness of the individually AFS system with respect to the conventional systems with no controllers. Due to the capacity of actuators

and control strategy the integrated AFS/ARD controller showed better capabilities than AFS to guide the vehicle to safe region of side-slip angle – owing to AFS limitations in membership function of side-slip angle (Fig.10b) and its capacity to generate proper steer angle, vehicle cannot be severed by AFS lonely even though a couple of changes is provided in applied steer angle of which effects is not serious.

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