Bulletin of Environment, Pharmacology and Life Sciences
Bull. Env.Pharmacol. Life Sci., Vol 4 [Spl issue 1] 2015: 142-151

Online ISSN 2277-1808
Journal's URL:http://www.bepls.com
CODEN: BEPLAD
Global Impact Factor 0.533
Universal Impact Factor 0.9804

FULL LENGTH ARTICLE **OPEN ACCESS**

Designing of Active Differential System for the Improvement of Vehicle Stability during the Maneuver of Sudden Arrival to Low Friction Road

Mohammad Taha Shafaei*[1], Hassan Zarabadipour[2]
[1] Department of Engineering, College of Mechanical Engineering, Takestan Science and Research Branch, Islamic Azad University, Takestan, Iran
[2] Department of Electrical Engineering, Imam Khomeini international University, Qazvin, Iran
Email: Tahashafaei@gmail.com

ABSTRACT
This paper aims to examine the intelligent controller design for the integration of active steering control and anti-lock braking systems and throttle and active differential whose direct yaw moment control is done through active differential located in the rear axle. Also, an anti-lock braking system and active steering control that only operates on each of the front wheels accompany the active differential in emergencies in the production of external torque. This intelligent integrated control system is designed in two layers: using the optimal controller method of rear differential clutching torque, the upper layer of the controller announces the yaw moment, produced by front wheels braking system, and the front steering angle correction, which is applied to the steering wheel, to the subsystems in order to follow the desired dynamic behavior of the automobile. In the second layer with regard to these three transmitted values, the clutching force in active differential, braking torque and front steering angle are applied to the vehicle. Another process is running simultaneous with these processes which will reduce the throttle angle if the slip of rear wheels exceeds a certain amount. The simulation results, using nonlinear six-degree-of- freedom model and magic tire model, show that the designed integrated control system is able to maintain the vehicle stability in all moving and road conditions. In this paper, only the maneuver of sudden arrival in the low friction road is discussed.
Keywords: *intelligent integrated controller, active differential, active steering, optimal controller, magic tire model, low friction road*

INTRODUCTION

In automotive engineering the term active safety commonly refers to the systems that automatically reduce the likelihood of an accident. These systems help the driver control the car more and more by controlling longitudinal or lateral dynamics. Among the active safety systems, direct yaw moment control (DYC) and active steering control (ASC) systems are the effective methods in vehicle lateral dynamics control. These systems control the vehicle yaw moment and consequently the yaw angular velocity and the side slip angle. Since each one of the systems has the highest efficiency within a specific area of the tire performance, the use of an integrated control system which makes the optimal use of both systems is useful to improve the systems performance and to maintain the vehicle stability and steering in all cases.

According to the conducted research, it has been proved that the active front steering system is more effective than other active steering systems and thus active front steering (AFS) system has been used for active steering which controls the vehicle with regard to the vehicle lateral dynamics and the driver's desire.

One of the most common methods for the yaw moment control is the use of asymmetric braking on both sides of the vehicle. Asymmetric braking is the most effective and the most common way to create rotational torque in vehicles due to the use of ABS subsystem and its effectiveness on maintaining the vehicle stability in critical situations.

As it is observed in Figure (1), while the front wheels lateral force decreases and the front part of the vehicle desires to be pulled out of the road turn (low steering vehicle), DYC system by using ABS system applies a breaking force to the rear wheel within the turn which causes a rotational torque towards the inner part of the turn and directs the vehicle towards the inner turn.

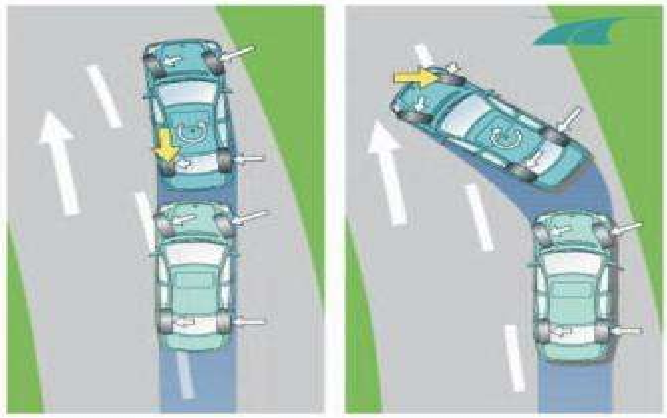

Figure (1): Vehicle stability control by applying asymmetric braking force

1.2. Vehicle Stability Control via Active Differential

Recently, the researchers have focused on the use of asymmetric traction torque to create rotational torque. The use of this mechanism (Figure 2) when the vehicle is accelerating causes the further acceleration of the vehicle in addition to the system stability. This point indicates the most important advantage of active differential (AD) rather than the conventional stability control systems which decrease the longitudinal velocity by braking.

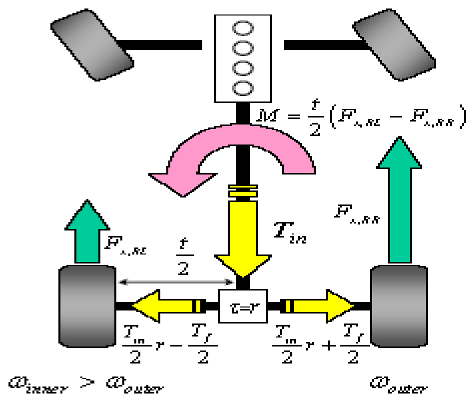

Figure (2): Vehicle stability control via active differential

Integrated Vehicle Dynamics Control:

Active steering and direct yaw moment control systems are the effective methods used in vehicle handling and stability control. Nevertheless, each one of these systems has the best performance in certain circumstances. As it is observed in Figure (3), the lateral force of vehicle tire based on its slip angle in different road conditions (various coefficients of friction) has three areas: linear area, nonlinear area, and saturation area.

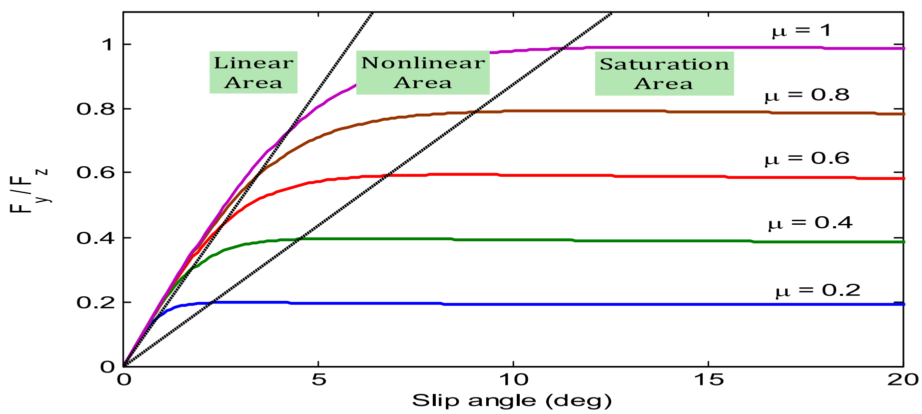

Figure (3): linear, nonlinear, and saturation areas of tire lateral force based on slip angle

For instance, active steering control system in the linear area of the tire is able to control the vehicle stability and DYC system is also more effective when the vehicle is in the nonlinear area of motion, and wheels steering is inefficient due to the saturation of tires lateral force. Therefore, the integrated control of these two systems can overcome this problem and improve the vehicle dynamic behavior at different circumstances of road and vehicle maneuvering.

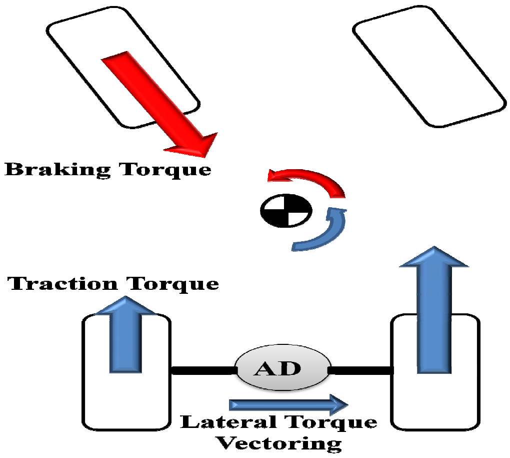

Figure (4): Direct yaw moment control via differential and braking

The goals of this article and the ways to deal with them are discussed in five parts. In the second part, vehicle dynamics modeling and active differential for simulation are examined including the introduction of a vehicle nonlinear six-degree-of-freedom model and nonlinear model of tire in compound slip and an active differential dynamics model with clutch. In the third part, the integrated controller design for the active front steering system, rear differential clutching torque, and front wheels braking torque will be examined. In the fourth section, the designed controller efficiency in pursuing the desired values of side slip angle and yaw rate will be investigated. Then, the effect of changing vehicle parameters such as mass, position of mass center, and error in estimation of road friction coefficient on controller efficiency will be reviewed and finally the conclusion and the obtained results in this paper will be presented.
Vehicle dynamics control modeling requires the mathematical modeling of its dynamic behavior. In this paper, the six-degree-of-freedom model of vehicle with DOF of longitudinal velocity, lateral velocity, yaw rate, sprung mass roll angle, a degree of freedom for the rear wheel yaw rate, and the sub model of a degree of freedom of steering mechanism are used.
In this research, an ordinary differential plus two electromagnetic clutches are used and the use of clutches enables the differential to control the torque of each axis independently.

Vehicle Dynamic Modeling

Vehicle modeling is required for stimulation its motion. Therefore, the model needs to be able to show the main properties of the vehicle motion. There are numerous models for the vehicle with different complexities. More complicated models produce more accurate results provided that their input parameters are accurate. However, obtaining accurate results require stronger processors and more time for simulation. On the other hand, simpler models are less accurate, but their simulation is easier. Therefore, a model which owns the main features of the system is suitable for most applications and provides acceptable results. According to the experiences gained from doing much work on modeling vehicle motion, a suitable model for vehicle should have nonlinear factors and degrees of freedom of plate motion and roll and the rotational dynamics of wheels and steering mechanism. Therefore, a model which owns the main features of the system is suitable for most applications and provides acceptable results. With regard to the experiences gained from numerous work that has been done on vehicle motion modeling, a suitable vehicle model should include nonlinear factors and degrees of freedom of plate and roll motion in addition to wheels rotational dynamics and steering mechanism. Thus, in this paper, for vehicle dynamics simulation, a comprehensive nonlinear model with degrees of freedom of longitudinal velocity (u), lateral velocity (v), yaw rate (r), sprung mass roll angle (ϕ), a degree of freedom related to rear wheel rotational rate (ω_i), and front steering angle (δ_f) is used. The schematic model of six degrees of freedom, coordinate axes, and forces acting on vehicle is illustrated in Figure (5).

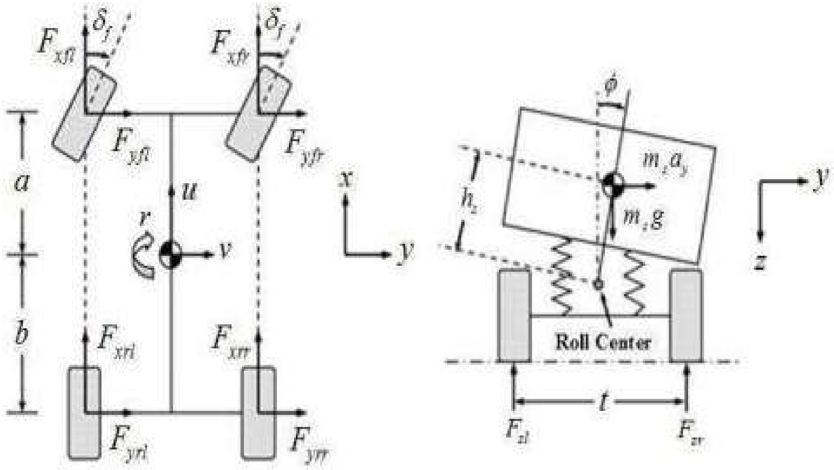

Figure (5): Vehicle six-degree-of-freedom model

Tire Model:

To solve the differential equations of vehicle model mentioned in the proviso section, a model is needed to obtain longitudinal force, lateral force, and tire aligning torque. In order to obtain results consistent with reality, the tire model outputs must be close to the actual results as much as possible. Various models have been proposed for the tire each with their own advantages and disadvantages. In this regard, nonlinear models are more accurate and practical than linear models.

A nonlinear tire model which is used in most vehicle dynamics simulations is the Magic Formula tire model known as Pacejka 2002 Tire Model. This model is mainly experimental in nature and is composed of a set of mathematical functions which is partly based on the physical properties of the tire. That is why it is called the semi-empirical tire model, as well.

Using longitudinal slip ratio, slip angle. Camber angle, and vertical force of tire as the input, the magic formula tire model calculates longitudinal and lateral forces and aligning torque of tire in two states of pure slip and compound slip.

Vehicle Rear Differential Dynamic Model

Ordinary differentials always divide the torque evenly between the wheels. If we want to create yaw moment via differential, we must be able to apply unequal traction force and consequently unequal traction torque to both wheels. It is possible to meet the demand by adding two clutches to the ordinary differential. As it is shown in the following figure, active differential is in fact an ordinary differential plus two electromagnetic clutches which are controlled electrically. These clutches are much faster than hydraulic clutches, so that clutching time in this sample has reduced to less than 0.05 sec. In this research, the transient behavior of clutching is modeled with a first degree function of time delay and a fixed time coefficient is considered for it. Therefore:

Eq.1) $\tau \dot{T}_{diff} + T_{diff} = T_{diffss}$

In the above equation, clutching torque is in steady state. τ is also the time coefficient of the differential equation which is considered to be 0.05 sec with regard to the clutches inertia.

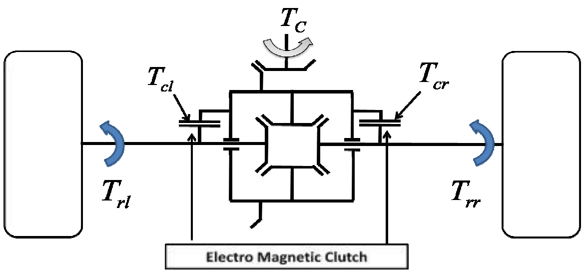

Figure (6): Schematic view of active electromagnetic differential

Engine Model

In this research, the data of XU7JP4 engine was used. The engine torque based on its round at different throttle angles is observed in the following illustration. In this study, the change of throttle angle as one of the controlling parameters of the system causes the change of engine torque.

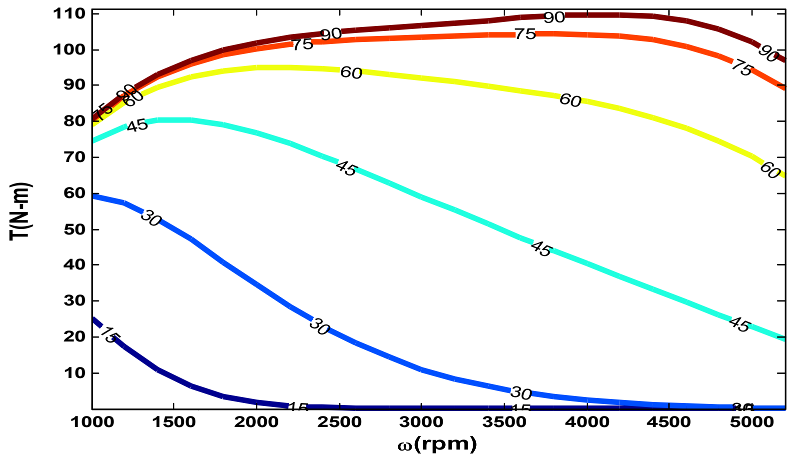

Figure (7): Diagram of XU7JP4 engine

In order to verify the performance of nonlinear 6-degree-of-freedom model build in Simulink environment in MATLAB software, the model was compared with the laboratory data of a Pickup which was obtained in Zamyad Company. It should be noted that the parameters of the vehicle in 6-degree-of-freedom model and the pickup are the same and in both models the compound slip tire of Pacejka 94 has been used.

Controller Design

In this study, a two-layer controller is designed. Using the optimal controller method of rear differential clutching torque (T_{diff}), the upper layer of the controller announces the yaw moment produced by front wheels braking system (M_{zb}), and the front steering angle correction to the subsystems in order to follow the desired dynamic behavior of the vehicle.

In the second layer, with regard to these three transmitted values, the clutching force in active differential, braking torque and front steering angle are applied to the vehicle. Another process is running simultaneous with these processes which will reduce the throttle angle if the slip of rear wheels exceeds a certain amount. A detailed block diagram of the integrated control system is schematically displayed in Figure (8).

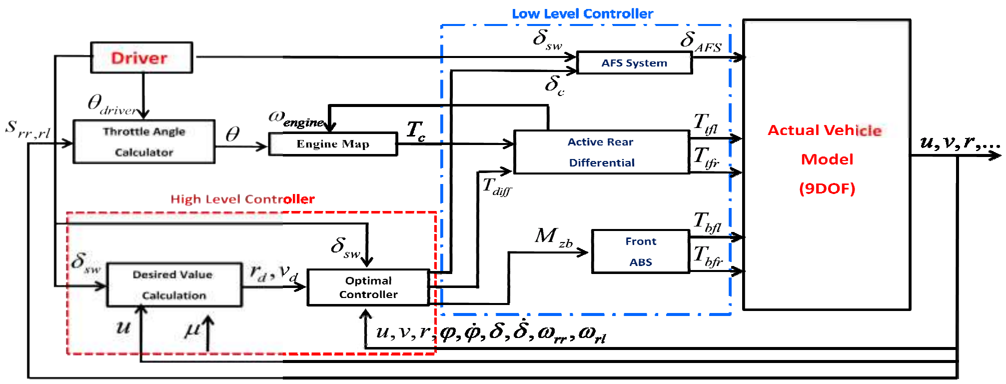

Figure (8): The block diagram of the vehicle control system

Equations Linearization

In this study, the six-degree-of-freedom equations of the vehicle have been linearized in order to design the controller. The general technique for the linearization of nonlinear equations is motion based on the small perturbation theory. For the application of this famous technique, a series of stable conditions is considered for the movement of the vehicle in which all the variables gain constant values for a fixed steering input. For a small perturbation on the steering input, all state variables experience small perturbations around the trim levels. Such perturbations are also created for external forces and moments which are functions of state and input variables. If the steady state equations (reference) are subtracted from dynamic equations by removing and discarding the quadratic and higher terms of small perturbations, a set of linear equations will be found based on the perturbed variables. Small perturbation values of external forces and torques are expressed as the linear functions of small perturbation and input variables as the following:

$$\text{Eq.2)}\quad [M]_{8\times 8}[\dot{X}]_{8\times 1} = [A]_{8\times 8}[X]_{8\times 1} + [B]_{8\times 3}[U]_{3\times 1} + [E]_{8\times 1}[W]_{1\times 1}$$

$$\text{Eq.3)}\quad X=\begin{bmatrix} v \\ r \\ \varphi \\ \dot{\varphi} \\ \delta \\ \dot{\delta} \\ \omega_3 \\ \omega_4 \end{bmatrix},\ M=\begin{bmatrix} m & 0 & 0 & m_s h_s & 0 & 0 & 0 & 0 \\ 0 & I_{zz} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ m_s h_s & 0 & 0 & I_{xx} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & I_{fw} & 0 & 0 & 0 & I_{fw} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & J_W & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & J_W \end{bmatrix},\ B=\begin{bmatrix} 0 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{K_{ss}}{N_g} \\ 1 & 0 & 0 \\ -1 & 0 & 0 \end{bmatrix},U=\begin{bmatrix} T_{diff} \\ M_{zb} \\ \delta_{swc} \end{bmatrix},\ E=\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{K_{ss}}{N_g} \\ 0 \\ 0 \end{bmatrix},\ W=[\delta_{sw}]$$

Stability derivatives in different trim levels can be achieved by using a computer code. Therefore, a program is written in MATLAB software in order to linearize the system.

One of the most useful methods of control is the optimal control of linear design chaser because it has a firm and understandable mathematical foundation in logic. This method requires creating a linear model from the controlled nonlinear system and by applying a set of matrix relations, it reaches the equation system and gives au the control benefits. These control advantages together with the system quantity supply the operator input in a controlled system which leads to the logical stabilization of the controlled system.

Simulation:

After designing the integrated controller in previous section, the vehicle behavior with and without the controller will be investigated. Therefore, the integrated controller is compared with AFS, ESP, and AD controllers and the simulation has been run for each of the four controllers.

In order to observe the overall effect of controller on the vehicle dynamic behavior, the vehicle motion has been simulated without controller and with integrated controller and AFS, AD, and ESP controllers separately, with the primary longitudinal velocity of 30 m/s and the stable steering angle of 30°. Three seconds after the launch of simulation the vehicle suddenly goes from the dry road with friction coefficient of 0.9 $(\mu=0.9)$ to the slippery road with the friction coefficient of 0.2 $(\mu=0.2)$. The simulation results are shown in Figures (9 to 19).

As seen in Figure (9), in the vehicle without control system due to the increase of longitudinal slip the speed has dropped while the use of active differential has increased the vehicle longitudinal velocity. Moreover, ESP controller has reduced the longitudinal velocity due to front wheels braking.

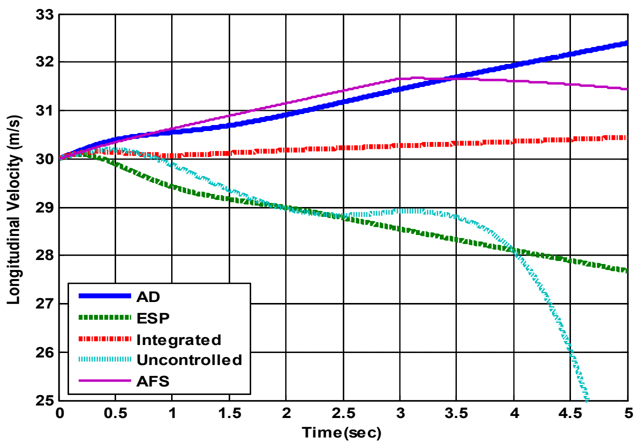

Figure (9): Vehicle longitudinal velocity in the maneuver of sudden arrival to the low friction road

According to the figures (10 , 11) the vehicle without controller, after entering the low friction road, becomes instable because of the saturation of tire lateral forces and its yaw rate and side slip angle increase rapidly while in the vehicle with integrated control the yaw rate and the side slip angle of the vehicle will converge to the optimal values. As it is observed, braking rather than active differentiation has made the vehicle real values closer to the desired values.

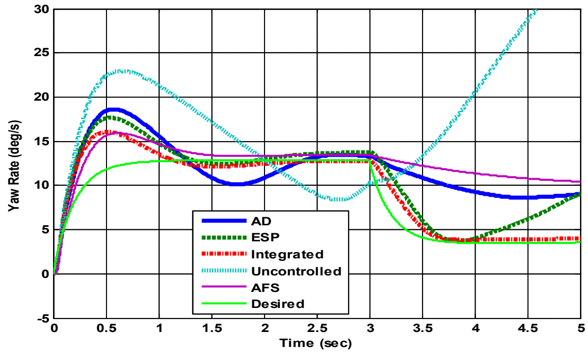

Figure (10): Yaw angular rate in the maneuver of sudden arrival to low friction road

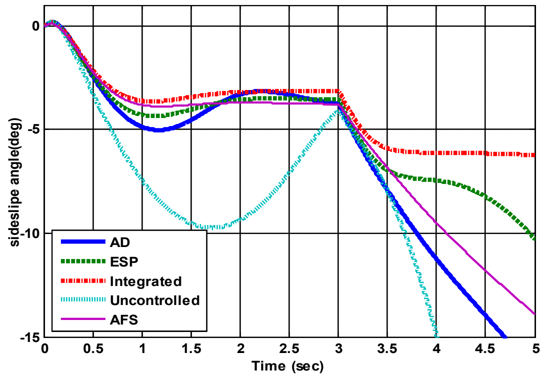

Figure (11): Side slip angle in the maneuver of sudden arrival to low friction road

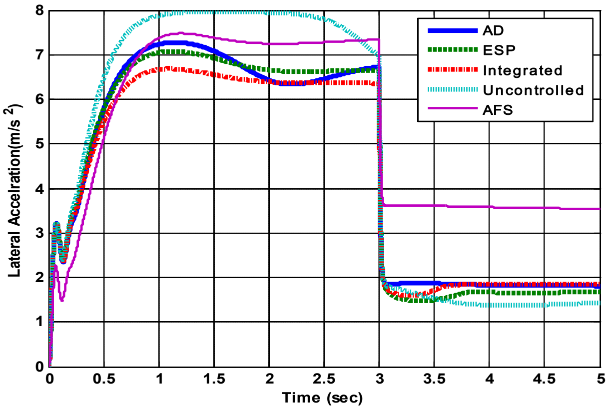

Figure (12): lateral acceleration in the maneuver of sudden arrival to low friction road

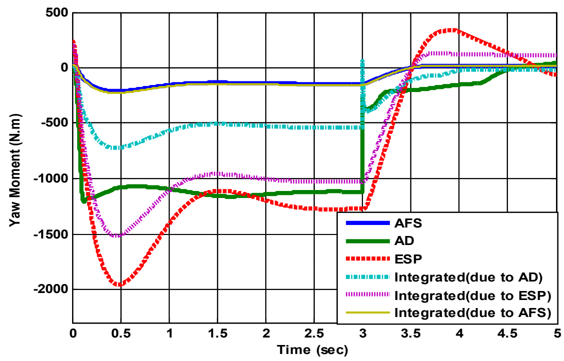

Figure (13): Yaw moment applied to the vehicle in the maneuver of sudden arrival to low friction road

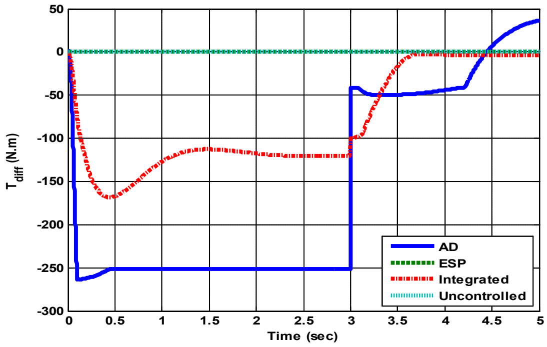

Figure (14): Differential clutching torque in the maneuver of sudden arrival to low friction road

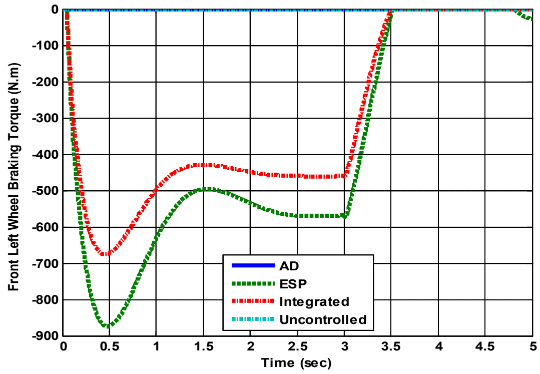

Figure (15): Braking torque applied to front left wheel in the maneuver of sudden arrival to low friction road

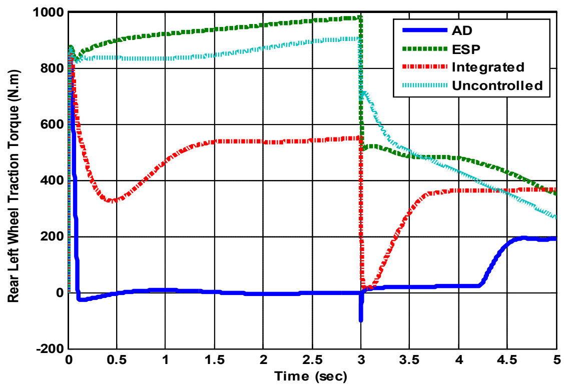

Figure (16): Traction Torque applied to rear left wheel in the maneuver of sudden arrival to low friction road

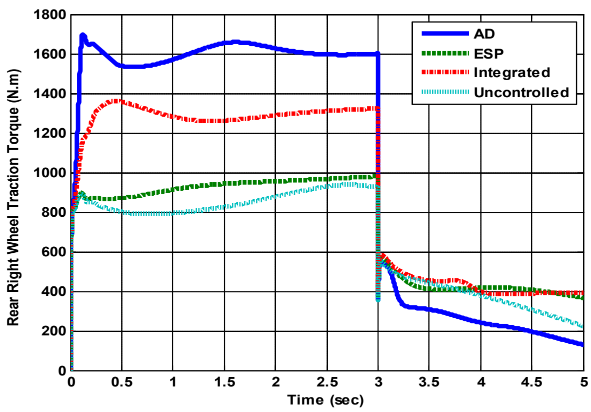

Figure (17): Traction Torque applied to rear right wheel in the maneuver of sudden arrival to low friction road

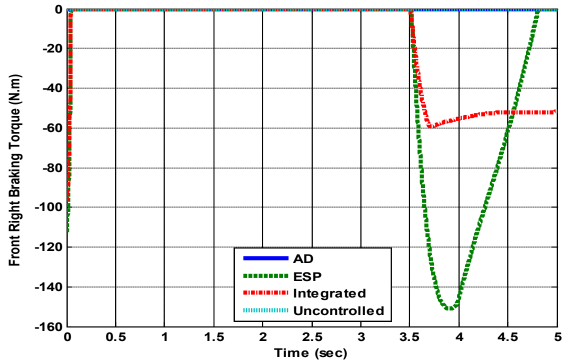

Figure (18): Braking torque applied to front right wheel in the maneuver of sudden arrival to low friction road

Interpretation of results:
According to the simulation carried out in this paper, it is concluded that the integrated control system designed in the paper is able to maintain the stability and to follow the desired values of yaw rate and side slip angle of the vehicle in all motor and road conditions. According to the presented diagrams, each controller has the best performance in certain situations. Due to the high cost of active steering system in vehicle and its subtle role in the improvement of lateral dynamic behavior, using this system is actually uneconomic. ESP system might face problems in the maneuver of $\mu\text{–}Split$. However, active differential covers this fault. Furthermore, active differential has a significant effect on the improvement of vehicle longitudinal dynamics especially in high speeds. Therefore the researcher suggests using an integrated controller as a combination of active differential and ESP which not only improves the vehicle longitudinal dynamics but also is able to stabilize the vehicle in difficult maneuvers.

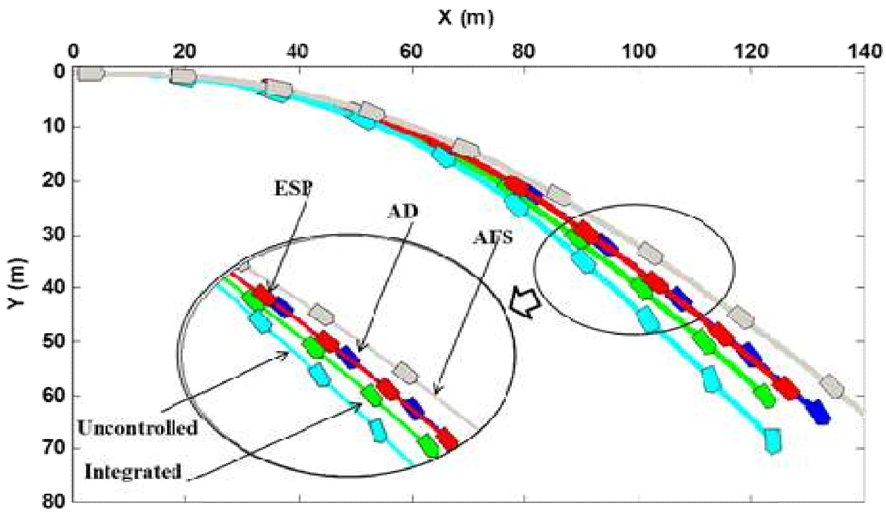

Figure (19): the path of vehicle movement in the maneuver of sudden arrival to low friction road

CONCLUSION
In this paper, an intelligent integrated control system is designed for the active front steering and direct yaw moment control so that its direct yaw moment creates a difference in front wheels via the active differential located in the rear axle and braking. Therefore, at first the active front steering system and direct yaw moment control and different methods to apply yaw moment in vehicles were introduced. Then, the integrated control system, and the reasons to use it and its advantages in the improvement of vehicle handling and stability, and also the works carried out on the integrated controller design were reviewed.
In this paper, in order to design an intelligent integrated controller for active front steering and yaw moment resulting from active rear differential and front wheels brake, the following innovations have been introduced:

- Since the desired value for vehicle must be consistent with its real nonlinear behavior in normal driving conditions, the desired values obtained from linear two-degree-of-freedom and even four-degree-of-freedom models are not close to the real behavior of vehicle; therefore, in this paper, in order to create the desired values of the rate of yaw and the side slip angle of vehicle the steady state of nonlinear six-degree-of-freedom model has been used and the values are presented as two-dimensional procedure based on longitudinal velocity and steering angle.
- In order to consider all dynamic aspects of the vehicle in controller design, the six-degree-of-freedom equations have been linearized using small perturbation technique.
- In order to design top and bottom layer controllers the optimum control – which is an appropriate control method for uncertain systems- has been used. In this research, the top layer controller directly determines the values of differential clutching torque, steering angle correction, and yaw moment resulting from front wheels braking.
- In order to study the effect of controller on longitudinal dynamics in all maneuvers the driver intends to increase the longitudinal velocity. Therefore, accelerating maneuvers have been selected.
- In order to reduce slip in driving wheels, the gas pedal is controlled separately.
- Active differential is effective in high speeds when the engine torque is high. Therefore, the use of this system is not affordable in urban passenger cars.
- While DUC system is able to control the vehicle lateral dynamics in all lateral accelerations, the performance of AFS system at high acceleration maneuvers reduces dramatically due to the saturation of front tires lateral force.
- Integrated control system in addition to being able to provide vehicle stability in difficult maneuvers does not lead to the destruction of longitudinal dynamics.

REFERENCES

1. Sharp R.S, “Relationship between the Steady-Handling Characteristics of Automobiles and Their Stability”, Mechanical Engineering Science, Vol.15, 1973, p. 326.
2. Rengaraj, C. and Crolla, D., Integrated Chassis Control to Improve Vehicle Handling Dynamics Performance, SAE Technical Paper, 2011, p.958
3. R. Rajamani, Vehicle Dynamics and Control. New York: Springer, 2006.
4. Shamim, R., Islam, M., and He, Y., A Comparative Study of Active Control Strategies for Improving Lateral Stability of Car-Trailer Systems, SAE Technical Paper, 2011, p.959.
5. Tandy, D., Pascarella, R., Ault, B., Coleman, C. et al., "Steering and Handling Performance During a Full Tire Tread Belt Separation, SAE Int. J. Passeng. Cars – Mech. Syst., 2011, p.806
6. Rizzo, M., Song, D., Klingler, T., and Howland, D., Light Vehicle Dry Stopping Distance - Vehicle Speed Correction, Tire Burnish, and Surface Friction Correction, SAE Int. J. Passeng. Cars – Mech. Syst. 4(1):, 2011, p.763.
7. Laghari, N., Simulation Analysis of the Effect of Tire and Suspension Non-Linearities on the Handling Response of Rigid and Articulated Vehicles, SAE Int. J. Passeng. Cars – Mech. Syst. 4(1): 2011, p.722.
8. J. R. Ellis, Vehicle Handling Dynamics. London, UK: Mechanical Engineering Publications, 1996.
9. T. Tomari, A. Mori, Y. Shibahata, Development of SH-AWD Based on DC (Direct Yaw Control) Concept, 8th International Symposium on Advanced Vehivle Control (AVEC), Taipei, Taiwan, August 20-24, 2006.
10. E. Esmailzadeh, A. Goodarzi, G.R. Vossoughi, “Optimal yaw moment control law for improved vehicle handling, Mechatronics, vol 13, pp. 659-675, 2003.
11. Bulman DN and Leadbetter Lt NA, Vehicle Handling: Steady-State Theory and Practice, IMechE, Automotive Engineer, 1979, p. 29.
12. Deur, J., Hancock, M., and Assadian, F., Design and Comparative Study of Yaw Rate Control Systems with Various Actuators, SAE Technical Paper, 2011, p.
13. G Lechner, H Naunheimer, J Ryborz, “Automotive transmissions: fundamentals, selection, design and application”, Springer, 2009
14. J. Y. Wong, Theory of Ground Vehicles, 4th edition. New York: Wiley, 2008.