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THE INFLUENCE OF ELECTRONIC STABILITY CONTROL, ACTIVE SUSPENSION, DRIVELINE AND FRONT STEERING INTEGRATED SYSTEM ON THE VEHICLE RIDE AND HANDLING

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Ahmed Elmarakbi [α], Chandrasekaran Rengaraj [σ], Alan Wheatley [ρ] & Mustafa Elkady [ω]

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I. Introduction

Mechatronics and active control systems are playing an ever increasing role in automobiles. Modern vehicles typically include more than 40 actively controlled systems that play a major role in vehicle directional stability, ride comfort and safety. At present these systems generally work independently but it is widely accepted that integration of these stand-alone systems will lead to improved vehicle dynamic performance. Additional benefits include cost and weight reductions and reduced sensor requirements.

Both the automotive industry and the end users will benefit directly from this research. However, successful integration of such control systems is still largely in the research phase. Previous studies have identified that these systems were originally developed independently to perform specific tasks and some Systems do co-exist, Junjie et al. (2006), Karbalei et al. (2007), and Kazuya et al. (2000). Researchers have succeeded in the successful integration of several systems, March and Shim (2007); however, potential conflicts are still a problem. Complete integration of many sub-systems is still a real technical challenge.

The overall aim of this research is to develop new control strategies/algorithms to enable successful integration of a subset of vehicle control systems. However, this paper focuses primarily on the methods of improving vehicle stability and emergency handling through the integration of four specific vehicle control systems: Active Front Steering (AFS), Active Suspension (AS), brake-based Electronic Stability Control (ESC) and driveline based Variable Torque distribution (VTD) system.

There are many ways to compare the performance improvement obtained by an integrated chassis controller against its standalone counterpart. Few of the techniques include comparing the reduction in energy consumption, reduced cost, less/modular parts, improvement in performance variable etc. In this paper, the improved performance objectives established from using the integrated chassis control approach are defined as a reduction in yaw rate and vehicle side-slip angle that lead to better handling capabilities.

The main aim of this paper is to investigate the influence of integration of vehicle dynamics control systems by proposing new control architecture to integrate the braking, steering, suspension and driveline. The active control systems investigated include brake-based electronic stability control (ESC), active suspension (AS) and active front steering (AFS) and variable torque distribution (VTD). The paper is organized as follows. In the vehicle modeling section, a detailed passive vehicle dynamics model with nonlinear tires suitable for combined slip and transient conditions is developed in Matlab/Simulink environment along with the dynamics of steering, braking, suspension and driveline systems.

In the standalone control systems section, the development of standalone control system models of active front steering, active suspension, a brake based

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electronic stability control and a variable torque distribution system are discussed. Various possible integrated control strategies amongst those systems in consideration are analysed and investigated in the integrated control system section. This section also explains a new integrated control strategy (ICC) developed from the results of the analysis and implemented then in MATLAB/Simulink. Finally the conclusions based on the new ICC strategy are presented.

II. Vehicle Modelling

A detailed study on various vehicle models available in literature was conducted for their suitability in this research. Considering the vehicle operations in a wide lateral acceleration range and interactions amongst various degrees of freedom (DoF) of vehicle, a non-linear vehicle handling model with 6 DoF for the chassis has been developed for this research. The ISO vehicle axis system is assumed throughout the modelling process. It is assumed that the steering angles of both front wheels are considered identically, the effect of un-sprung mass is only considered in the vertical direction and ignored in the vehicle's lateral and longitudinal directions and the tires and suspension remain normal to the ground during vehicle maneuvers. The chassis equations of motion based on Newton's laws can be derived as follows:

$$m_v\left(\dot{V}_x - V_y\dot{\psi} + V_z\dot{\theta}\right) = F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr} - (F_{yfr} + F_{yfl})\delta \quad (1)$$

$$m_v\left(\dot{V}_y - V_z\dot{\varphi} + V_x\dot{\psi}\right) = F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr} + (F_{xfl} + F_{xfr})\delta \quad (2)$$

$$m_s\left(\dot{V}_z - V_x\dot{\theta} + V_y\dot{\varphi}\right) = F_{sfl} + F_{sfr} + F_{srl} + F_{srr} - m_s g \quad (3)$$

$$J_r\ddot{\varphi} = \frac{t}{2}\left(F_{sfl} + F_{srl}\right) - a\left(F_{sfl} - F_{sfr}\right) - h\left(F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr}\right) \quad (4)$$

$$J_p\ddot{\theta} = b\left(F_{srl} + F_{srr} - F_{sfr} - F_{srr}\right) + h\left(F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr}\right) \quad (5)$$

$$J_y\ddot{\psi} = a\left(F_{yfl} + F_{yfr}\right) - b\left(F_{yrl} + F_{yrr}\right) + \frac{t}{2}\left(F_{xfr} + F_{xrr} - F_{xfl} - F_{xrl}\right) \quad (6)$$

Figure 1 shows the schematic of the vehicle model used in this study. The vehicle model is divided into sub-models that describe the wheel, brake, suspension and steering dynamics.

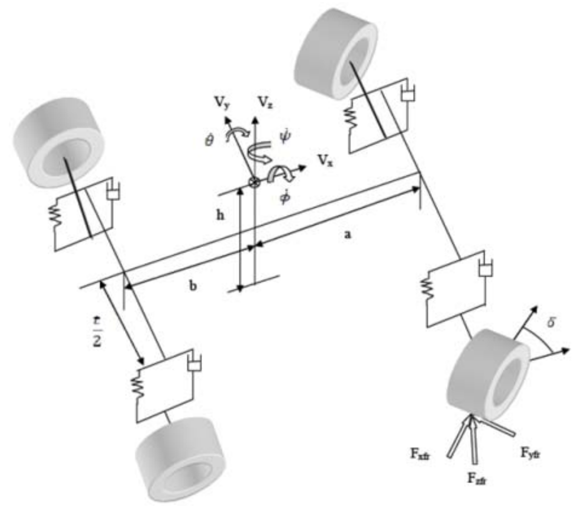


Fig. 1 : Vehicle Model

The dynamics of the tire-road interaction are dependent on the lateral and longitudinal tire slips. The lateral tire slip angles for each wheel can be calculated as follows:

$$\alpha_i = \tan^{-1}\left(\frac{V_y + a\dot{\psi}}{V_x \mp \frac{t}{2}\dot{\psi}}\right) - \delta; \quad \alpha_j = \tan^{-1}\left(\frac{V_y - b\dot{\psi}}{V_x \mp \frac{t}{2}\dot{\psi}}\right) \quad (7)$$

The component of the vehicle velocity of the wheel centre that is parallel to the wheel vertical plane is given as

$$V_i = \cos(\alpha_i)\sqrt{(V_x \mp \frac{t}{2}\dot{\psi})^2 + (V_y + a\dot{\psi})^2} \quad (8)$$

$$V_j = \cos(\alpha_j)\sqrt{(V_x \mp \frac{t}{2}\dot{\psi})^2 + (V_y - b\dot{\psi})^2} \quad (9)$$

The longitudinal wheel slip is defined as

$$\lambda_{ij} = \begin{cases} \dfrac{R_w\omega_i - V_{xi}}{R_w\omega_i}, (driving) \\ \dfrac{V_{xi} - R_w\omega_i}{V_{xi}}, (braking) \end{cases} \quad (10)$$

Capturing the tire behavior is probably the most difficult and important problem to tackle while building a vehicle model as realistically as possible. In the past a lot of different models have been created to solve this problem. The most realistic models are the most complicated but probably they are not useful in every kind of research. On account of our objectives, a too simple model is not applicable because it can provide correct results only if the slip angles are very small, but it cannot represent for example the forces the tires transfer during an emergency handling manoeuvre. For this reason a semi-empirical model usually called

"Magic formula", suggested by Pacejka and Besselink, (1997) is chosen which proved to be a good model but not too complicated. For simplification the camber has been set to zero in the current vehicle model. The general equation of the tire model is

$$y(x) = D\sin[C\tan^{-1}(Bx - E(Bx - \tan^{-1}(Bx)))] \qquad (11)$$

Where $y(x)$ is F_x and F_y, respectively, if x is λ or α. The tire forces generated using the above equations in longitudinal and lateral directions are a function of pure slips in their respective directions. But in reality, these tire forces are generated as a function of combined slip that exists during typical combined braking and cornering situations such as braking before entering a corner and accelerating before exiting it. Weighing functions G as described by Bakker (1987) are introduced which when multiplied with the original pure slip functions produce the interactive effects of longitudinal slip on F_y and lateral slip on F_x. The cosine version of the magic formula is used to represent the hill shaped weighing function, G:

$$G = D\cos[C.\tan^{-1}(Bx)] \qquad (12)$$

The combined side force is described by the following formulae:

$$F_y = G_{yk} \cdot F_{yo} + S_{vyk} \qquad (13)$$

Where, S_{vyk}, the effect due to ply-steer is assumed in this paper to be zero to reduce the complexity. The function G_{yk} is used as described by Rajamani and Hedrick (1995). And, the combined side force is described by the following formulae:

$$F_x = G_{x\alpha} . F_{xo} \qquad (14)$$

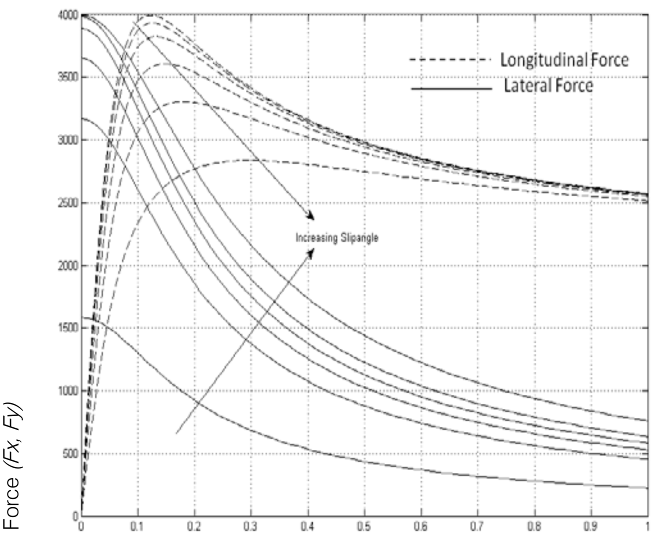
where G_{yk} and $G_{x\alpha}$ are described as follows:

$$G_{yk} = \frac{\cos\left[C_{yk}\tan^{-1}\left(B_{yk}k_s\right)\right]}{\cos\left[C_{yk}\tan^{-1}\left(B_{yk}S_{Hyk}\right)\right]} \qquad (15)$$

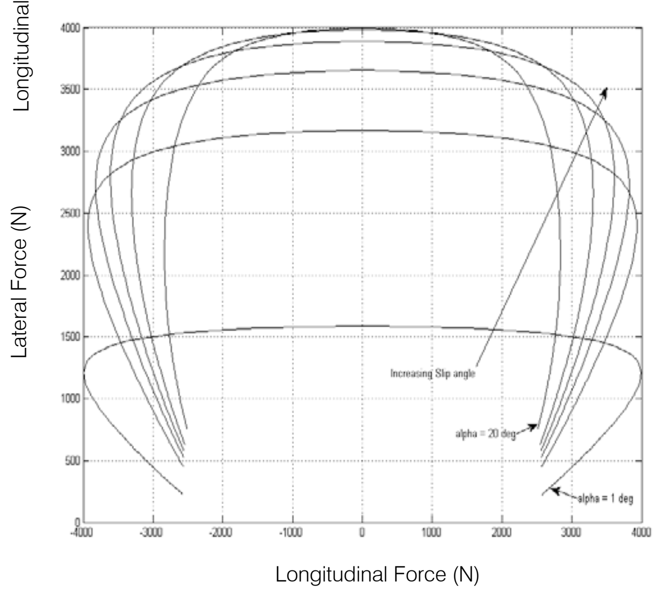
$$G_{x\alpha} = \frac{\cos\left[C_{x\alpha}\tan^{-1}\left(B_{x\alpha}\left(\alpha + S_{Hx\alpha}\right)\right)\right]}{\cos\left[C_{x\alpha}\tan^{-1}\left(B_{x\alpha}S_{Hx\alpha}\right)\right]} \qquad (16)$$

A detailed description of these weighing functions can be found in Bakker (1987). Figure 2 show the longitudinal and lateral tyre forces in combined braking and cornering conditions used in this paper obtained using the above mentioned equations. The effect of tire force lag (Rajamani and Hedrick, 1995) is also taken into account according to the following equation.

$$\left.\begin{aligned} \frac{L_{Fx}}{V_x}\left(\frac{d}{dt}\left(F_x\right)\right) + F_x &= F_{xss} \\ \frac{L_{Fy}}{V_x}\left(\frac{d}{dt}\left(F_y\right)\right) + F_y &= F_{yss} \end{aligned}\right\} \qquad (16)$$



Combined longitudinal and lateral tyre force vs. slip ratio



Tire forces during combined braking and cornering

Fig. 2 : Tyre forces in combined slip -friction ellipse

The equation of motion for each wheel in the wheel dynamics model is defined as:

$$I_w\dot{\omega}_{ij} = T_{ij} - F_{xij}R_w \qquad (17)$$

The steering system modeled in this work has a hydraulic power steering mechanism. The input for the steering system is the angle of the steering wheel and steering column, while the output is the position of the rack, which determines the angle of the front wheels. There is a mechanical connection between the rack and the steering column with a pinion gear, which converts the rotational motion of the steering column to translational motion of the rack to turn the wheels.

The power assistance is provided by a hydraulic piston attached to the rack. A torsion valve determines which side of the piston receives pressurized hydraulic fluid. This torsion valve is attached to the steering column. The difference between the angular position of the steering wheel and the angular position of the pinion determines the fractional opening of the torsion valve. If the angular difference is positive, the pressure is applied to one side of the piston, and if the angular difference is negative, the pressure is applied to the other side of the piston. The power assistance continues until the difference between the steering wheel position and pinion position is approximately zero. The steering power assist curve is shown in figure 3.

The hydraulic brake system considered in this study is built upon a standard braking model. The standard passive brake system considered for this study consists of a mechanical brake pedal, a servo brake booster, a master cylinder, a hydraulic brake caliper and the friction pad. The brake mechanics considered here are explained as follows. The mechanical brake input is amplified by the servo booster. This is further amplified and converted to a hydraulic pressure called line pressure/supply pressure, which is fed through the brake lines. The line pressure is further amplified and converted to mechanical actuation at the brake calipers. This force moves the friction pads against the rotating wheel disc. The effect of pipe friction is taken into account in line with the real world brake dynamics.

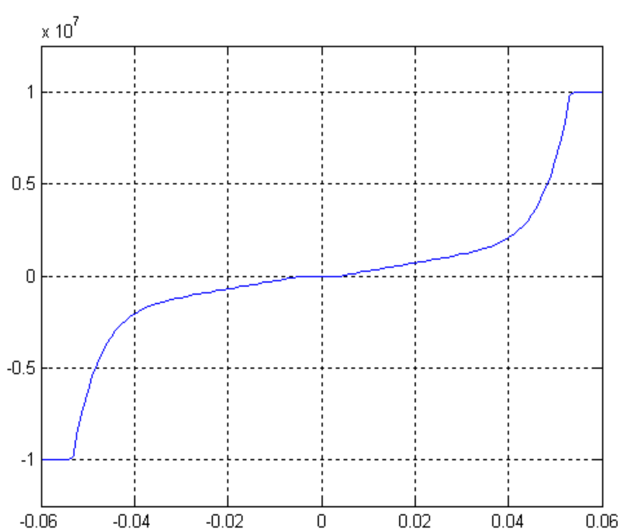


Fig. 3 : Hydraulic steering power assist curve

The resulting brake system model assumes non-laminar flow through restriction as described in the following equation by Fletcher et al. (2004).

$$Q = C_d A \sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{18}$$

III. Development Of Automotive Toolbox In Matlab/Simulink

Simulation of dynamic systems such as vehicles is a complex and time consuming task. Most of the time the modeling tasks need to be repeated in order to perform system analysis such as "if-what" scenarios. Developing a toolbox will modularise the whole modeling process and reduce the model development and analysis time. Rodic (2003) developed a specialized piece of commercial software for modeling, control design and simulation of road vehicles. Poussot-Vassal et al. (2007) developed a unique toolbox during the course of his doctoral research to analyze active suspension and active brake systems. The automotive toolbox developed in this paper provides Simulink models and Matlab tools for vehicle dynamic simulation, analysis and development of vehicle dynamic control systems. It has modular Simulink models for quarter car, extended quarter car, half car for roll and pitch, vertical vehicle model, full vehicle model, linear, nonlinear tire models and other vehicle subsystem models.

This toolbox provides a flexible environment for vehicle dynamic research. It contains libraries with Simulink graphical blocks and Matlab functions, which can be connected to build vehicle models. Using this toolbox, it is also possible to subdivide the whole vehicle model into a number of smaller vehicle subsystems, which can be arranged in a neat way and validated separately. The use of block-diagrams greatly facilitates computer representation of vehicle dynamic systems.

IV. Vehicle Model Validation

This section describes the validation of the full vehicle model developed in the previous sections. The handling dynamics is evaluated and the simulation results were compared against industry standard software. Any software vehicle models developed needs to be validated either against experimental results or against other proven simulation software results.

The vehicle model developed in this paper is validated against the well-known commercial software CarSim. CarSim is vehicle dynamics simulation software developed by Mechanical Simulation Corporation in Ann Arbor, USA. It is parametric modeling software widely used both in academia and industry to simulate, predict and analyze vehicle dynamic behavior.

The validation methodology consists of three phases:

- Describing the validation test condition and procedures
- Simulation of full vehicle model
- Comparison of simulation prediction with the CarSim vehicle model simulation data

In order to be used in this research the model developed must be capable of evaluating vehicle dynamics both in normal and limit driving situations. Two standard test maneuvers are used to evaluate the vehicle model. First, a step steer input at a constant speed was provided so that it generates a lateral acceleration (latac) of 0.3g, 0.6g and 0.8g respectively. This evaluates the model across all the lateral acceleration range from low to the limit handling. Figures 4-7 present the results for the step steer input, showing the comparison of yaw rate at 0.3g latac, of vehicle side slip angle at 0.3g latac, of yaw rate at 0.6g and 0.8 latac, respectively. It is clearly shown that there are a great match between both CarSim and full vehicle model.

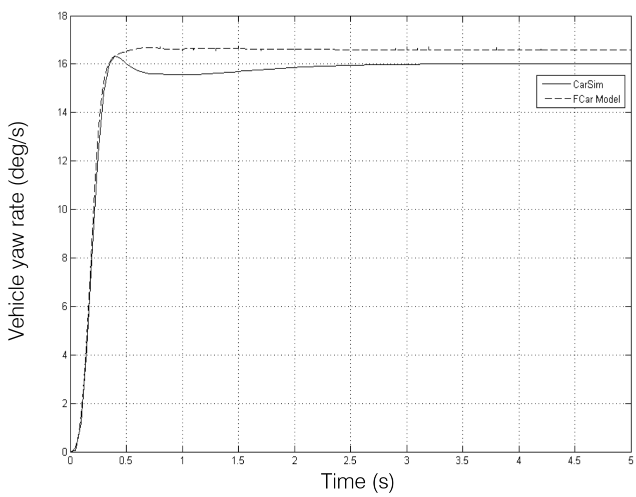


Fig. 4 : Comparison of yaw rate at 0.3g latac

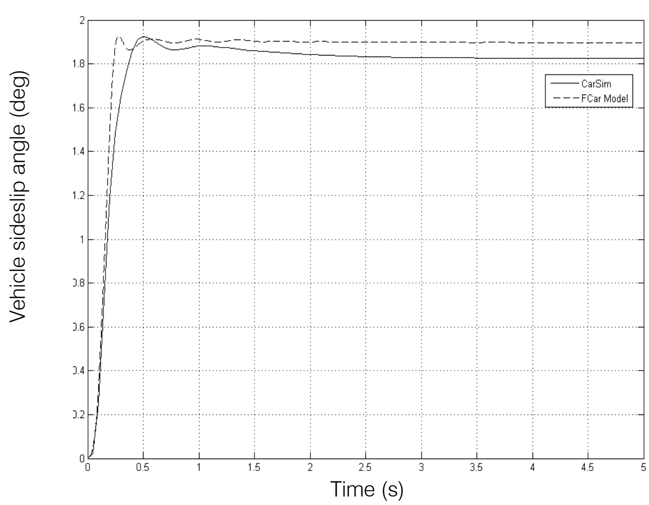


Fig. 5 : Comparison of vehicle sideslip angle at 0.3g latac

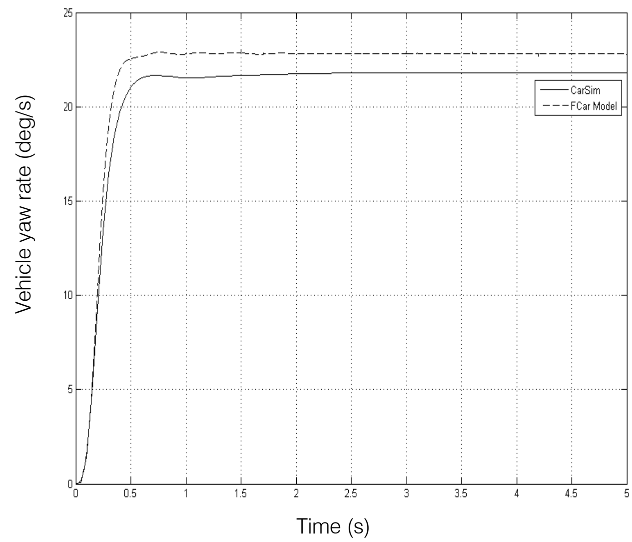


Fig. 6 : Comparison of yaw rate at 0.6g latac

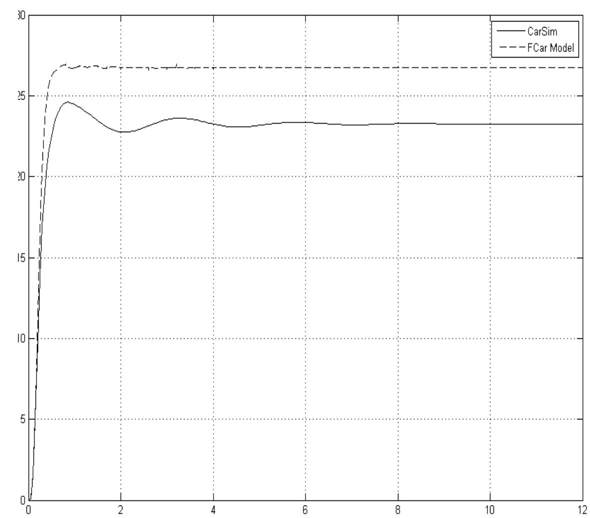


Fig. 7 : Comparison of yaw rate at 0.8g latac

Then a double land change manoeuvre was also performed to validate the vehicle model. The test was performed at a speed of 80km/h on a flat dry surface whose coefficient of friction was 1 (μ =1) First the test was carried out using CarSim software. The vehicle parameters for a D Class Sedan were used. The results were imported to Matlab/Simulink workspace. Then the Full vehicle model was characterised with the CarSim vehicle parameters. The same steering data used to simulate the CarSim model was used as steering input to the full vehicle model. The simulation results of the full vehicle model were plotted along with the CarSim results for comparison. Figures 8-10 show comparisons of the vehicle yaw rate, of the vehicle sideslip angle, and of the vehicle path between CarSim and full vehicle model during an 80km/h double lane change manoeuvre. It is also clearly noticed that there are a great match between both results.

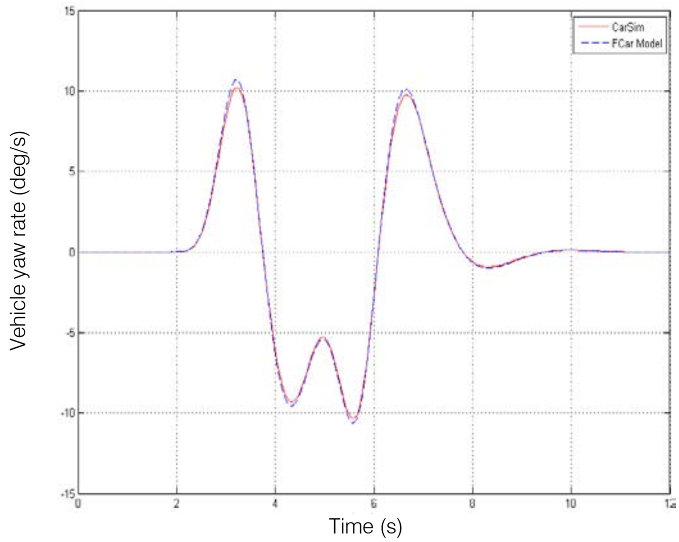


Fig. 8 : Comparison of vehicle yaw rate between CarSim and Full vehicle model (80km/h double lane change maneuver)

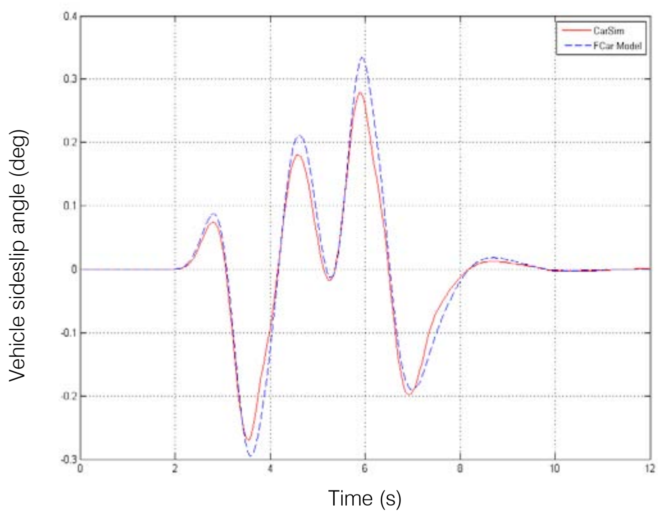


Fig. 9 : Comparison of vehicle sideslip angle between CarSim and full vehicle model during (80km/h double

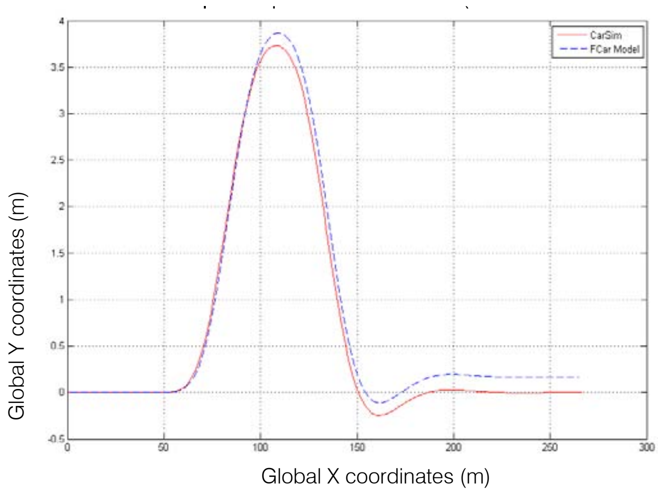


Fig. 10 : Comparison of vehicle path between CarSim and full vehicle model during (80km/h double lane change maneuver)

From these figures, it can be concluded that the responses of full vehicle model developed follows closely the responses of the CarSim vehicle model across various lateral acceleration range. The little deviations observed in the medium and high latac range are largely due to the differences in the suspension kinematics between the models, the nonlinearity in the suspension elements in CarSim and limitations in transferring all the CarSim vehicle parameters in to the Full vehicle model. Moreover, as the CarSim model is validated against real-time experiments, a conclusion can be derived that the full vehicle model is also validated indirectly against experimental results. So it can be concluded that the full vehicle model developed is in par with the widely used commercial software vehicle model and is suitable to use in the vehicle dynamics studies such as integrated chassis control systems.

V. Standalone Control Systems

a) Electronic Stability Control

Electronic stability control is used to stabilize a vehicle by generating an external yaw moment. The three strategies explained in literature to perform this are differential braking, active steering and differential drive torque distribution. In this section of the research, the differential braking, a brake (ABS) based ESC strategy is used. First a fuzzy logic ABS controller was developed and simulated for its performance. Then the ABS controller was extended to develop an ESC controller by additional sensor inputs, like steering angle, yaw rate and sideslip angle and supplemented with an ESC controller algorithm that is capable of enhancing the vehicle stability.

The control architecture as shown in figure 11, is designed to be a hierarchical, two layer control (Rajamani, 2006). The upper controller has the desired objective of ensuring yaw stability control and assumes that it can command any desired value of yaw torque. The lower controller ensures that the desired value of yaw torque commanded by the upper controller is indeed obtained from the differential braking system based on ABS.

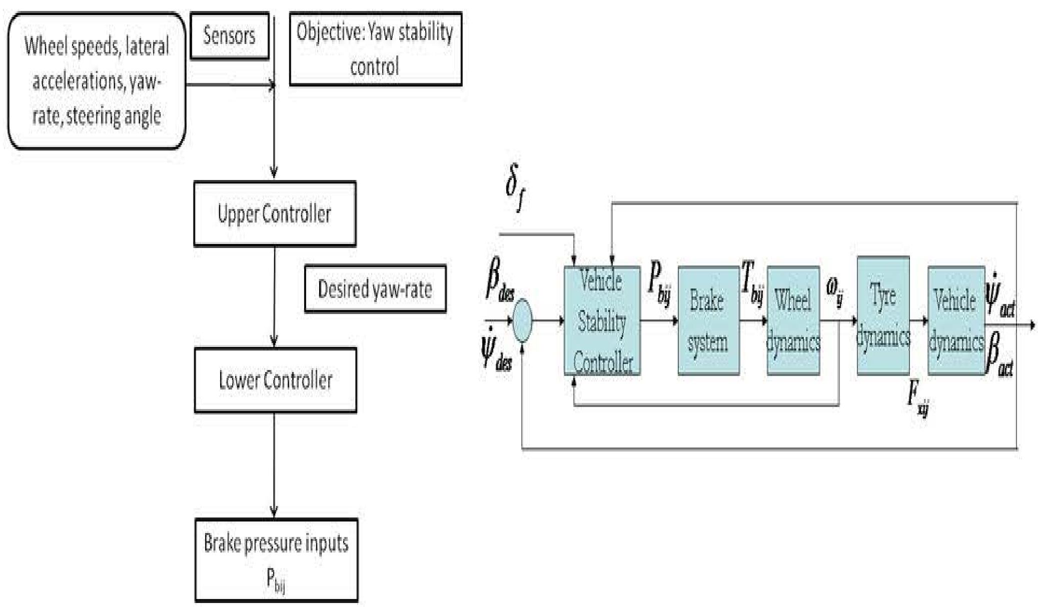


Fig. 11 : Schematic of electronic stability control (ESC)

The lower controller utilizes the wheel rotational dynamics and controls the braking pressure at each of the four wheels to provide the desired yaw torque for the vehicle. Figure 12 describes the relationship between the yaw rate error and its derivative to the controller output. Results of the earlier research show that the brake based ESC are more effective in a wide lateral acceleration range is validated through simulations.

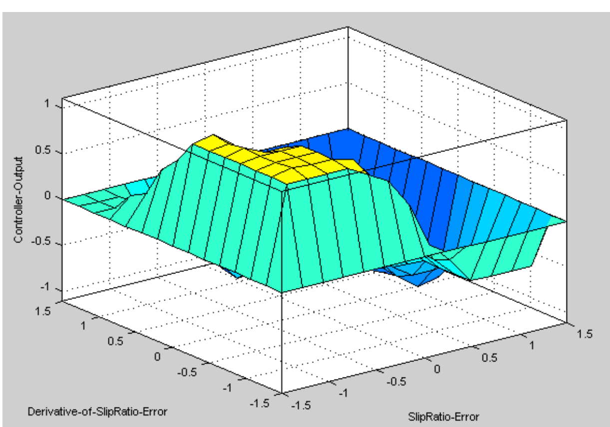


Fig. 12 : Control surface of ESC fuzzy controller

b) Active Suspension

Active suspension in this paper is another active vehicle control system that minimizes the longitudinal and lateral load transfer between the wheels. The AS model used in this research has hydraulic actuators at each wheels as shown in figure 13 that either add or subtract an extra force on each wheels and designed to optimize the normal forces on wheels based on signals from the active suspension controller as a function of various vehicle dynamic states.

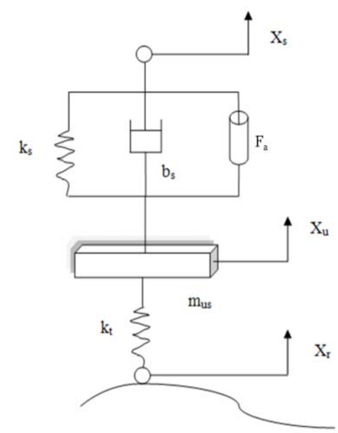


Fig. 13 : Quarter car suspension model

It ensures the tracking of the desired suspension force using PID and Fuzzy logic control strategies. Considerable literature can be found on the dynamics and control of hydraulic actuators for active automotive suspensions.

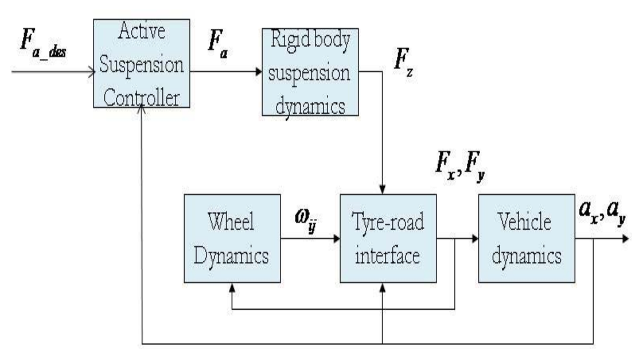


Fig. 14 : Schematic of active suspension system (AS)

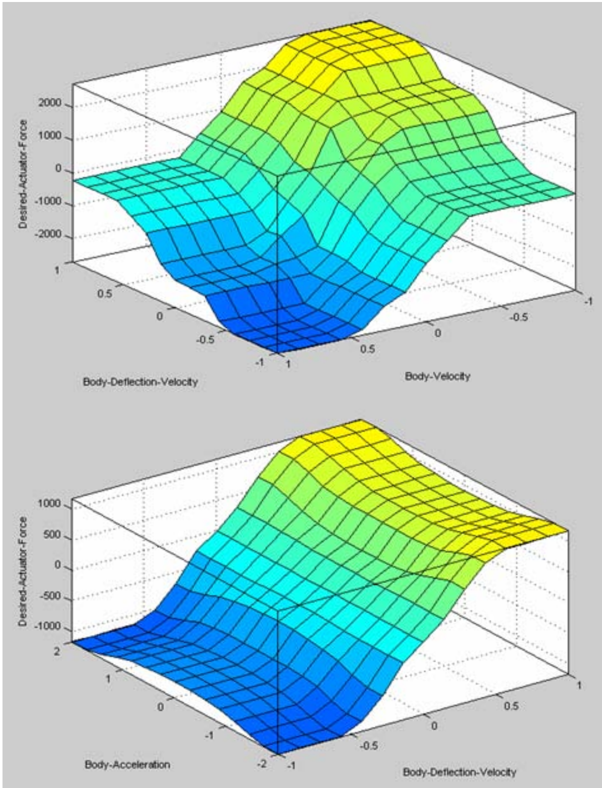


Fig. 15 : Control surface of AS fuzzy controller

The hydraulic actuator dynamics used in this paper includes the dynamics of a spool valve controlled hydraulic actuator model explained by Rajamani and Hedrick (1995). The schematic of the active suspension control strategy followed is described in figure 14 and few of the control surfaces used to develop the fuzzy logic suspension controller are shown in figure 15.

c) Variable Drive Torque Distribution

Another important way to stabilize a vehicle is active drive torque control. One of the recent and widely applied active driveline control techniques is variable drive torque distribution (Pinnel et al., 2004). The objective of this control strategy is to increase vehicle stability and handling capability by suitably distributing the drive torque between wheels. Different drive torque on left and right wheels yield a yaw moment about the vehicle's vertical axis and can be used to stabilize the yaw motion. A two layer control architecture similar to the one shown in figure 11 is used. A PI controller strategy is followed in this case and developed in Matlab/Simulink. The PI controller takes the yaw rate and side-slip angle errors as inputs and returns a control value between 0 and 1 giving the ratio of the drive torque transmitted to the left and right wheels. The control architecture of VTD used in this paper is as shown in figure 16.

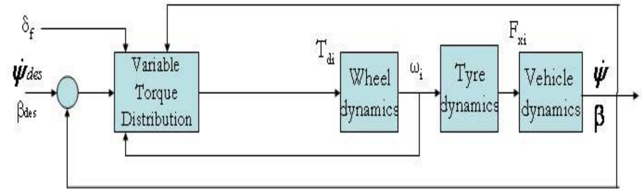


Fig. 16 : Schematic of variable torque distribution control (VTD)

d) Active Front Steering

The active front steering improves the vehicle dynamics in the lateral direction by extending the linear handling region experienced by the driver in a passive vehicle. In a typical vehicle active steering system, the steering angle at the tyre is set in part by the driver through the vehicle classical steering mechanism while an additional steering angle can be set by the AFS controller using hydraulic or DC motor actuators combined with a differential mechanical device. The schematic of the AFS control strategy followed in this paper is described in figure 17.

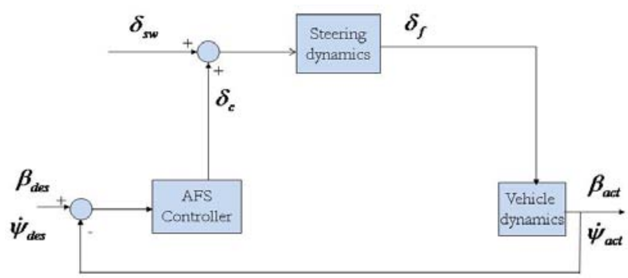


Fig. 17 : Schematic of active front steering (AFS)

Two commonly used control strategies, PID and Fuzzy logic, were used in the development of standalone steering controller in this paper. The vehicle yaw rate and sideslip angle errors (which are the functions of their nominal and actual values respectively) and their time derivatives are fed to the AFS controller to determine the controlled steer addition. Figure 18 shows one of the control surfaces used to develop the fuzzy logic steering controller. Results of earlier research literature in this field are validated here and confirm that performance of active front steering is limited within the linear vehicle handling region, i.e., low to medium lateral acceleration range.

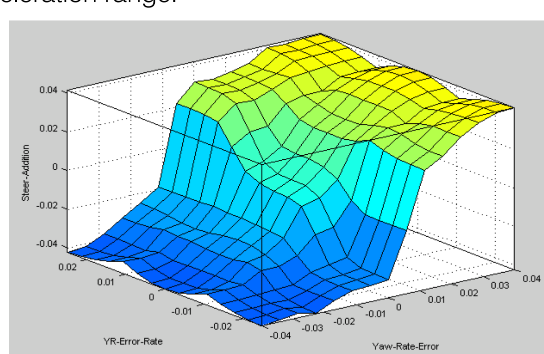


Fig. 18 : Control surface of AFS fuzzy controller

VI. Integrated Control System

The goal of the integrated controller is to cohabit the active steering, active suspension, electronic stability control and variable torque distribution control strategies in order to attain a level of performance that would not otherwise be achievable by them in a stand-alone manner. All the integration strategies explained in this paper are obtained through simulation investigation of the vehicle handling dynamics by performing the ISO3888 double lane change maneuver.

The integrated controller was based primarily on yaw rate and vehicle side-slip angle errors, derived from the actual and desired values. A supervisory fuzzy logic controller (ICC) is used to coordinate among individual control systems. The transition of control authority between individual control systems takes place using a smooth blending function (Junjie et al., 2006) so that better stability and handling performance is maintained throughout the vehicle performance envelop.

To begin the investigation in developing the integrated controller, first, various possible combinations of control strategies between these four systems were analyzed. From the standalone controller systems development in the previous section, we observe that the control objective of ESC is to improve the vehicle lateral stability by minimizing the yaw rate and side-slip angle errors. And the control goal of the active suspension is to optimize the normal wheel forces as a function of various vehicle dynamic states irrespective of ESC's control objective. So, one possible concept of integrating ESC and AS is to optimize the normal wheel forces taking into account the vehicle's yaw rate and side-slip angle errors as well. The schematic of this strategy is described in figure 19 and leads to the co-existence of ESC and AS without affecting the present vehicle handling behavior.

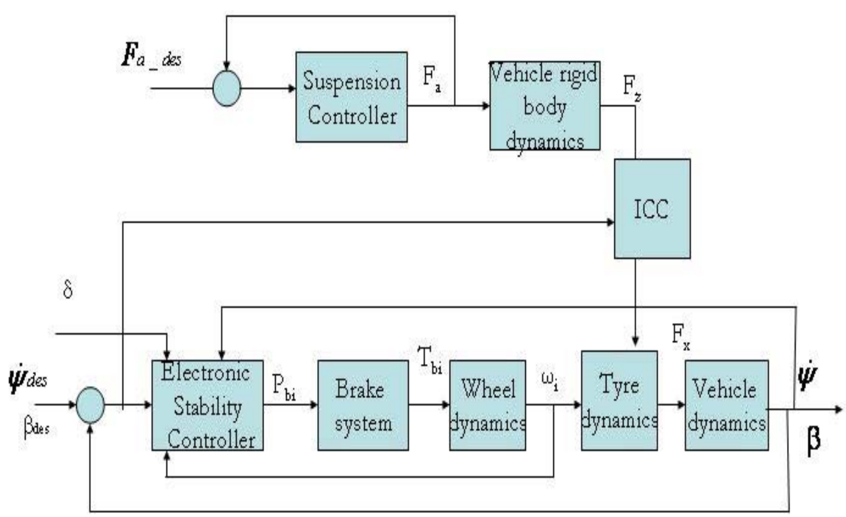


Fig. 19 : Schematic of ICC strategy between ESC & AS

The control objectives of both ESC and AFS are to improve vehicle lateral stability. From the analysis of these two standalone controllers and from the research literature, it is observed that the AFS is more effective in fulfilling its control goal in the linear vehicle handling region whereas the ESC is effective in achieving its control objective in a wide range of lateral accelerations.

Further results of the analysis and literatures show that the ESC is affecting the longitudinal dynamics of the vehicle while improving the lateral stability. The effect of ESC in longitudinal dynamics is perceived as an intrusion in drivers' point of view. Based on the above analysis, an integrated control strategy between ESC and AFS was developed and analyzed through simulation.

This integrated control strategy gives the control authority to AFS during low to medium lateral acceleration range thus providing more comfortable lateral stability compare to ESC. Once the AFS's ability to stabilize the vehicle diminishes, the integrated controller switches the control authority to ESC, in medium to high lateral acceleration range to improve the vehicle stability in critical driving situations. The schematic of above mentioned strategy is shown in figure 20.

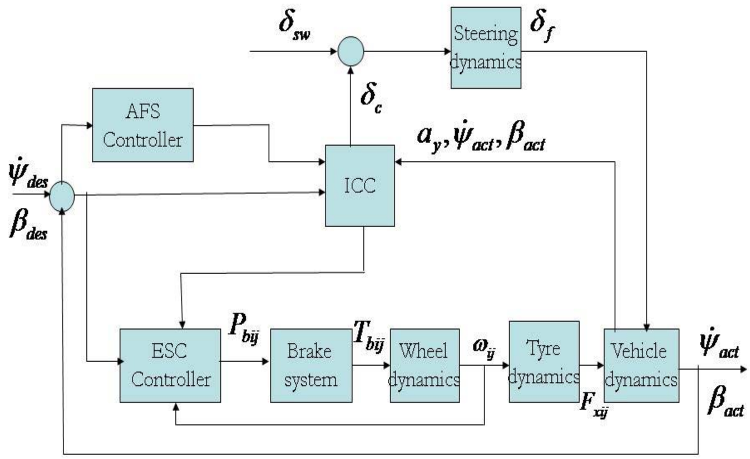


Fig. 20 : Schematic of ICC strategy between ESC & AFS

Similarly an integrated control strategy to improve the vehicle lateral dynamics stability by integrating AFS and AS is also developed and investigated. Here the active suspension optimizes the normal wheel forces taking into account the vehicle yaw rate and side-slip angle errors. Figure 21 describes this control strategy.

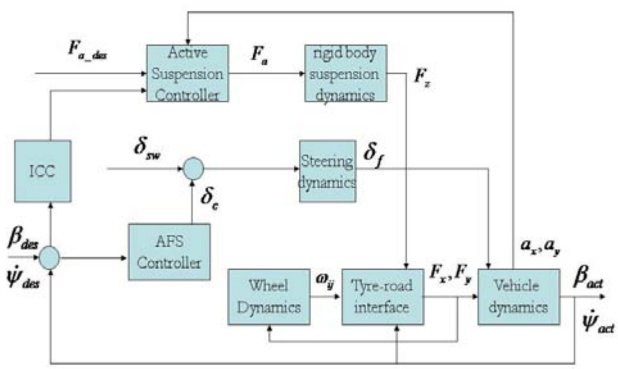


Fig. 21 : Schematic of ICC strategy between AFS & AS

Finally the control objectives of both ESC and VTD are to improve vehicle lateral stability by reducing the yaw-rate and side-slip angle errors. From the development of these two standalone controllers, it is observed that both the ESC and VTD are effective in stabilizing the vehicle in both the linear and non-linear handling regions, but the ESC is intruding and affecting the longitudinal dynamics of the vehicle whereas the intrusion of the VTD in longitudinal dynamics is more acceptable. The schematic of this control strategy is described in figure 22.

Based on these simulated investigations a new integrated control strategy (ICC) is proposed among AFS, ESC, VTD and AS to improve the present vehicle handling performance as follows. Here, the stability of the vehicle is determined using a stability criterion.

This stability criterion is based on a phase-plane characteristics (Vientinghoff, 2008) between the vehicle side-slip angle and its rate of change. The vehicle's stability is classified based on the stability criterion. Then a supervisory fuzzy logic controller is designed that decides the weighting of control authority among the four active control systems to generate the necessary control steer angle, brake pressure, drive torque and suspension force to be added.

The proposed strategy is explained in figure 23 in a schematic form. Based on the new integrated control strategy the AFS, ESC, VTD and AS controllers were evaluated in both standalone and integrated manner and found that the integration of these controllers improves the vehicle lateral stability by reducing yaw rate and side-slip angle during an ISO 3888 DLC maneuver on both high and low frictional surfaces as shown in figures 24-27, respectively.

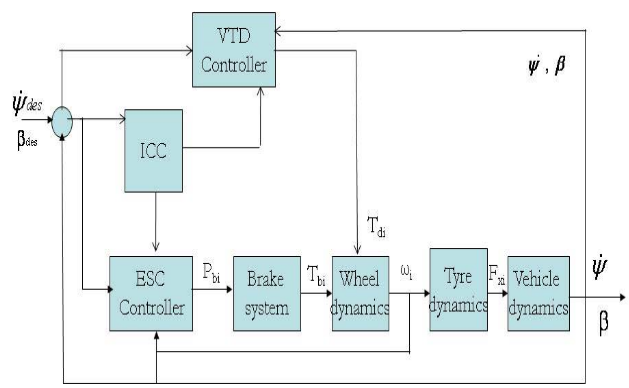


Fig. 22 : Schematic of ICC strategy between ESC & VTD

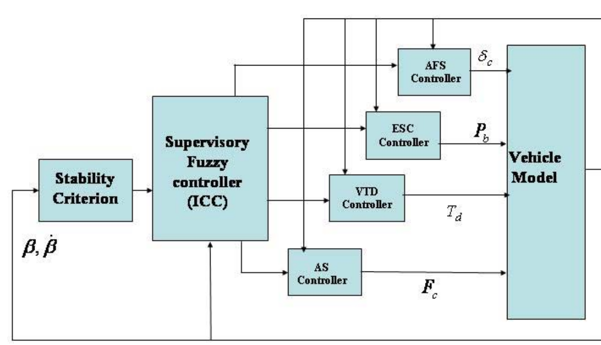


Fig. 23 : Schematic of proposed ICC strategy amongst ESC, AFS, VTD & AS

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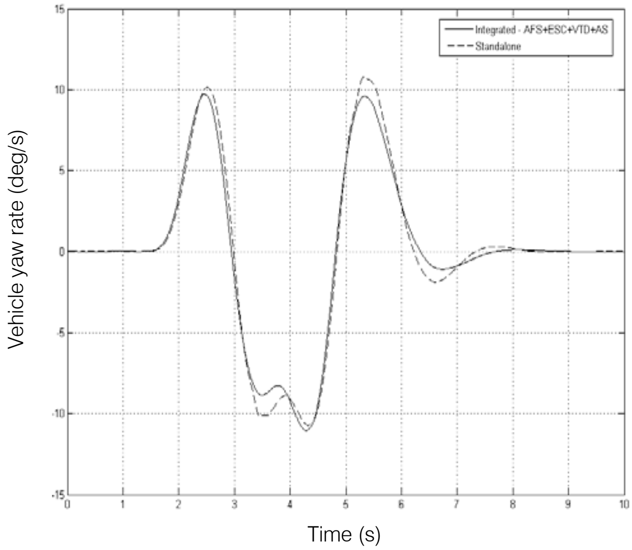

Fig. 24 : Yaw rate – integrated and standalone controllers – high friction surface

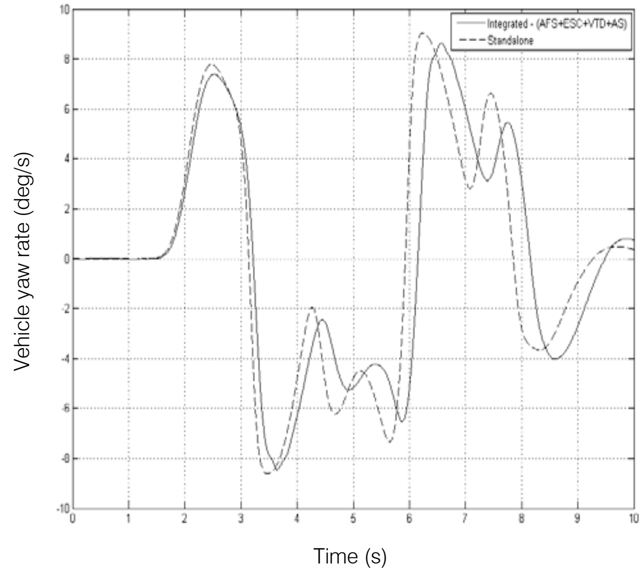

Fig. 26 : Yaw rate– integrated and standalone controllers –low friction surface

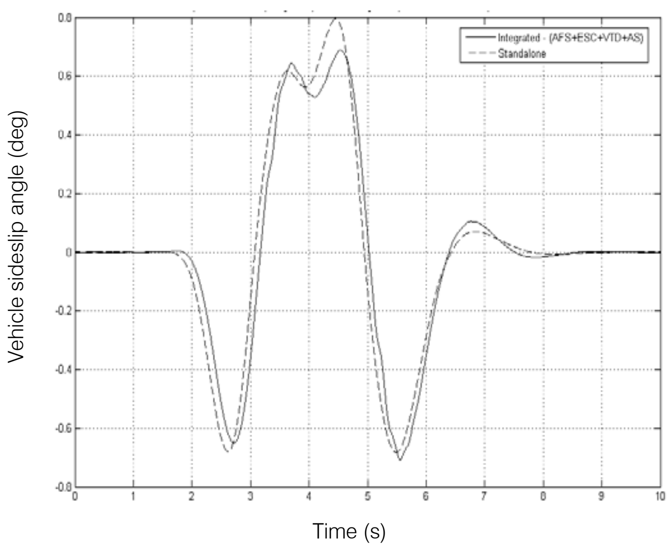

Fig. 25 : Sideslip angle – integrated and standalone controllers – high friction surface

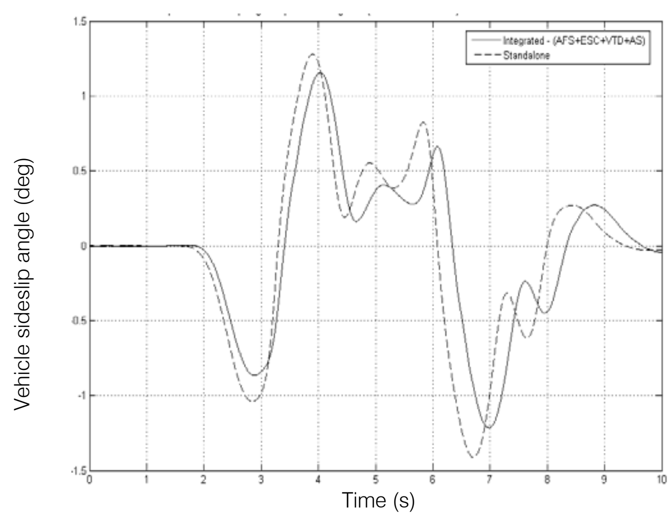

Fig. 27 : Sideslip angle – integrated and standalone controllers –low friction surface

VII. Conclusions

A detailed non-linear vehicle dynamics model is developed and four standalone vehicle control systems AFS, ESC, VTD and AS are modeled and simulated in MATLAB/Simulink. Various possible combinations of integrated strategies between these four systems were investigated through simulations. Based on these investigations a new integrated control strategy is proposed to make these four systems cohabit to improve the present vehicle handling performance. Simulations with the integrated controller demonstrated a significant improvement in the performance objectives. The vehicle motion during an emergency manoeuvre is much improved, showing better handling and characteristics during sudden, high speed maneuvers

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Nomenclature

a	Longitudinal distance of CG from front axle
b	Longitudinal distance of CG from rear axle
B,C,D,E	Pacejka tire parameters
F_{sij}	Suspension force in N
F_{xij}	Longitudinal tire force
F_{xss}	Longitudinal steady state tire force
F_{yij}	Lateral tire force
F_{yss}	Lateral steady state tire force
g	Gravitational constant
h	Height of center of gravity from ground
i	Stands for front/rear *{f, r}*
I_w	Wheel rotational inertia
j	Stands for left/right *{l, r}*
J_r	Roll inertia of the vehicle in km/m^2
J_p	Pitch inertia of the vehicle in km/ m^2
J_y	Yaw inertia of the vehicle in km/ m^2
L_{Fx}	Longitudinal tire force lag
L_{Fy}	Lateral tire force lag
m_s	Sprung-mass of vehicle
m_v	Mass of total vehicle
R_w	Dynamic tire radius
t	Track width of vehicle
V_x	Sprung mass longitudinal velocity at CG
$\dot{V}_x$	Derivative of longitudinal velocity
V_y	Sprung mass lateral velocity at CG
$\dot{V}_y$	Derivative of lateral velocity
V_z	Sprung mass vertical velocity at CG
$\dot{V}_z$	Derivative of vertical velocity
α	Tire lateral slip angle
δ	Steering angle at the front wheels
θ	Vehicle pitch angle at CG

$\dot{\theta}$	Vehicle pitch rate at CG
$\ddot{\theta}$	Derivative of vehicle pitch velocity
λ	Tire longitudinal slip ratio
μ	Coefficient of friction
ϕ	Vehicle roll angle at CG
$\dot{\phi}$	Vehicle roll rate at CG
$\ddot{\phi}$	Derivative of vehicle roll velocity
ψ	Vehicle yaw angle at CG
$\dot{\psi}$	Vehicle yaw rate at CG
$\ddot{\psi}$	Derivative of vehicle yaw velocity
ω	Angular wheel velocity