DINAME 2017 - Proceedings of the XVII International Symposium on Dynamic Problems of Mechanics
A. T. Fleury, D. A. Rade, P. R. G. Kurka (Editors), ABCM, São Sebastião, SP, Brazil, March 5-10, 2017

USE OF INTEGRATED CONTROL TO ENHANCE THE SAFETY OF VEHICLES IN RUN-OFF-ROAD SCENARIOS

Abel A. Castro[1,2], Rafael B. Chaves[1,2], Georg Rill[2], Hans I. Weber[1]

[1] PUC-Rio, abel.arrieta@aluno.puc-rio.br, hans@puc-rio.br

[2] OTH Regensburg, rafael.chaves@st.oth-regensburg.de, georg.rill@oth-regensburg.de

Abstract: In this paper, an integrated vehicle control system (IC) is tested in run-off-road scenarios. The integrated control was used in order to manage the interaction between the Anti-Lock Brake System (ABS), Four-wheel Steering system (4WS), and Electronic Stability Programme (ESP). For all the simulations performed in this work, a virtual test driver, based on fuzzy control, is employed. By receiving the lateral position of an obstacle and vehicle's relative yaw angle, the virtual driver must be able to make an avoiding collision maneuver. Nine fuzzy rules were established considering that the controller must not only perform the avoiding maneuver, but also to minimize the behavior difference between under-steering and over-steering tendencies. The vehicle without any control system is used as reference behavior. For testing the efficiency of the control systems, i.e. ABS, ESP and 4WS, the vehicle perform an avoidance maneuvers in off-road conditions. The ESP and 4WS systems were implemented in order to maintain the vehicle's stability while performing off-road maneuvers. The run-off-road scenarios were built for simulating μ-split in order to evaluate the capability of ESP, 4WS and the IC to improve the vehicle's stability and keep its maneuverability. The ABS model was developed also using fuzzy control. The system receives information regarding the wheels' current velocities for performing a slip-based control. The fuzzy database was composed only by three rules, which were enough to make the system work properly not only on surfaces with normal and reduced friction, but also in μ-split conditions. The 4WS system is used to improve the maneuverability at low velocities and the lateral response and stability at high velocities. The magnitude and orientation of rear steer angle was defined by the magnitude of the front wheels steer angles. The ESP is applied only when the tires are operating in the non-linear region, that means when the driver is no longer able to control the vehicle. A simple switch approach was used to model the ESP, where the yaw angular velocity and side slip angle errors, calculated using a simple handling model, were used as references. The simulations were done in a virtual environment that contains a fully non-linear vehicle model with a real suspension geometry and the TMeasy tire model, also used in commercial software. This environment is also capable to simulate 3D road profiles with different friction conditions and obstacles, e.g. bumps and holes as well as random profiles.

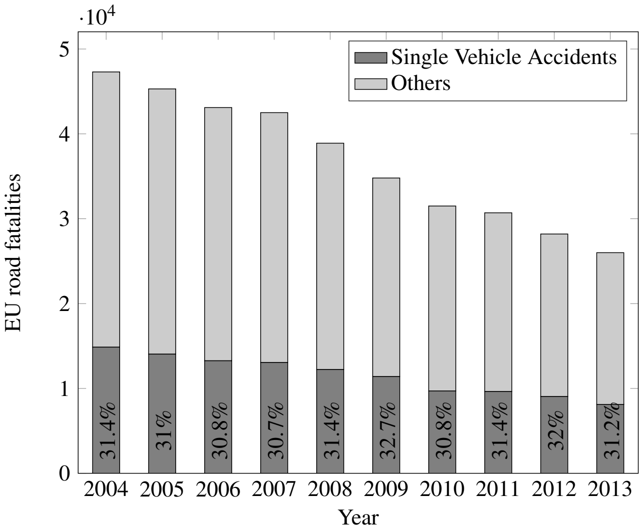
Keywords: Integrated control, non-linear vehicle model, fuzzy control, run-off-road scenarios

INTRODUCTION

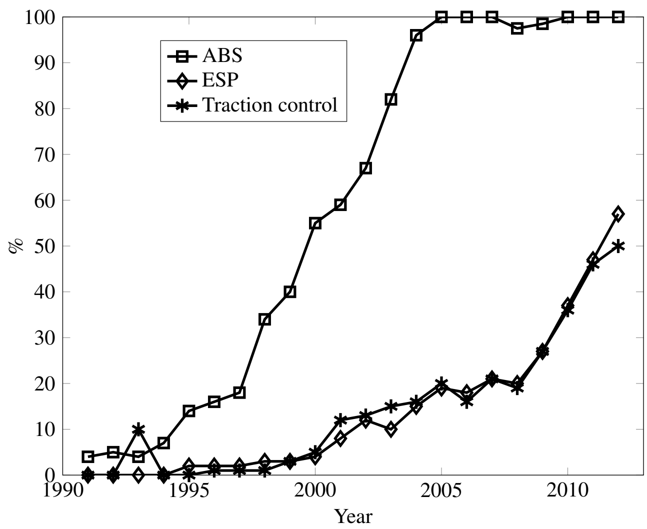
In spite of the decreasing number of Single Vehicle Accidents (SVA) during the last decade, it still represents a high percentage of road fatalities, see Fig 1a. Due to these numbers, there is a constant concern of the society in mobility and traffic safety. Therefore, the development of control system that can reduce the number of road fatalities is of high priority for the automotive industry and engineers.

Since the introduction of the Anti-lock Braking System (ABS) in 1978 by Bosch, the number of electronic systems to assist drivers and improve the vehicle safety is increasing. For instance, the Electronic Stability Program (ESP) is capable of reducing in 49% (Erke Alena, 2008) SVA. Consequently, many companies are fitting its best seller cars with safety technologies (BOVAG-RAI, 2013), see Fig. 1b. Despite of the absence of the Four-Wheel Steering System (4WS) in the statistic shown, this technology, that was a trend during the late of 1980s, has been used again since the twentieth century. This trend is possible because modern and powerful electronic components are present in cars. The 4WS system can improve the maneuverability at low velocities and the vehicle stability at high velocities. These two main advantages make this technology an interesting option to improve, in combination with ABS and ESP, the overall vehicle dynamics.

In instability scenarios, the vehicle does not follow the driver's commands. In these cases, the active safety systems should actuate in order to return the vehicle to a maneuverability state. This task, is done by exploiting the tire limits by the control systems presents in the vehicle. Nevertheless, if these controllers try to exploit the tire potential at the same time, this may lead to a conflict that becomes a worse scenario to the driver. Therefore, it is important to develop a methodology to integrate these safety systems in order to improve the vehicle performance. In a previous study (Castro *et al*, 2015) it was shown that the integration of an Active Front Steering system (AFS) and the ESP can improve the vehicle lateral stability. In the present paper, an integration of the 4WS and the ESP was proposed. This new integrated system was tested in run-off-road scenarios in order to prove its benefits against the non-integrated approach.



(a) Single vehicle accidents and all road fatalities in EU.



(b) Active safety systems in top 50 best selling cars.

Figure (1) Statistics of road accidents in EU and safety systems using by top seller cars.

VEHICLE AND TIRE MODEL

In this paper, two vehicle models are established. The first one, is a three-dimensional model with a non-linear axle kinematics and it is used to describe the vehicle dynamics behavior. The second model, is the well know simple handling vehicle model and is utilized for control design and to calculate the vehicle desired response.

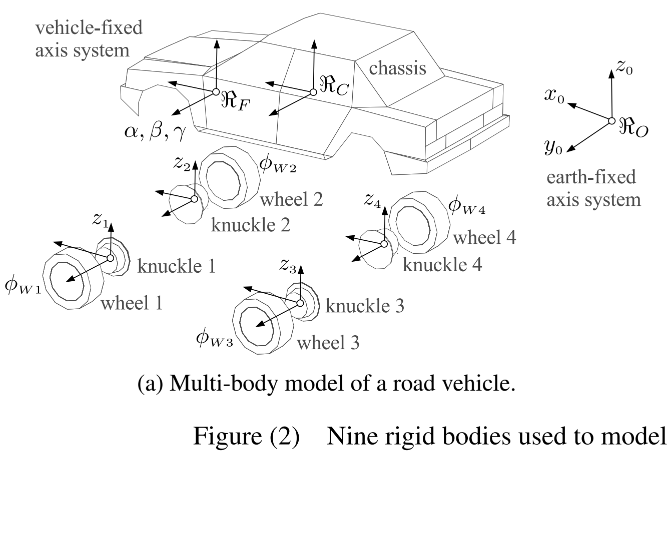
The tires are responsible for generating the main forces to move the vehicle. Therefore, a proper tire model is a crucial part for vehicle dynamic simulation. In order to obtain accurate results in a reasonable time, the TMeasy tire model is employed.

Full non-linear vehicle equations

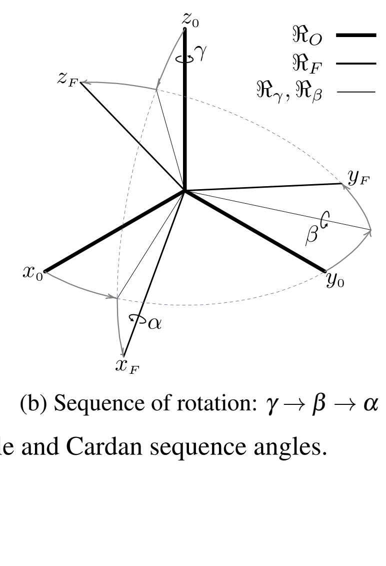
The mathematical model of the vehicle is built applying the multi-body approach. Using this method, it is possible to modeling the vehicle by subsystems as presented in (Rill, 2006) and (Rill, 2011). In this work, nine rigid bodies are used, i.e. 4 knuckles, 4 wheel-tires and 1 chassis, see Fig. 2a. The equations of motion are obtained using Jourdain's Principle, also called the method of virtual power. The relative motion of each body related to another is required to calculate the relative forces between the bodies. For this task, a reference frame and a sequence of rotations need to be defined as shown in Fig. 2b. Finally, the vehicle dynamics is characterized by a set of non-linear first order differential equations.

$$K(y)\,\dot{y} = z, \quad M(y)\,\dot{z} = q(y,z,s,u), \quad \dot{s} = f(y,z,s,u), \tag{1}$$

where y is a vector that collects the generalized coordinates of the vehicle and K is the kinematic matrix used to define an appropriate vector of generalized velocities z. The mass matrix of the multi-body vehicle model is denoted by M. The vector of generalized forces q is function of the input u and the additional states s. Furthermore, the vector s collects the internal states of the dynamic force elements and tire deflections respectively.



(a) Multi-body model of a road vehicle.

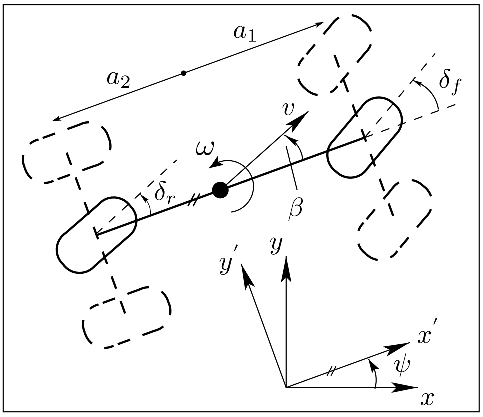


(b) Sequence of rotation: $\gamma \to \beta \to \alpha$

Figure (2) Nine rigid bodies used to model the vehicle and Cardan sequence angles.

Simple handling model

For control design, the desired response of the vehicle to driver's input is necessary. Generally, the simple handling model is used as a reference, see Fig. 3. This model captures some important features of lateral dynamics, e.g. yaw rate $\dot{\psi}$ and the side slip angle β, that are important variable to measure the vehicle stability. According to the assumptions in



- tires operates in the linear region.
- only planar motion.
- no driving or braking is applied i.e, $\dot{v} = 0$.
- δ_f represents the mean of the front steering angles.
- δ_r represents the mean of the rear steering angles.
- δ_f, δ_r and β are assumed to be small.

Figure (3) Simple handling vehicle model and assumptions.

Fig. 3, this handling model has two degrees of freedom (DOF), i.e. lateral and yaw motions, and are described by:

$$m\left(v\omega + |v|\dot{\beta}\right) = F_{yf} + F_{yr} \qquad \text{and} \qquad \Theta\dot{\omega} = a_1 F_{yf} - a_2 F_{yr} \tag{2}$$

Furthermore, the tire lateral forces are described as a function of the cornering stiffness c_{si} and the lateral slips s_{yi} $(i = f, r)$ as follows:

$$F_{yi} = c_{si} s_{yi} \qquad s_{yf} = -\beta - \frac{a_1}{|v|}\omega + \frac{v}{|v|}\delta_f \qquad s_{yr} = -\beta + \frac{a_2}{|v|}\omega + \frac{v}{|v|}\delta_r \tag{3}$$

Finally, rearranging and simplifying (2) and (3), the equations of motion are as follows:

$$\underbrace{\begin{bmatrix} \dot{\beta} \\ \dot{\omega} \end{bmatrix}}_{\dot{x}} = \underbrace{\begin{bmatrix} -\dfrac{c_{sf}+c_{sr}}{m|v|} & \dfrac{a_2 c_{sr} - a_1 c_{sf}}{m|v||v|} - \dfrac{v}{|v|} \\ \dfrac{a_2 c_{sr} - a_1 c_{sf}}{\Theta} & -\dfrac{a_1^2 c_{sf} + a_2^2 c_{sr}}{\Theta|v|} \end{bmatrix}}_{A} \underbrace{\begin{bmatrix} \beta \\ \omega \end{bmatrix}}_{x} + \underbrace{\begin{bmatrix} \dfrac{v}{|v|}\dfrac{c_{sf}}{m|v|} & \dfrac{v}{|v|}\dfrac{c_{sr}}{m|v|} \\ \dfrac{v}{|v|}\dfrac{a_1 c_{sf}}{\Theta} & -\dfrac{v}{|v|}\dfrac{a_2 c_{sr}}{\Theta} \end{bmatrix}}_{B} \underbrace{\begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix}}_{u} \tag{4}$$

where the time derivative of the yaw angular velocity ω and of the vehicle side slip angle β are the output states of the model, and are collected in the vector $\dot{x}$. The cornering stiffness of the front and rear axle are represented by c_{sf} and c_{sr} respectively. The inertia properties of the handling model are represented by its mass m, the moment of inertia around the z axis Θ and the distances, a_1 and a_2, from its center of gravity to the front and rear axle respectively. The vehicle velocity is represented by v and the steering angles δ_f and δ_r are inputs of the simple handling vehicle model.

TMeasy tire model

TMeasy (Rill, 2013), is a semi-empirical tire model that uses a small number of parameters to characterize the tire-road contact phenomena. The main advantages of TMeasy, against other tire models, is a good balance between accuracy and time consuming, and the small number of parameters that it requires. In normal driving maneuvers, e.g. acceleration and deceleration in a curve, the longitudinal slip s_x and lateral slip s_y occur at the same time. Therefore, the combination of slips and thus of the longitudinal and lateral forces should be handled by the tire model. In order to achieve the contribution of longitudinal and lateral slips to the combined slip, with a similar weight, TMeasy performed a normalization process as follows:

$$s = \sqrt{\left(\frac{s_x}{\hat{s}_x}\right)^2 + \left(\frac{s_y}{\hat{s}_y}\right)^2} = \sqrt{(s_x^N)^2 + (s_y^N)^2}, \tag{5}$$

where s_x^N and s_y^N are the normalized slips. Furthermore, the normalizing factor $\hat{s}_x$ and $\hat{s}_y$ take into account the longitudinal and lateral force characteristics and are defined as follows:

$$\hat{s}_x = \frac{s_x^M}{s_x^M + s_y^M} + \frac{F_x^M/dF_x^0}{F_x^M/dF_x^0 + F_y^M/dF_y^0} \qquad \hat{s}_y = \frac{s_y^M}{s_x^M + s_y^M} + \frac{F_y^M/dF_y^0}{F_x^M/dF_x^0 + F_y^M/dF_y^0} \tag{6}$$

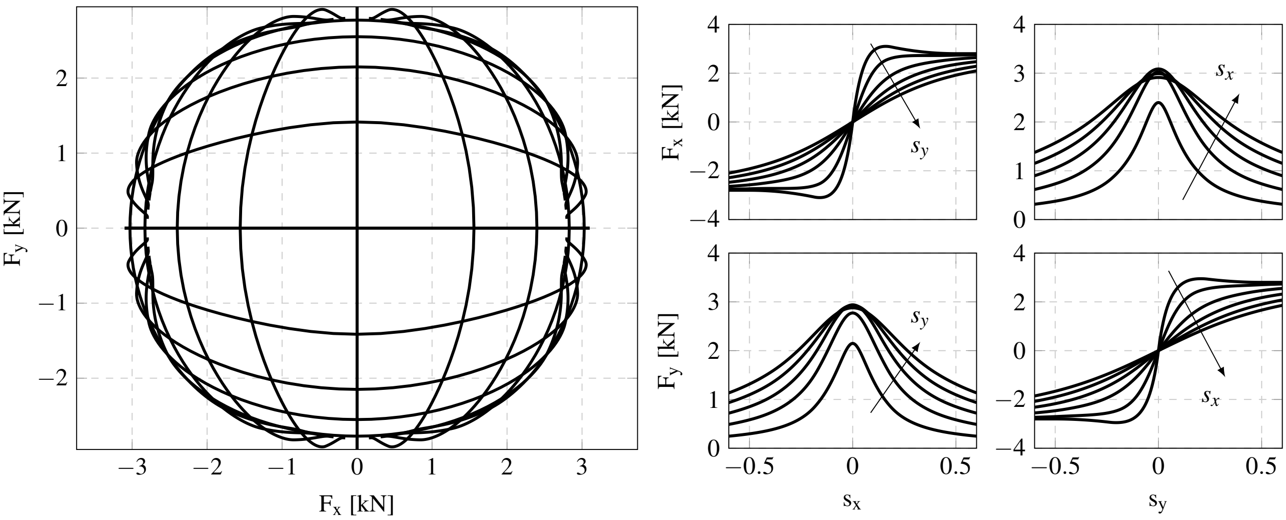
Similar to the curve of longitudinal and lateral forces, the combined force $F = F(s)$ can be defined by their characteristic parameters dF^0, s^M, F^M, s^S, and F^S. These parameters are defined as:

$$
\begin{aligned}
dF^0 &= \sqrt{(dF_x^0 \hat{s}_x \cos\phi)^2 + (dF_y^0 \hat{s}_y \sin\phi)^2} & & \\
s^M &= \sqrt{\left(\frac{s_x^M}{\hat{s}_x}\cos\phi\right)^2 + \left(\frac{s_y^M}{\hat{s}_y}\sin\phi\right)^2} & F^M &= \sqrt{\left(F_x^M\cos\phi\right)^2 + \left(F_y^M\sin\phi\right)^2} \\
s^S &= \sqrt{\left(\frac{s_x^S}{\hat{s}_x}\cos\phi\right)^2 + \left(\frac{s_y^S}{\hat{s}_y}\sin\phi\right)^2} & F^S &= \sqrt{\left(F_x^S\cos\phi\right)^2 + \left(F_y^S\sin\phi\right)^2}.
\end{aligned}
\tag{7}
$$

The angular function ϕ is used to grant a smooth transition from the longitudinal and lateral force to the combined force. Finally, the longitudinal and lateral result forces are derived from the combined force as follows:

$$
F_x = F\cos(\phi) \quad \text{and} \quad F_y = F\sin(\phi) \qquad \text{where} \qquad \cos(\phi) = \frac{s_x^N}{s} \quad \text{and} \quad \sin(\phi) = \frac{s_y^N}{s}. \tag{8}
$$

Fig. 4 shows, (a) tire friction limits and (b) the mutual influence of longitudinal and lateral forces (computed by TMeasy) for a standard commercial tire.



(a) Tire friction limits.

(b) Longitudinal and lateral forces against slip variations.

Figure (4) Typical force curves for a passenger vehicle tire.

CONTROL DESIGN

The ABS, ESP and 4WS control design was done in this section. For all the simulations performed, a full size car is used. Characteristic parameters of the vehicle are defined in Tab. 1. Furthermore, road and off-road scenarios are characterized by friction coefficients of $\mu = 1$ and $\mu = 0.4$ respectively. In addition, the road surface is located 5cm above the off-road surface. Furthermore, an obstacle is placed at a distance of 150m in front of the vehicle. Avoiding this obstacle will force the vehicle to go off-road. Finally, the driver model was done using fuzzy logic and simple rules were used in order to follows a path defined by the road center line. The lateral error and the current orientation of the vehicle are taken into account as a inputs of the virtual driver.

Table (1) Full-size vehicle parameters

Parameter	**Value**	**Units**
Velocity	90	km/h
Chassis mass	1927.8	kg
Chassis inertia	$I_{xx} = 452, I_{yy} = 2645.3, I_{zz} = 2813.5$	$kg \times m^2$
Suspension	Double Wishbone (front and rear)	-
Tires	P265/40 R18	-
Wheel base	2.9	m
Track front	1.53	m
Track rear	1.52	m

Anti-lock Braking System - ABS

The longitudinal force at each wheel depends of the friction coefficient μ and wheel load F_z. In this work, μ is assumed to be unknown, for this reason the slip was taken as control variable. This variable is also used in commercial systems, e.g., ABS system implemented by Bosch. The longitudinal slip s_x is defined by Eq. 9 as follows:

$$s_x = -\frac{(v_x - \Omega r_D)}{|\Omega| r_D + v_N}, \tag{9}$$

where v_x is the vehicle velocity, Ω is the wheel angular velocity, r_D is the tire dynamic radius and v_N is a fictitious velocity, introduced to avoid numerical problems.

The chosen control strategy for developing an ABS model was based on Fuzzy Logic. Taking into account that the model is highly non-linear, this approach was considered superior to a linear control strategy. In addition, the Fuzzy controller can handle non-linear systems without needing a linearization process. The proposed ABS model works with a simple set of rules, as shown in Tab. 2.

Table (2) Fuzzy control rules for the proposed ABS model.

Longitudinal slip	Braking torque
Low	High
Medium	Medium
High	Small

The main feature of an ABS model is the capacity to control the slip, avoiding the wheels from locking, in a region in which the longitudinal tire force is close to the maximum. Therefore, the vehicle's maneuverability is still possible even if the vehicle is in full braking. In order to prove the effectiveness of the ABS model proposed in this work, two scenarios were simulated. In the first one, a straight line braking maneuver is used. With the ABS off, all wheels are locking instantly as we can see in the left upper plot (dashed lines) of Fig. 5. In the other hand, with the ABS on, the wheels are still rolling and then, it results in a smaller slip ($0.1-0.2$), left upper plot (solid lines) of Fig. 5. In addition, with the ABS on, the vehicle is reaching decelerations up to $\frac{\dot{v}}{g} = 1.0$, left bottom plot (solid line) of Fig. 5, which is the limit imposed by the friction coefficient ($\mu = 1.0$). The last scenario is built in order to test the gain in stability using the ABS model. In this simulation, the vehicle is driving in a straight line and at $t = 3.6s$ the steering wheel is applied and then at $t = 4.0s$, the brakes are triggered. With the ABS off, the wheels are locked by the braking torque. Therefore, the vehicle follows a straight trajectory because the front wheels are not capable to generate neither lateral or longitudinal forces. In the case of ABS on, the vehicle follows the driver's intentions because the front wheels are not locked completely and then, it can generate lateral forces. These results are shown in the right multi-frame shots of the trajectories in Fig. 5.

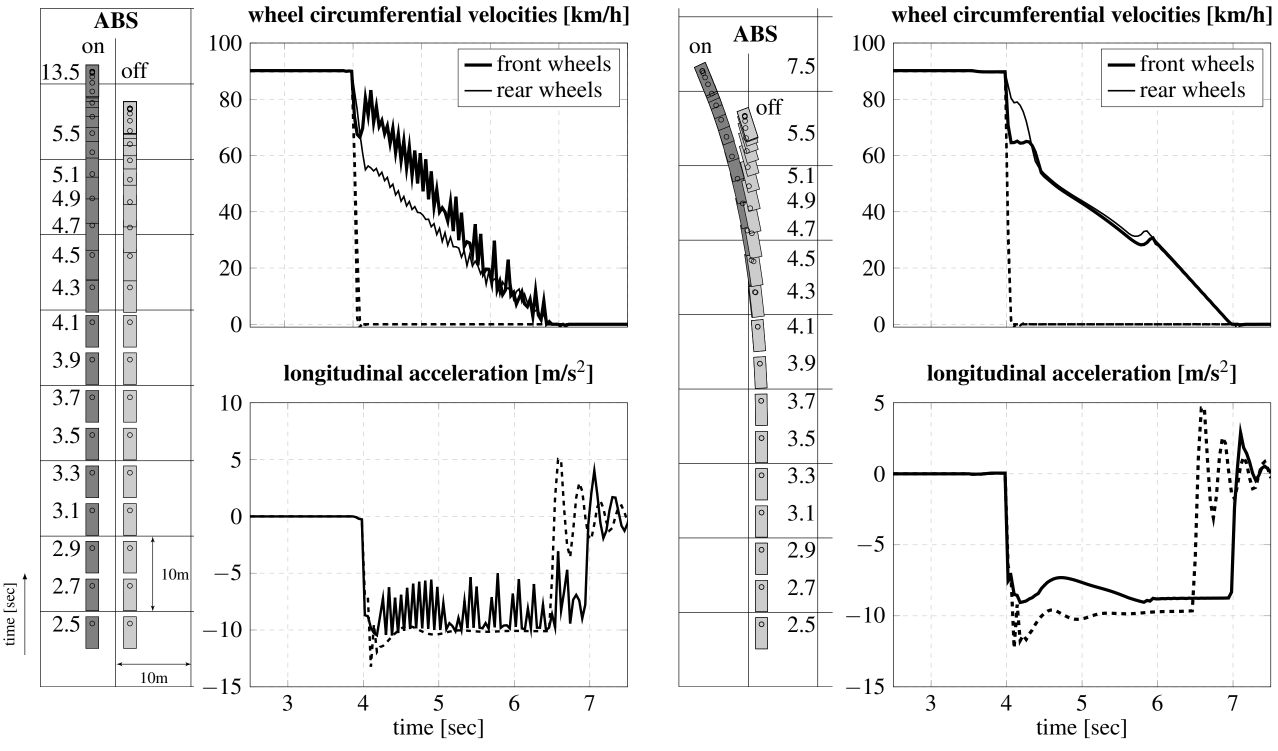


Figure (5) Left: braking in a straight line. Right: braking in a turn (— ABS on, - - - ABS off)

Electronic Stability Programme - ESP

The main objective of this system is to assist the driver in a critical driving situations, e.g. to avoid an unexpected obstacle on the road, and also compensate the disturbance produced by the driver in this critical scenario and thus prevent loss of vehicle stability. Before ESP can respond to a critical driving situation, it is necessary to analyze the current states of the vehicle. This analysis taking into account two variables: direction of the driving steering and direction in which the vehicle is moving. In order to determine these two variables, ESP uses the simple handling vehicle model as a reference. The difference between the actual yaw rate ω and the based on linear behavior (Eq. 2) ω_d, gives to ESP the necessary information to respond in critical scenarios, see Fig 6. Finally, ESP triggers the ABS in order to apply the correct brake torque to the selected wheel and then, a compensatory yaw moment T is generated.

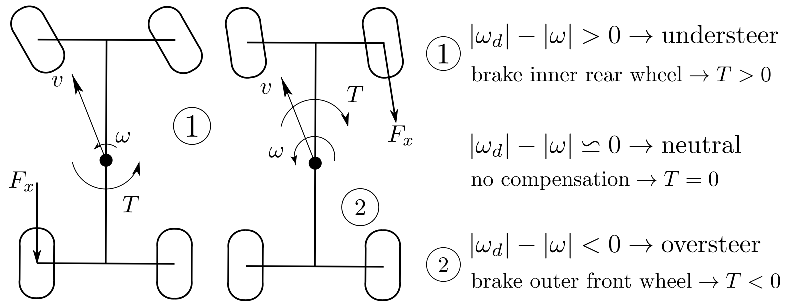


Figure (6) ESP control process.

In Fig. 7, two simulations are compared. The solid line indicates the results of an avoidance maneuver with ESP on, the dashed one considers the same maneuver with ESP off. It is possible to distinguished that, the torque frequency applied by ESP to the outer front wheel is high when the vehicle is on off-road $t = 5.75 \rightarrow 8$s. This is because, there is a μ-split condition when the vehicle goes off-road and come back to the road again, and this requires a torque compensation to maintain the vehicle stability. In this period, the yaw angular velocity (left upper plot), lateral acceleration (right upper plot) and total the overall yaw moment (left bottom plot) are maintained in a safe range for the ESP. Finally, ESP assisted the driver to maintain the desired path as indicated by the multi-frame shot at the very left of Fig. 7.

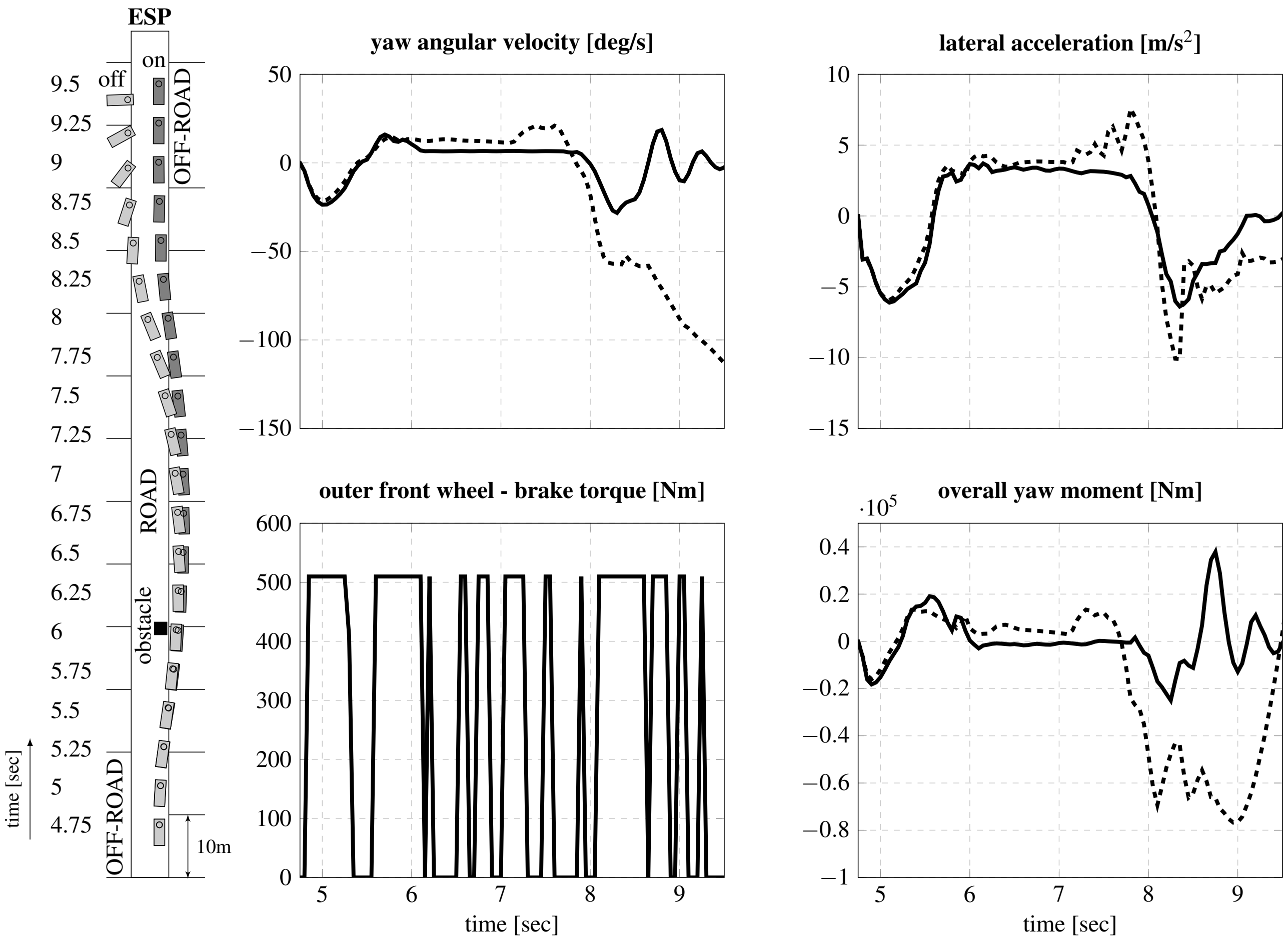


Figure (7) Trajectory and main states of the vehicle : — ESP on, - - - ESP off.

Four-Wheel Steering System - 4WS

The 4WS was introduced by Nissan in its model R31 Skyline in the late of 1985. Since this decade, many researches studied this technology in order to clarify its benefits against the normal front wheel steer vehicles. The main advantages of this system are: improve the maneuverability at low velocities and the lateral stability at high velocities. In (Furukawa *et al.*, 1989), it is concluded that a simple feed-forward control can improve the vehicle lateral stability. This controller monitors the front steering angle and depending of this value a rear wheel angle is imposed. Furthermore, a boundary is established to determine the rear wheels orientation, see Fig.8. In this study, the steering at hand wheel of 200° ($\delta_f \approx 11.5°$) was chosen as boundary.

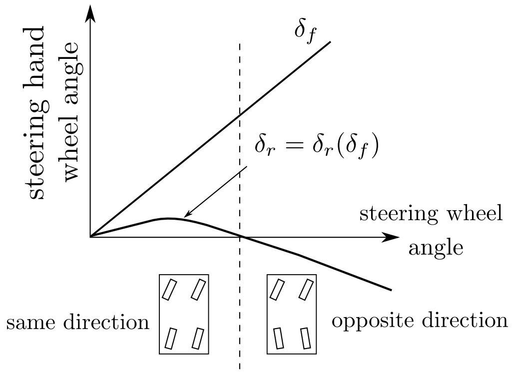


Figure (8) Proposed feed-forward control law.

In Fig. 9, a simulation of the vehicle with 4WS on and with 4WS off is shown. In this comparison, it can be distinguished some benefits of the 4WS system. From $t \approx 5$ s to 7 s, there is a slight improvement of the 4WS system on the vehicle stability. A large enhancement can be noticed between $t \approx 7$ s to 9.5 s. In this interval, the yaw angular velocity, lateral acceleration and the overall yaw moment are maintained in a safe range due to the 4WS. However, the 4WS system also produce undesirables oscillation on the rear wheels. This behavior is because, the driver model is trying to maintain the vehicle close to the center path of the road.

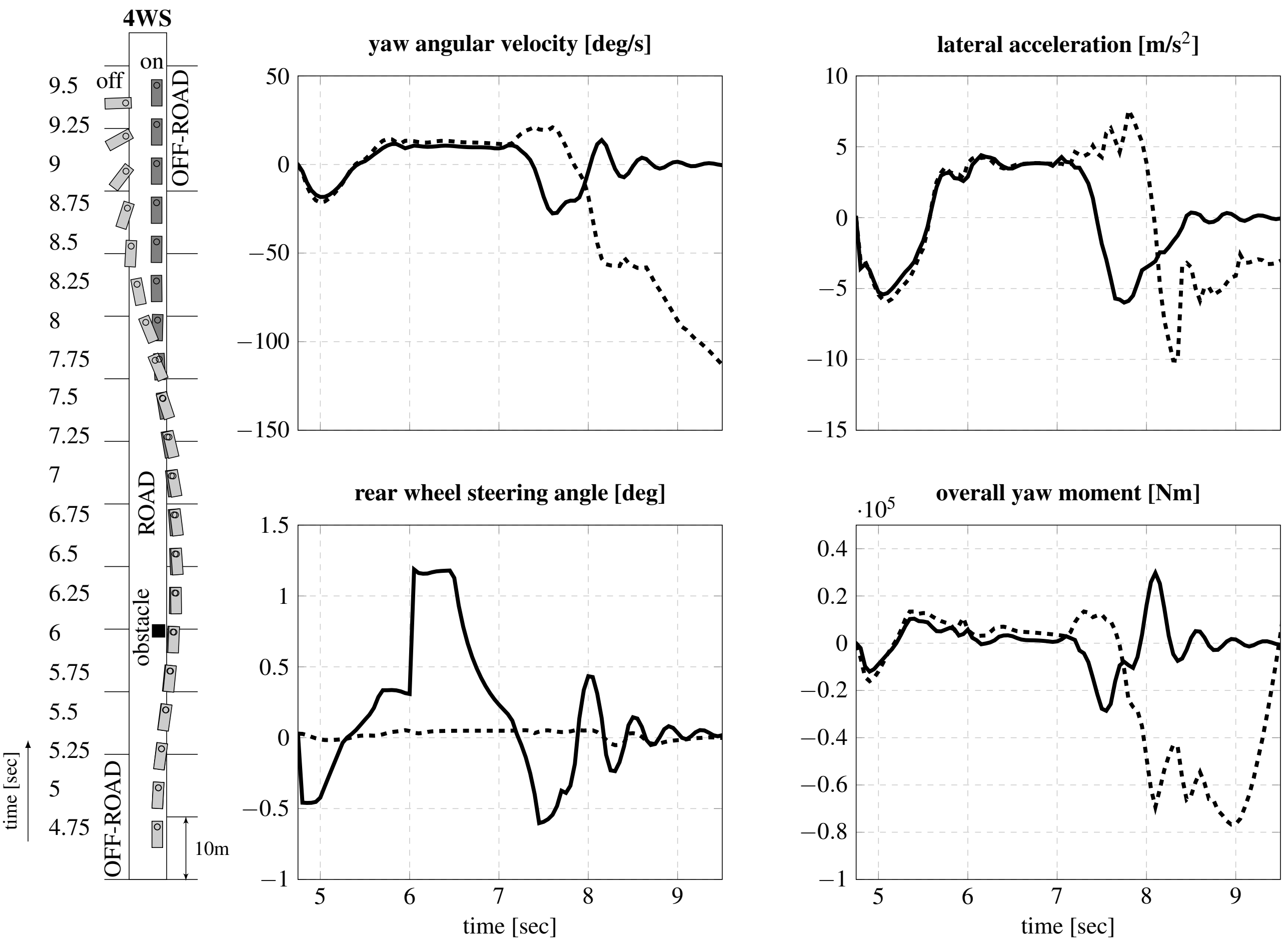


Figure (9) Trajectory and main states of the vehicle : — 4WS on, - - - 4WS off.

INTEGRATED CONTROL

A certain level of integration is required in order to enhance the vehicle stability. In this section, in order to prove the benefits of the integrated vehicle control, two simulation are performed. In the first one, a vehicle equipped with ESP and 4WS without any integration between them is used, this system will be refer as ESP+4WS. For the second simulation, a rule to avoid the conflict between ESP and 4WS was defined. This rule consist in limit the use of brake torque when the rear wheels are steered by certain angle. In other words, the ESP is applied only when the rear steer angles are bellow to a boundary value. The abbreviation name of this new system is IC and it is defined as follows:

$$IC = 4WS + ESP \begin{cases} \textbf{on}, & \text{if } |\delta_r| \leq \delta_U \\ \textbf{off}, & \text{otherwise} \end{cases} \tag{10}$$

where δ_r is mean of the rear steering angles and $\delta_U = 1°$ is the upper bound value for δ_r.

In order to a better analysis, the action of each subsystem (ESP and 4WS) of the IC and ESP+4WS is shown in the left and right plots of Fig. 10 respectively. For the ESP+4WS system (right plots), we can distinguish a conflict between the ESP and 4WS in the gray area of the plots. It is because, both systems contribute at the same time to the vehicle yaw reaction. In addition, this conflict can be notice in Fig. 7, where the yaw angular velocity and overall yaw moment are controlled by the ESP without having to steer the rear wheels. This kind of interactions between control systems inside the vehicle can reduce the vehicle's stability in a critical scenario. Therefore, in order to avoid this conflict, a rule Eq. 10 was introduced to limit the functionality of one subsystem. This new strategy is called IC as mentioned above. In the left upper and left bottom plots of Fig. 10, is noticed that the IC system avoid the conflict between 4WS and ESP in the periods of $t \approx 5.2 \rightarrow 7.2$ s and $t \approx 8.5 \rightarrow 9.5$ s. Furthermore, the lateral deviation using IC is less than the vehicle equipped with ESP+4WS, as indicated by the multi-frame shots at the very left of Fig. 10.

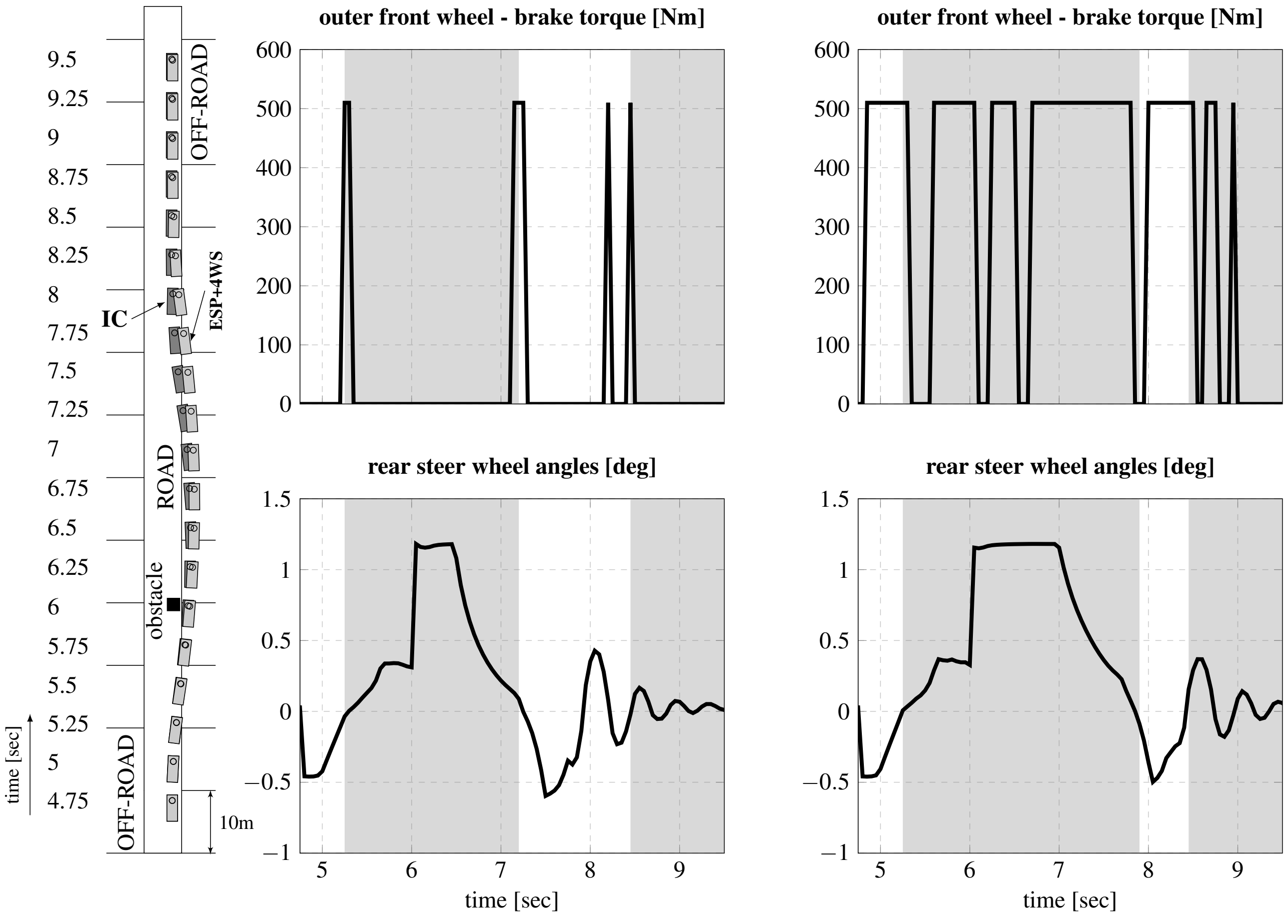


Figure (10) Trajectory and main states. Left upper (ESP action) and left bottom (4WS action) plots of IC system. Right upper (ESP action) and right bottom (4WS action) plots of ESP+4WS system.

In Fig. 11, a full comparison between a vehicle equipped with IC and only with ESP is shown. As we can see, in the multi-frame shots at the very left of Fig. 11, the lateral deviation of the vehicle with the integrated systems IC is smaller than the vehicle equipped with the ESP system. This difference happens because, the 4WS steer the rear wheels when the vehicle is in off-road condition, i.e., in the period $t \approx 5.2 \rightarrow 7.2$ s. The use of rear wheels improves the vehicle lateral stability because it decrease the yaw reaction produced by the front wheels. Therefore, in this period, is not necessary to use the ESP in order to produce a compensatory torque and thus, maintain the yaw stability. In the period of $t \approx 8.5 \rightarrow 9.5\,$s, a oscillation on the vehicle is produced by the driver model, because it try to maintain the vehicle close to the center path. However, the 4WS compensate this oscillations and stabilizes the vehicle rapidly. In this two periods, shadows areas in Fig. 11, the use of a brake torque trigger by the ESP is limited in cases in which the 4WS is not capable to maintain the vehicle lateral stability. This is easily distinguished because, the longitudinal forces in the outer front wheel of the vehicle equipped with the integrated control are nearly zero during this periods, see right bottom plot of Fig. 11.

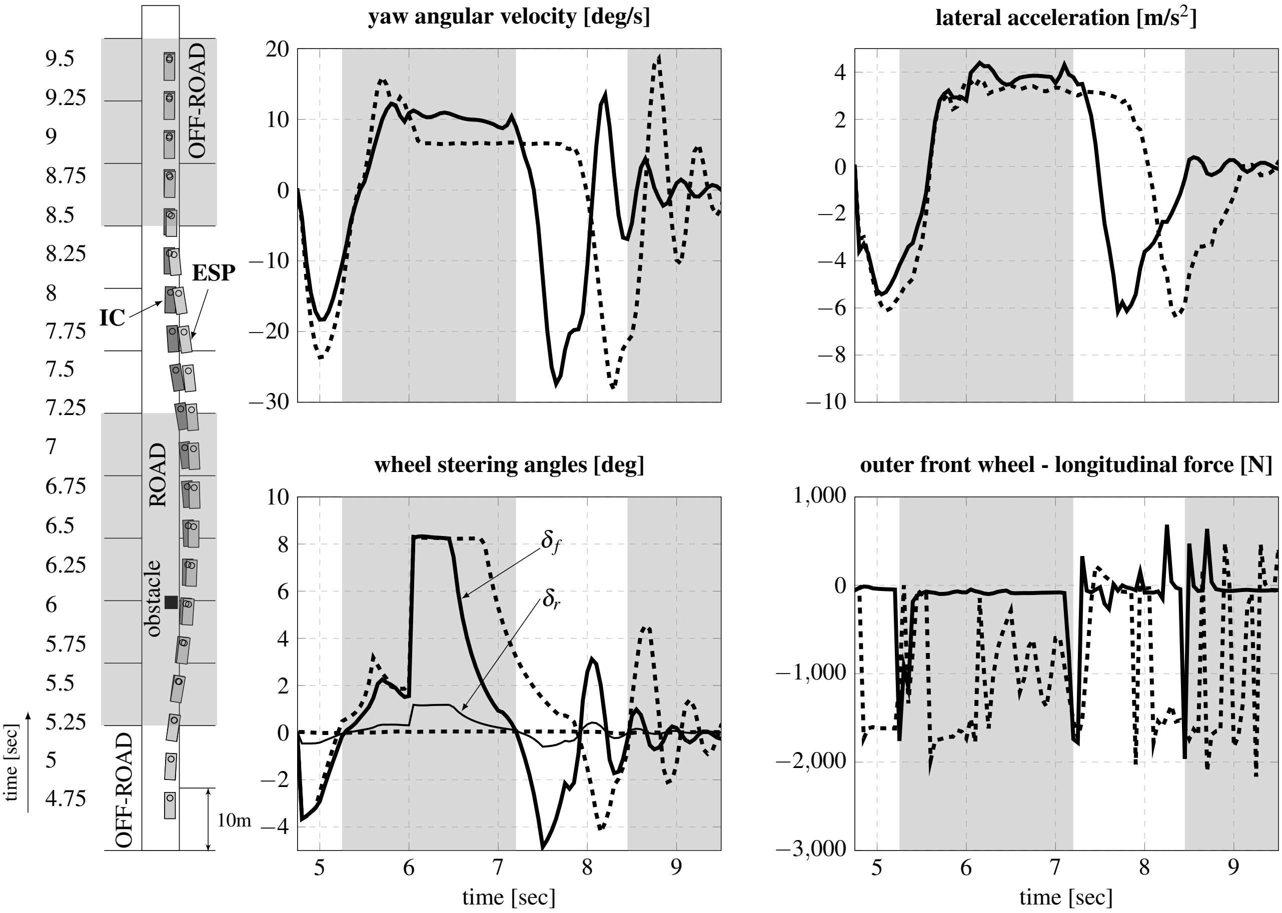


Figure (11) Trajectory and main states: — IC and - - - ESP.

CONCLUSIONS

The design of the individual controllers was done using different methods. The ABS model, based on fuzzy, was tested in two scenarios: braking in a straight line and in a curve. The simulations shown a good performance of this system, e.g. avoid wheels locking and hence maintain the steer-ability on braking in a turn scenario.

The ESP and 4WS system improved the yaw and lateral stability as visualized in previous sections. The direct use of the ESP and 4WS create a conflict between theses system in run-off-road scenario. A simple rule that limits the application of the ESP was used. From the simulation performed, we can concluded that the proposed integrated system IC can reduce the conflict between the ESP and 4WS. Using this enhanced strategy, an improvement of the vehicle lateral response and stability was achieved.

REFERENCES

Castro, A. and Rill, G. and Ingo Weber, H., 2015, “Designing an integrated control based on a hierarchical architecture to improve the performance of ground vehicles”, 23rd ABCM International Congress of Mechanical Engineering, 2015, Brazil.

BOVAG-RAI Foundation, Mobility in figures - 2013/2014 cars edition, available in: <http://www.bovagrai.info/auto/2013/8.9.html>, access in July 2, 2016.

Furukawa, Y. and Yuhara, N. and Sano, S. and Takeda, H. and Matsushita, Y., 1989, “A review of four-wheel steering studies from the viewpoint of vehicle dynamics and control”, Vehicle system dynamics, Vol.18, pp. 151-186.

Erke, A., 2008, “Effects of electronic stability control (ESC) on accidents: A review of empirical evidence”, Accident Analysis & Prevention, Vol.40, pp. 167-173.

Rill, G., 2011, “Road vehicle dynamics: fundamentals and modeling”, CRC Press, Boca Raton.

Rill, G., 2013, "TMeasy - A Handling Tire Model based on athree-dimensional slip approach", XXIII International Symposium on Dynamic of Vehicles on Roads and on Tracks - IAVSD, 2013, China.

Rill, G., 2006, "Vehicle modeling by subsystems", XI DINAME International Symposium on Dynamic Problems of Mechanics, 2006, Brazil.

RESPONSIBILITY NOTICE

The authors are the only responsible for the material included in this paper.