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DESIGNING AN INTEGRATED VEHICLE CONTROL BASED ON A HIERARCHICAL ARCHITECTURE TO IMPROVE THE PERFORMANCE OF GROUND VEHICLES

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Abstract. *Nowadays, there is a steady increase of electronic components in vehicles due to the high and underlying importance of improving both, the safety and the driving experience of the driver and passengers. In this paper, a hierarchical control architecture approach with two layers is used in the integrated control system of ground vehicles. The active front steering (AFS) and electronic stability program (ESP) are integrated in order to improve the dynamic performance of the vehicle in both, stability and handling. The upper layer monitors the driver's intentions and coordinate the interactions between the AFS and ESP subsystems. The lower layer is the stand-alone actuator control of the two subsystems studied. In order to achieve the proposed objectives, two vehicle dynamic models are used: a tree-dimensional and fully non-linear model for simulating the dynamic behavior of the vehicle and two degrees of freedom (DOF) model for designing controllers and calculating the desired response of the driver's input. Using this approach to integrate the vehicle controllers, the simulations result illustrates improvements on stability and handling vehicle performance over the non-integrated control approach.*

Keywords: *integrated control, hierarchical architecture, non-linear vehicle model.*

1. INTRODUCTION

Nowadays, most of the commercial vehicles are increasingly using active controls systems instead of traditional mechanical system in order to improve the vehicle handling, stability, and comfort. These active control systems can be classified into three categories, according to the direction of action in the vehicle behavior, i.e., vertical, lateral, and longitudinal directions. In the first one, we have, e.g., the active suspension system (ASS) and the active body control (ABC). The second one, e.g., electric power steering system (EPS), active front steering (AFS), and 4-wheel steering control (4WS). Finally, in the longitudinal direction, e.g., anti-lock brake system (ABS), electronic stability program (ESP), and traction control (TRC). In general, these active controls systems are designed and built by different suppliers with their own technologies and components to accomplish local functionalities. When these controls are included into commercial vehicles, they operate independently and the overall control system results in a parallel vehicle control architecture. In this architecture, two major problems arise. First, the increasing number of hardware and software, because each active control need its own sensors. Second, conflicts between local objectives of each stand-alone control. In order to solve these problems, an approach called integrated vehicle dynamics control was proposed in the 90's by Fruechte et al. Fruechte *et al.* (1989). This special control, is an advanced control system that integrates optimally all the stand-alone controls inside the vehicle in order to improve the overall vehicle performance, including safety, comfort and economy.

The aim of the integrated vehicle dynamics control is to improve the overall vehicle performance. This objective is achieved through the optimal combination between hardware, software and control strategies. According to Gordon et al. Gordon *et al.* (2003), control techniques for integrated vehicle dynamics control are classified in two categories, i.e., multi-variable and hierarchical control.

The second technique proves to be more effective than the first, because its facilitates the modular design of the integrated control system, use layers for masking the complex details of stand-alone controls, favoring scalability, and share information of sensors and therefore reduce the cost of implementation. Accordingly to these features, many researchers are interested in this technique, proof of this interest are some work developed in recent years, e.g., Gordon et al. Gordon *et al.* (2003), Rodic and Vukobratovic Rodic and Vukobratovic (2000), and Chang and Gordon Chang and Gordon (2007).

In this paper, a comprehensive study of integrated vehicle dynamics control is performed. The study consists in, use a two-layer hierarchical control architecture to integrate the active front steering control (AFS) and the electronic stability program (ESP). In this architecture, the upper layer coordinate the interactions between the AFS and ESP, and the lower layer is conformed by the stand-alone controls of AFS and ESP. A simulation is performed to demonstrate the effectiveness of the hierarchical control over the parallel control system.

2. VEHICLE MODELS

In this study, two vehicle models are established: a three-dimensional and fully non-linear model simulating the dynamic behavior of the vehicle, and a single-track model for designing the controllers and calculating the desired response of the driver's input.

2.1 Full non-linear vehicle model

The multi-body approach is used to model the dynamics of a ground vehicle. For the full vehicle model there are considered nine rigid bodies, i.e. four knuckles and wheels and one chassis, see Fig. 1 (a). This rigid bodies are attached to each other with mechanical elements, e.g. the suspension system, steering knuckle, etc. Therefore, the full vehicle model is coupled in nature and this phenomenon has to be considered. A full vehicle model consists of a framework supplemented by separate modules, e.g. the steering system, drive train, tires, among others. The framework represents the kernel of this model, and it includes, at least, the chassis and the suspension systems.

To obtain the kinematics of the full vehicle model, it necessary to define properly some reference frames. The kinematics of each body is demonstrated in the moving reference frame $\Re_F$. At first, we describe the different motions of references frames with each others. As shown in Fig. 1 (b), the moving frame $\Re_F$ is obtained after three successive rotations relative to the inertial reference $\Re_O$. First, a yaw rotation γ around $\Re_O - z$ axis that defines an intermediate reference frame that we call $\Re_\gamma$. Second, a pitch rotation β around $\Re_\gamma - y$ axis that defines again an intermediate reference frame $\Re_\beta$. Finally, the roll rotation α around $\Re_\beta - x$ axis defines the $\Re_F$ reference frame. Generally in ground vehicles and in steady state regimen, the pitch motion is depreciated.

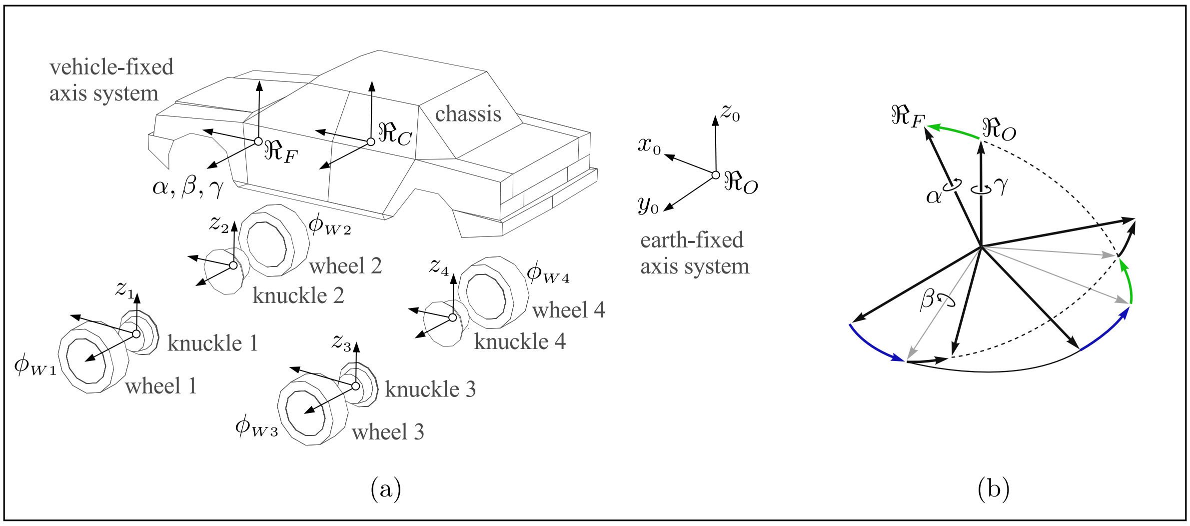

Figure 1. (a) Multi-body model of a ground vehicle. (b) Sequence of rotations of reference frames.

One of the most important phenomena in vehicle dynamics is the interaction between the tire and road. In the actual literature there exist many models but, these can be classified into two categories, e.g., empirical and physical models. The choice of the model to study this phenomena depends on the type of analysis to be performed. In this work, a balance between accuracy and time consuming is very important. This characteristics are important for handling and real time applications. Therefore, the TMeasy tire model Rill (2011) is used in order to obtained reliable and fast simulations.

In general driving maneuvers, i.e., acceleration and deceleration in a curve path, the longitudinal slip s_x and lateral slip s_y occur at the same time. Therefore, the combination of the slips and thus of the longitudinal and lateral forces requires a normalization process:

$$s = \sqrt{\left(\frac{s_x}{\hat{s}_x}\right)^2 + \left(\frac{s_y}{\hat{s}_y}\right)^2} = \sqrt{\left(s_x^N\right)^2 + \left(s_y^N\right)^2} \tag{1}$$

where s_x^N and s_y^N are the normalized slips, and $\hat{s}_x$ and $\hat{s}_y$ are the normalized factors defined by the characteristic of longitudinal and lateral forces. These factors balance the contribution of the individual slips s_x and s_y in the combined slip s and are defined as:

$$\hat{s}_i = \frac{s_i^M}{s_x^M + s_y^M} + \frac{F_i^M/dF_i^0}{F_x^M/dF_x^0 + F_y^M/dF_y^0} \tag{2}$$

where $i = x, y$ and F_i^M, dF_i^0, and s_i^M are the characteristic of the lateral and longitudinal curve of force against to their respective slips. Similar to the curve of the 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:

$$dF^0 = \sqrt{(dF_x^0 \hat{s}_x \cos\phi)^2 + (dF_y^0 \hat{s}_y \sin\phi)^2} \tag{3}$$

$$s^M = \sqrt{\left(\frac{s_x^M}{\hat{s}_x}\cos\phi\right)^2 + \left(\frac{s_y^M}{\hat{s}_y}\sin\phi\right)^2} \tag{4}$$

$$F^M = \sqrt{\left(F_x^M \cos\phi\right)^2 + \left(F_y^M \sin\phi\right)^2} \tag{5}$$

$$s^S = \sqrt{\left(\frac{s_x^S}{\hat{s}_x}\cos\phi\right)^2 + \left(\frac{s_y^S}{\hat{s}_y}\sin\phi\right)^2} \tag{6}$$

$$F^S = \sqrt{\left(F_x^S \cos\phi\right)^2 + \left(F_y^S \sin\phi\right)^2} \tag{7}$$

The angular function ϕ is used to grant a smooth transition from the longitudinal and lateral force to the combined force, and can be defined as follows:

$$\cos(\phi) = \frac{s_x^N}{s} \quad \text{and} \quad \sin(\phi) = \frac{s_y^N}{s} \tag{8}$$

Finally, the longitudinal and lateral result force are calculated from the combined force as follow:

$$F_x = F\cos(\phi) \quad \text{and} \quad F_y = F\sin(\phi) \tag{9}$$

For standard values of commercial vehicles, we obtain the following results for tire combined forces using the TMeasy tire model, Fig. 2.

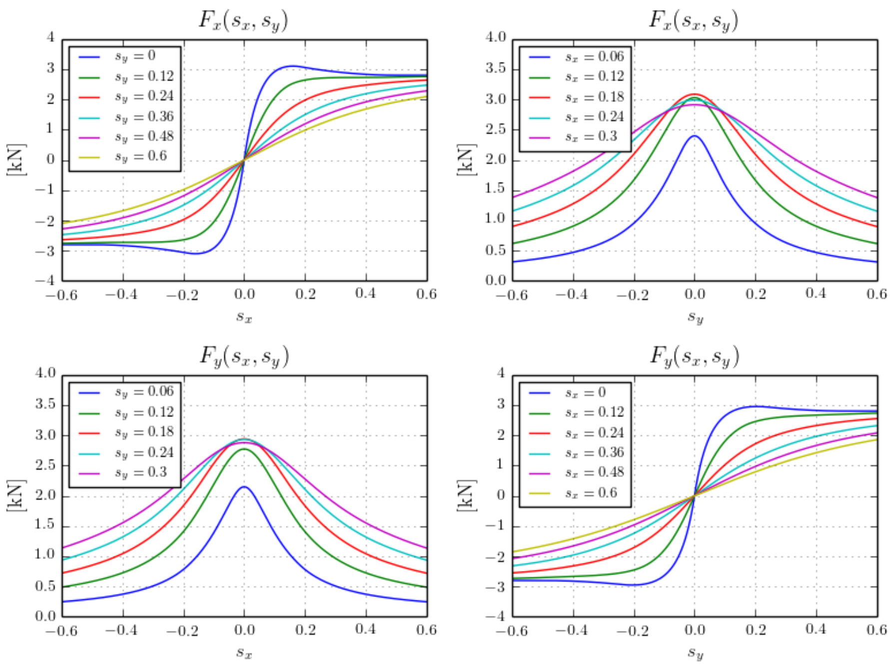

Figure 2. Combined tire force using TMeasy tire model.

2.2 Single-track model

The single-track model is used to obtain the desired response of the vehicle. In vehicle dynamics, this classical model as shown in Fig. 3 is used for yaw stability control analysis and controlled design, the single-track model has two principal states: the yaw rate $\dot{\psi}$ and the sideslip angle β. This model is based on the following assumptions:

- The tire dynamics operates in the linear region.
- The vehicle moves on a plane surface.
- Left and right wheels at the front and rear axle are lumped.
- Constant longitudinal vehicle speed, i.e, $a_x = 0$.
- Steering angle δ and vehicle sideslip β angle are assumed small.
- No braking is applied.
- The two front wheels have the same steering angle.
- Vehicle sideslip angle β is assumed to be zero in steady state.

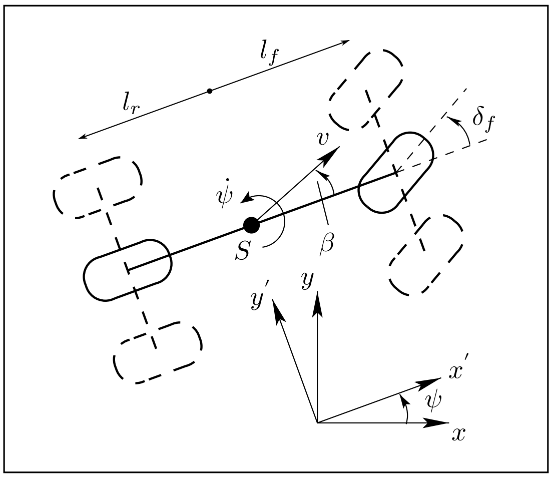

Figure 3. Single-track model of the vehicle with your principal states.

Follow the above assumptions, the single-track model have only 2 DOF, i.e., lateral and yaw motions and are described in the following equations, respectively:

$$mv(\dot{\beta} + \dot{\psi}) = (F_{y,f} + F_{y,r}) \tag{10}$$

$$I_z \ddot{\psi} = l_f F_{y,f} - l_r F_{y,r} \tag{11}$$

Then, the front and the rear tire always working in the linear region and are described as a linear function of the front and rear cornering stiffness, C_f and C_r respectively as follow:

$$F_{y,i} = C_i \alpha_i \tag{12}$$

where $i = f, r$ is used to identify the front and rear tires and the front α_f and read sideslip angle α_r are given in the following equations:

$$\begin{aligned} \alpha_f &= \delta_f - \beta - \frac{l_f \dot{\psi}}{v} \\ \alpha_r &= -\beta + \frac{l_r \dot{\psi}}{v} \end{aligned} \tag{13}$$

Finally, rearranging and simplifying the equations from (10) to (13) the steady state model of the single-track model is:

$$\begin{bmatrix} \ddot{\psi} \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{l_f^2 C_f + l_r^2 C_r}{I_z v_x} & \frac{l_r C_r - l_f C_f}{I_z} \\ 1 + \frac{l_r C_r - l_f C_f}{m v_x^2} & -\frac{C_f + C_r}{m v_x} \end{bmatrix} \begin{bmatrix} \dot{\psi} \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f l_f}{I_z} \\ \frac{C_f}{mv} \end{bmatrix} \delta_f \tag{14}$$

where the yaw rate $\dot{\psi}$ and the vehicle sideslip angle β are the output states, C_f and C_r are the cornering stiffness of the front and rear tires, m is the total mass of the vehicle, I_z is the moment of inertia around the z axis, l_f and l_r are the distance from front and rear axle to the center of gravity respectively, v is the vehicle speed, and the steering angle δ_f is the input of the steady state model.

3. INTEGRATED CONTROL

Generally, the integrated vehicle control involves the following tasks. First, the desired vehicle motion is inferred from the driver action. e.g., steering angle, acceleration-pedal, and brake. Second, the actual vehicle motion, measured by the sensors, is compared with the desired motion. Finally, the control actions are applied to the actuator of the vehicle, e.g., wheel-brake pressure, steering wheel, to regulate the vehicle motion in order to approximate it to the desired motion under certain stability constraint, e.g., vehicle sideslip angle β and roll motion. In certain driving situations, the stability of the vehicle is reduced, normally when the tire are working in the non-linear region. Fig. 4 shows that the authority of front-axle steering is reduced greatly when the vehicle sideslip angle is large. In other words, regardless if the human driver is steering the front wheels to the left or to the right, similar amounts of lateral force and yaw moment are generated due to tire saturation.

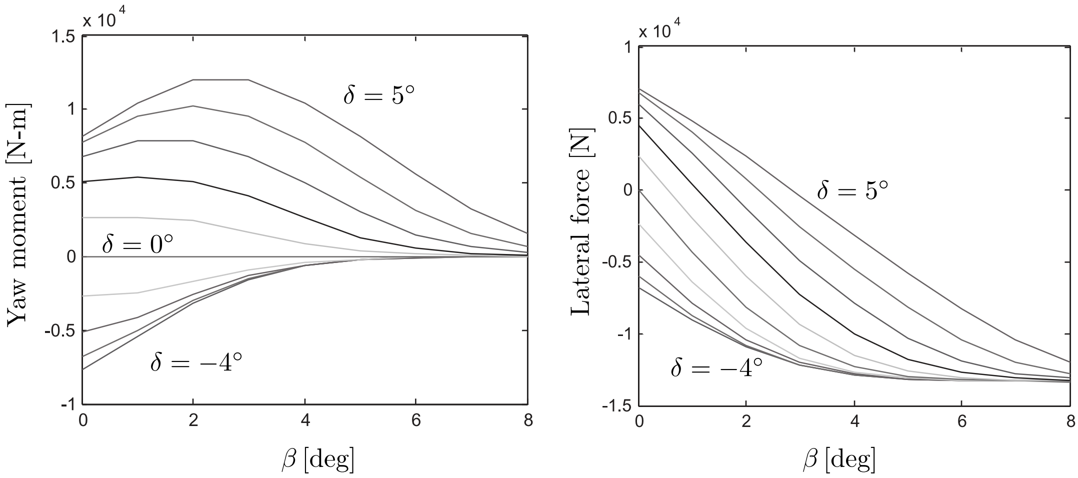

Figure 4. Control authority reduction.

3.1 Yaw stability control

The yaw rate $\dot{\psi}$ and the vehicle sideslip angle β are important states in vehicle yaw stability control. In the actual literature, the control objectives of yaw stability control may be classified into three categories, i.e., yaw rate control, vehicle sideslip angle control and the combinations of both.

In the yaw rate control, the objective is maintain the actual yaw rate $\dot{\psi}$ close to the desired yaw rate $\dot{\psi}_d$ (obtained from the single-track model). This control will improve the handling and maneuverability of the vehicle. In the steady state condition, the desired yaw rate response $\dot{\psi}_d$ is obtained using the following equation (from the single-track model):

$$\dot{\psi}_d = \frac{v}{(l_f + l_r) + k_{us} v^2} \delta_f \tag{15}$$

where the stability factor k_{us} is defined as:

$$k_{us} = \frac{m(l_r C_r - l_f C_f)}{(l_f + l_r) C_f C_r} \tag{16}$$

The second category considers, vehicle sideslip angle control β. The control of sideslip angle close to the steady state condition means controlling the lateral stability of the vehicle. In the steady state condition, the desired sideslip angle β_d is always zero. Therefore, to improve the vehicle handling and stability performances, it is necessary to control both states, i.e., the yaw rate $\dot{\psi}$ and vehicle sideslip angle β.

3.2 Active chassis control

The steering and braking subsystems are part of the vehicle chassis. The active control of yaw stability system can be realized through active chassis control, i.e., for the actuations of the steering, or brake subsystems or both. By controlling the brake pressure of individual wheels we have the possibility to control directly the yaw moment, this is called direct yaw moment control. As illustrated in Fig. 5, the required corrective yaw moment ΔM_z which is generated by the pressure distributions in individual wheels and this distribution is calculated by the designed controller based on the error of the yaw rate and the sideslip angle.

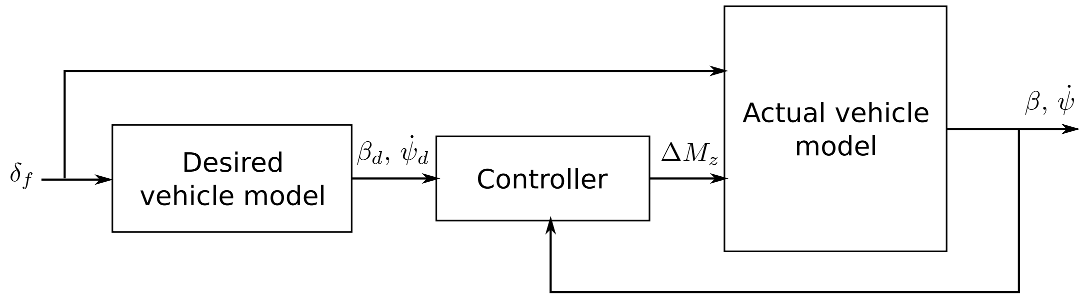

Figure 5. Direct yaw diagram control.

The design of direct yaw moment control is based on state space model of the single-track vehicle model Eq. (14), but in this case we have to consider the corrective yaw moment ΔM_z as an input, as shown in the following equation:

$$\begin{bmatrix} \ddot\psi \\ \dot\beta \end{bmatrix} = \begin{bmatrix} -\frac{l_f^2 C_f + l_r^2 C_r}{I_z v_x} & \frac{l_r C_r - l_f C_f}{I_z} \\ 1 + \frac{l_r C_r - l_f C_f}{m v_x^2} & -\frac{C_f + C_r}{m v_x} \end{bmatrix} \begin{bmatrix} \dot\psi \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f l_f}{I_z} \\ \frac{C_f}{mv} \end{bmatrix} \delta_f + \begin{bmatrix} \frac{1}{I_z} \\ 0 \end{bmatrix} M_z \tag{17}$$

Active steering control is another approach to improve the vehicle yaw stability. In literature, the active steering control is classified into three categories, i.e., active front steering (AFS), active rear steering (ARS) and four-wheel active steering (4WAS). In commercial vehicles the most common is the AFS. In the AFS control diagram, as shown in Fig. 6, the total front steer angle is the sum of the driver steer angle plus the corrective steer angle generated by the controller. Similar to the direct yaw moment, this corrective steer angle is computed based on the yaw rate and sideslip angle error.

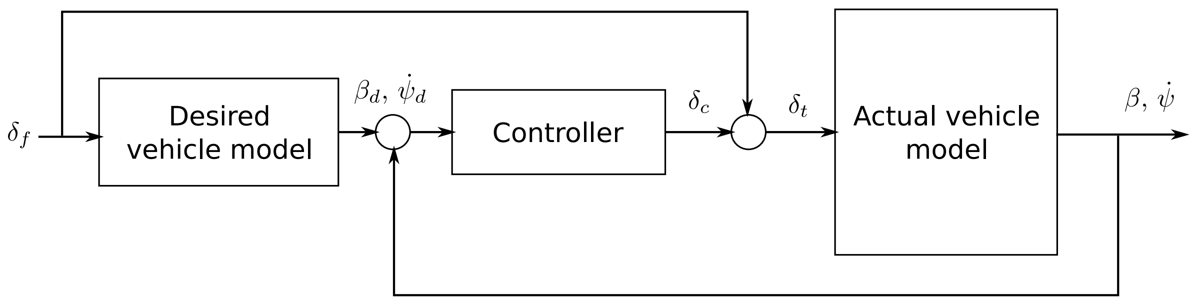

Figure 6. Active steer diagram control.

To analyse and design the active front steering control the linear state space model used is described as follows:

$$\begin{bmatrix} \ddot\psi \\ \dot\beta \end{bmatrix} = \begin{bmatrix} -\frac{l_f^2 C_f + l_r^2 C_r}{I_z v_x} & \frac{l_r C_r - l_f C_f}{I_z} \\ 1 + \frac{l_r C_r - l_f C_f}{m v_x^2} & -\frac{C_f + C_r}{m v_x} \end{bmatrix} \begin{bmatrix} \dot\psi \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f l_f}{I_z} \\ \frac{C_f}{mv} \end{bmatrix} (\delta_f + \delta_c) \tag{18}$$

However, the integrated vehicle control chassis, i.e., direct yaw moment and active front steering, has become very popular, because commercial vehicles normally have AFS and a brake subsystem and this characteristic facilities the implementation of the integrated vehicle control. In this approach, the corrective yaw moment and the corrective steer angle are used simultaneously to improve the handling and stability performance of the vehicle as illustrated in Fig. 7.

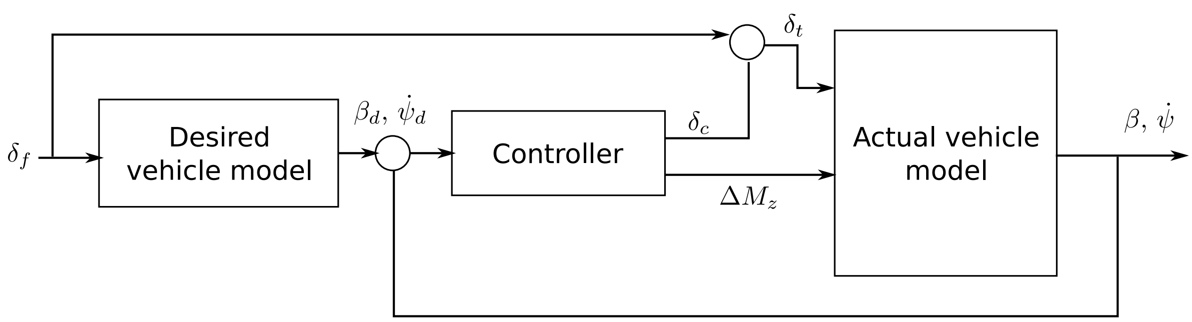

Figure 7. Integrated control: active front steering and direct yaw moment.

For the integrated control, to analyse and design it, the linear state space model used is described as follows:

$$\begin{bmatrix} \ddot{\psi} \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{l_f^2 C_f + l_r^2 C_r}{I_z v_x} & \frac{l_r C_r - l_f C_f}{I_z} \\ 1 + \frac{l_r C_r - l_f C_f}{m v_x^2} & -\frac{C_f + C_r}{m v_x} \end{bmatrix} \begin{bmatrix} \dot{\psi} \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{C_f l_f}{I_z} \\ \frac{C_f}{mv} \end{bmatrix} \delta_f + \begin{bmatrix} \frac{1}{I_z} & \frac{C_f l_f}{I_z} \\ 0 & \frac{C_f}{mv} \end{bmatrix} \begin{bmatrix} \Delta M_z \\ \delta_c \end{bmatrix} \tag{19}$$

In conclusion, the integrated control is implemented to improve the handling and vehicle stability. In this work, a simple PD control strategy is used for both controllers, i.e., the direct yaw moment and the active steering control. The following equations define the control law for the integrated control strategy:

$$\begin{array}{l} \text{If} \quad |\dot{\psi}\text{-}\dot{\psi}_d| \geq \dot{\psi}_{thersold} \quad \text{and} \quad \frac{|\dot{\psi} - \dot{\psi}_d|}{\dot{\psi}_d} \geq \dot{\psi}_{thersol_{perct}} \\ \text{Then} \quad \zeta_{M\psi} = K_{M\psi}(\dot{\psi} - \dot{\psi}_d) \text{ and } \zeta_{\delta\psi} = K_{\delta\psi}(\dot{\psi} - \dot{\psi}_d) \\ \qquad \text{Else} \quad \zeta_{M\psi} = 0 \text{ and } \zeta_{\delta\psi} = 0 \end{array} \tag{20}$$

$$\begin{array}{l} \qquad \text{If} \quad |\beta| \geq \beta_{thersold} \quad \text{and} \quad \beta\Delta\beta > 0 \\ \text{Then} \quad \zeta_{M\beta} = K_\beta \beta + K_{\beta_d}\Delta\beta \text{ and } \zeta_{\delta\beta} = K_{\delta\beta}\beta + K_{\delta\beta_d}\Delta\beta \\ \qquad \text{Else} \quad \zeta_{M\beta} = 0 \text{ and } \zeta_{\delta\beta} = 0 \end{array} \tag{21}$$

4. RESULTS AND CONCLUSIONS

The Figures 8, 9 and 10 show the state variables of the non-linear vehicle model with an integrated vehicle dynamics control. In Fig. 8 (a) we can see the input of the vehicle model, in this case we use a sinusoidal steering wheel angle with maximum amplitude of $5°$.

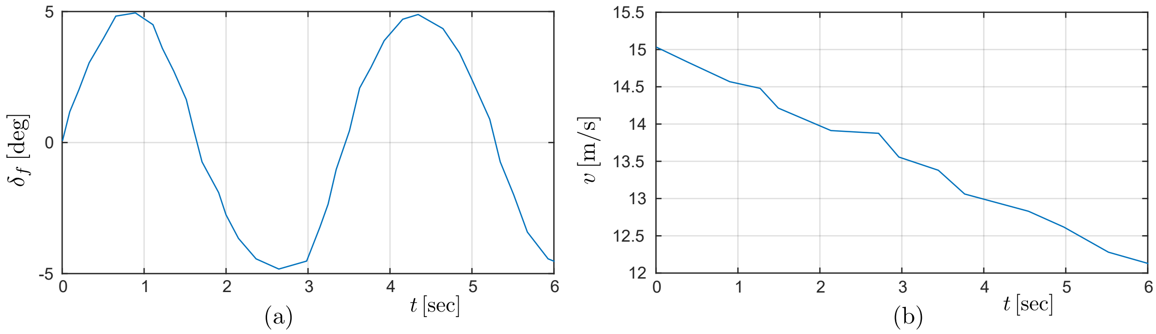

Figure 8. Steering angle: δ and Vehicle speed: v.

Fig. 8 (b) shows that the vehicle velocity is decreasing, this is because the torque distribution applied on individual wheels to maintain the vehicle stability in this sinusoidal maneuver.

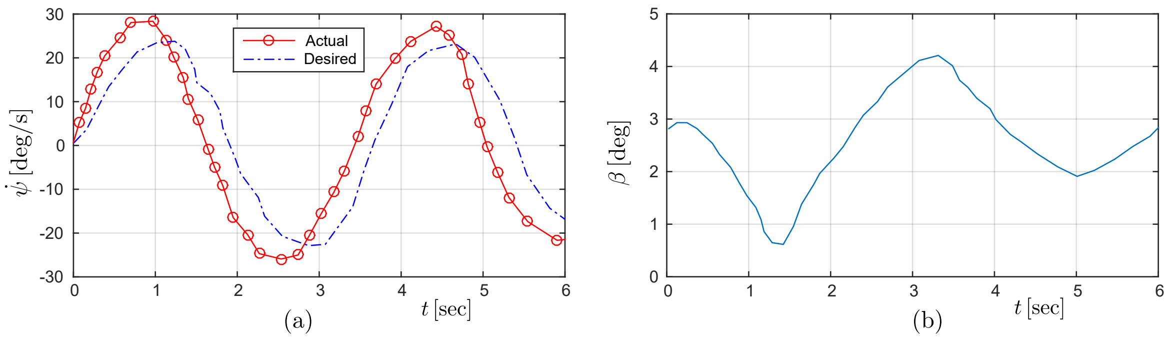

Figure 9. Yaw rate: $\dot{\psi}$ and Vehicle sideslip angle: β.

The actual yaw rate $\dot{\psi}$ tracking the desired yaw rate $\dot{\psi}_d$, for action of the integrated control, as illustrated in the Fig. 9 (a). Figure 10 show the integrated control signal.

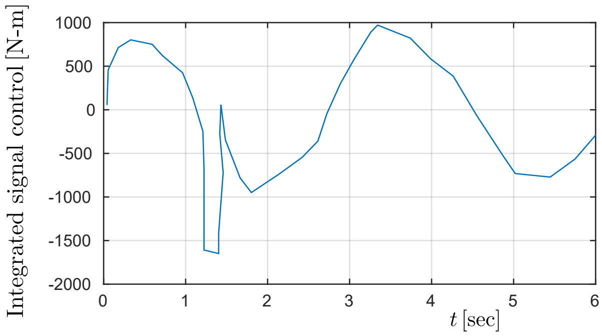

Figure 10. Integrated signal control and Roll angle.

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