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FUZZY LOGIC CONTROL FOR THE LATERAL AND LONGITUDINAL MOTION OF PASSANGER CARS AIMING AT VEHICULAR SAFETY

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Abstract. *The active safety systems must prevent accidents before they occur. They include braking and traction control systems, four wheel steering and the yaw dynamic control systems, that directly generate the right amount of momentum to improve the vehicular handling and stability performance. The momentum can be generated by an increase in the traction force, or by the controlled application of braking torques. This paper presents a yaw momentum control development and simulation, using fuzzy logic, in order to improve the vehicular handling, dirigibility and safety. The fuzzy estimator is composed of 36 rules and to correct the yaw rate, it determines and designates a chosen slip for each of the vehicle's wheels, according to its effectiveness in generate the required momentum. These slips are treated as an input slip. The vehicle model is non-linear and considers seven degrees of freedom, the yaw rotation, the angle between the vehicle X axis and its velocity vector, the center of gravity velocity and the four degrees of freedom related to the wheels rotation. Because the proposed control system controls each wheel individually, it is also effective in maneuvers with braking or acceleration, without the need of some additional control system.*

***Keywords**: Vehicular safety, fuzzy logic, yaw momentum, vehicular simulation, vehicular control*

1. Introduction

The vehicular dynamics research is concerned with the vehicular movement analysis, the way the vehicle is guided and the road irregularities. According to Wong (2001), the vehicle characteristics can be described in terms of its performance, handling and ride. The vehicular behavior is the result of the interaction between the driver, the vehicle and the environment. The active safety systems, as braking control (such as ABS) or yaw momentum control (such as ESP or VDC), help improve the handling and the performance characteristics of the vehicle, thus making its use safer for the driver, the passengers and the pedestrians.

This paper scope is the development and simulation of a yaw dynamics control. The yaw momentum control is a system that relies on the vehicle braking system in order to stabilize the vehicle. The aim of these systems is to reduce the yaw momentum generated by the different slips and friction between the wheels, when cornering

While the anti-lock braking systems can avoid and prevent wheel lock-up during braking and the traction control systems avoid excessive wheel spin during acceleration on slippery roads, these systems can not effectively control the directional vehicle behavior. When the yaw control systems start operation, they change the braking system priorities, in order to maintain the vehicle stable, steerable and on the intended course, independent from the road or environmental conditions. To accomplish this task, the desired correctional momentum can be generated by an active increase in the traction force transmitted by the driveline, or by the controlled application of braking torques on the wheels. The wheel to be controlled is chosen according to its capacity in generate the desired momentum (Kin et. al, 2002).

The yaw control systems characteristics can change according to the manufacturers, but the basic operation principles are very similar. Some commercial yaw momentum stability control systems utilize the assisted braking control system components and the traction control system components. Because it is impossible to change directly the lateral forces on the tire-road contact, the lateral movement occurs through the inducing of a yaw momentum, which originates the vehicle rotation around the Z axis. This rotation is so that the vehicle body side slip angle and the tire side slip angles change towards the optimum or desired values. To produce this angular momentum, the controller can also intervene on the selected tire slip value, to indirectly influence the longitudinal and lateral forces on each individual tire-road contact.

The developed yaw momentum control presented in this paper uses fuzzy logic to determine and designate a chosen slip for each of the vehicle's wheels, according to its effectiveness in generate the required momentum. Each of these slips is treated as a reference input slip, and the most suitable braking torques are then applied in order to obtain the required values. The control system fuzzy estimator is composed of 36 rules and the controller output is the individual torque variation on the wheels.

2. Vehicular model

The vehicle model for this work has seven degrees of freedom, one for the yaw rotation ($\dot{\psi}$), the angle between the vehicle X axis and his velocity vector (β) and the Center of Gravity (CoG) translation, besides the four degrees of freedom related with the wheels rotation ω_W. A single-track representation of the model, with the forces and variables involved, is shown on Fig. 1.

In order to simplify the model, some assumptions are made. The driveline dynamics and losses, the lateral aerodynamic forces and the drag due to the air friction in the Z direction are disregarded. There are no suspension or actuators dynamics in the model, and the road is considered flat, even and level, without the variations of roll and pitch, and no movement in the Z axe. No losses due to accessories are modeled as well.

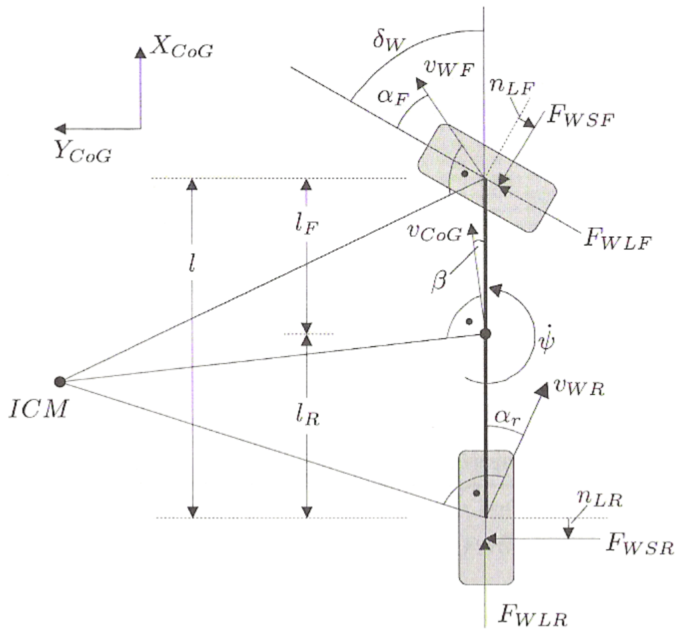

Figure 1 – Single-track representation of the vehicle model (Kiencke & Nielsen, 2000).

2.1. Model equations

The reduced model should contain only the state variables that are essential to the vehicle dynamic control and braking control. These variables are the vehicle velocity, v_{CoG}, the vehicle body sideslip angle, β, and the yaw rate ($\dot{\psi}$). The lateral wheel forces are approximated to be proportional to the tire sideslip angle (α). The model presented here is based on the model proposed for Kiencke & Nielsen (2000).

The road-tire contact forces are modeled according to the Buckhardt approximation. This is an empirical method to determine, based on the wheel slips Tab. 1, the value for the friction coefficients. With this coefficients and the normal force F_Z, it is possible to have the longitudinal and lateral forces values.

The model equations, in the state space form, are:

$$\{\dot{x}\} = \begin{bmatrix} \dot{v}_{CoG} \\ \dot{\beta} \\ \ddot{\psi} \end{bmatrix} = \{f(\{x\},\{u\})\} = \begin{bmatrix} f_1(\{x\},\{u\}) \\ f_2(\{x\},\{u\}) \\ f_3(\{x\},\{u\}) \end{bmatrix} \quad (1)$$

$$\{y\} = \{C(\{x\},\{u\})\}\{x\} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}\{x\} \quad (2)$$

Where, the states vector is:

$$\{x\} = [v_{CoG} \quad \beta \quad \dot{\psi}]^T \quad (3)$$

The control inputs are:

$$\{u\} = [F_{LFL} \quad F_{LFR} \quad F_{LRL} \quad F_{LRR} \quad \delta_W]^T \quad (4)$$

And the measurement vector is:

$$\{y\} = \begin{bmatrix} v_{CoG} & \dot{\psi} \end{bmatrix}^T \tag{5}$$

With:

$$\begin{aligned} f_1 = \dot{v}_{CG} = \frac{1}{m_{CoG}} \{ & F_{LFL} \cos(\beta - \delta_W) + F_{LFR} \cos(\delta_W - \beta) + \\ & + \left[F_{LRL} + F_{LRR} - c_{aer} A_L \frac{\rho}{2} v_{CoG}^2 \right] \cos\beta + \left[c_{FL} \delta_W - c_{FL} \beta - \frac{1}{v_{CoG}} c_{FL} l_F \dot{\psi} \right] \sin(\beta - \delta_W) + \\ & + \left[c_{FR} \beta - c_{FR} \delta_W + \frac{1}{v_{CoG}} c_{FR} l_F \dot{\psi} \right] \sin(\delta_W - \beta) + \\ & + \left[\frac{1}{v_{CoG}} c_{RL} l_R \dot{\psi} - c_{RL} \beta - c_{RR} \beta + \frac{1}{v_{CoG}} c_{RR} l_R \dot{\psi} \right] \sin\beta \} \end{aligned} \tag{6}$$

$$\begin{aligned} f_2 = \dot{\beta} = \frac{1}{m_{CoG} v_{CoG}} \{ & - F_{LFL} \sin(\beta - \delta_W) + F_{LFR} \sin(\delta_W - \beta) + \\ & + \left[c_{aer} A_L \frac{\rho}{2} v_{CoG}^2 - F_{LRL} - F_{LRR} \right] \sin\beta + \\ & + \left[c_{FL} \delta_W - c_{FL} \beta - \frac{1}{v_{CoG}} c_{FL} l_F \dot{\psi} \right] \cos(\beta - \delta_W) + \\ & + \left[c_{FR} \delta_W - c_{FR} \beta - \frac{1}{v_{CG}} c_{FR} l_F \dot{\psi} \right] \cos(\delta_W - \beta) + \\ & + \left[\frac{1}{v_{CoG}} c_{RL} l_R \dot{\psi} - c_{RL} \beta - c_{RR} \beta + \frac{1}{v_{CoG}} c_{RR} l_R \dot{\psi} \right] \cos\beta \} - \dot{\psi} \end{aligned} \tag{7}$$

$$\begin{aligned} f_3 = \ddot{\psi} = \frac{1}{2 J_Z v_{CoG}} \{ & [c_{FL} \delta_W v_{CoG} b_F - c_{FL} \beta \; v_{CoG} b_F - c_{FL} l_F \dot{\psi} \; b_F + 2 F_{LFL} v_{CoG} l_F] \sin\delta_W + \\ & + [2 c_{FL} \delta_W v_{CoG} l_F - 2 c_{FL} \beta \; v_{CoG} l_F - 2 c_{FL} l_F \dot{\psi} \; l_F - F_{LFL} v_{CoG} b_F] \cos\delta_W + \\ & + [c_{FR} \beta \; v_{CoG} b_F - c_{FR} \delta_W v_{CoG} b_F + c_{FR} l_F \dot{\psi} \; b_F + 2 F_{LFR} v_{CoG} l_F] \sin\delta_W + \\ & + [2 c_{FR} \delta_W v_{CoG} l_F - 2 c_{FR} \beta \; v_{CoG} l_F - 2 c_{FR} l_F \dot{\psi} \; l_F + F_{LFR} v_{CoG} b_F] \cos\delta_W + \\ & + [2 c_{FL} + 2 c_{FR}] l_F \dot{\psi} \; n_{LF} + [- 2 c_{RL} l_R - 2 c_{RL} n_{LR} - 2 c_{RR} l_R - 2 c_{RR} n_{LR}] l_R \dot{\psi} + \\ & + [2 c_{FL} v_{CoG} n_{LF} + 2 c_{FR} v_{CoG} n_{LF} + 2 c_{RL} v_{CoG} n_{LR} + 2 c_{RL} v_{CoG} l_R + 2 c_{RR} v_{CoG} n_{LR} + 2 c_{RR} v_{CoG} l_R] \beta + \\ & + [- 2 c_{FL} v_{CoG} n_{LF} - 2 c_{FR} v_{CoG} n_{LF}] \delta_W - F_{LRL} v_{CoG} b_R + F_{LRR} v_{CoG} b_R \} \end{aligned} \tag{8}$$

Table 1 –Tire slip equations

	Braking, $v_R \cos\alpha \le v_W$	Acceleration, $v_R \cos\alpha > v_W$
Longitudinal slip	$s_L = \dfrac{v_R \cos\alpha - v_W}{v_W}$	$s_L = \dfrac{v_R \cos\alpha - v_W}{v_R \cos\alpha}$
Lateral slip	$s_S = \dfrac{v_R \sin\alpha}{v_W}$	$s_S = \dfrac{v_R \sin\alpha}{v_R \cos\alpha} = \tan\alpha$

Where δ_W is the steering wheel angle, and F_{Lij} are the longitudinal wheel forces. With m_{CoG} as the vehicle mass, A_L as the frontal vehicle area, ρ the air density, c_{aer} the vehicle's aerodynamic drag coefficient, the distances l_F and l_R and the casters n_{ij} defined on Fig. 1, c_{ij} the individual wheels cornering coefficients, J_Z the vehicle's mass moment of inertia around the Z axe. b_F is the frontal vehicle axis width and b_R the vehicle rear axis width. The right and left steering angles are considered to be the same. What we want is to control the longitudinal velocity, the beta angle and the yaw rate using the longitudinal forces on the contact with the road. These forces manipulation is possible only with the brakes.

With the brakes application, the wheel rotation is changed, and consequently its rotational velocity. When the rotational velocity is changed, the wheel slip is changed as well. As the forces on the tires are proportional to the slip, one can change these forces just by the utilization of the brakes. The tire slips are in accordance to the equations on Tab. 1. The wheel steering angle and the longitudinal forces are the control inputs for the vehicle dynamics control through the steering and the presence of an appropriated braking pressure.

3. Yaw control

The strategy to maintain the vehicular dirigibility is to restrain the sideslip angle β and the yaw rate $\dot{\psi}$, so that the two values are between an inferior and a superior limits. Besides this strategy, it is important to note the angle β time-derivative, because high values in this time-derivative could result in some vehicular instability and too low values could result in slow vehicle reaction to the driver steering inputs.

The vehicle body side-slip angle β is limited, uppermost by the vehicle velocity v_{CoG}:

$$\beta_{max} = 10° - 7° \frac{v_{CoG}^2}{(40\, m/s)^2} \tag{9}$$

The reference side-slip angle is then given by:

$$\beta_{ref} = \begin{cases} \beta \;, & |\beta| \le |\beta_{max}| \\ \pm\beta_{max} \;, & \text{otherwise} \end{cases} \tag{10}$$

In the case of over-steering, the yaw rate $\dot{\psi}$ shall also be limited by the yaw dynamics control. From the vehicle model, the maximum possible yaw rate value is:

$$\dot{\psi}_{max} = \frac{1}{v_{CoG} \cos\beta}\left(a_{Ymax} - \dot{v}_{CoG} \sin\beta_{ref}\right) - \dot{\beta} \tag{11}$$

Some authors consider that the value for the side-slip angle time-derivative is null. According to our research, this is not totally correct in all situations, thus in the Eq. (11) we have used $\dot{\beta}$. This way, the reference yaw rate is then given by:

$$\dot{\psi}_{ref} = \begin{cases} \dot{\psi} \;, & |\dot{\psi}| \le |\dot{\psi}_{max}| \\ \pm\dot{\psi}_{max} \;, & \text{otherwise} \end{cases} \tag{12}$$

In the case of under-steering the angle β and the yaw rate are below their maximum. If the tire side-slip angle α rises overly, the vehicle leaves the desired course. The rear tire side-slip angle α_R is used as reference to determine when the front tire side-slip angle α_F reaches a critical value. The relation:

$$\left|\frac{\alpha_F}{\alpha_R}\right| = 1.5 \tag{13}$$

Is here used. Thus, in the case of under-steering the reference yaw rate is:

$$\dot{\psi}_{ref} = \begin{cases} \pm\dot{\psi}_{max} \;, & \left|\frac{\alpha_F}{\alpha_R}\right| \ge 1.5 \\ \dot{\psi} \;, & \text{otherwise} \end{cases} \tag{14}$$

When $|\beta| > |\beta_{max}|$, $|\dot{\psi}| > |\dot{\psi}_{max}|$ or $\left|\frac{\alpha_F}{\alpha_R}\right| = 1.5$, the yaw control becomes active.

3.1. Fuzzy logic yaw control system

As described above, it is possible to determine reference values for the yaw rate and the vehicle body sideslip angle. In the case that the actual yaw rate value is larger than the maximum tolerated by the road, the vehicle has an over-steering behavior. When the yaw rate value is lower than the maximum tolerated by the road, but the frontal tire sideslip angle α_F has reached its critical value, the vehicle has an under-steering behavior. In both cases, the control shall actuate in order to correct the vehicle trajectory.

The presented control utilizes a fuzzy logic system to determine the desired tire sleep for each tire. Defining a yaw rate relative error as:

$$e = \frac{|\dot{\psi}| - |\dot{\psi}_{des}|}{\max\left(|\dot{\psi}|, |\dot{\psi}_{des}|\right)} \tag{15}$$

With $\dot{\psi}_{des}$ as the yaw rate reference value. If $e > 0$, then the vehicle is over-steered and the yaw rate shall be reduced. If $e < 0$ then the vehicle is under-steered and the yaw rate shall be raised.

To correct the measured yaw rate, the controller determines and designates the desired wheel input slip. The relative error from Eq. (15) is the fuzzy logic input. The fuzzy logic system has four fuzzification membership functions, for each individual vehicle wheel. These functions, shown on Fig. 2, are just the same, and have the slip value as result. The differences between the wheels are only the rule table elements, shown on Tab. 3. This logic is responsible to designate the correct slip value to each inner or outer, rear or front, wheel. The output desired slip function is shown on Fig. 3.

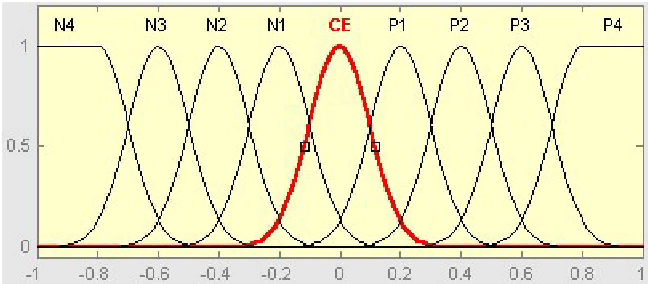

Figure 2 – Fuzzification membership function to the yaw fuzzy control

Table 2 – Rule table for the yaw fuzzy control

Input e	Outer front	Outer rear	Inner front	Inner rear
N4	CE	HI	CE	HI
N3	CE	HI	CE	MH
N2	CE	HI	CE	MD
N1	CE	CE	CE	LO
CE	CE	CE	CE	CE
P1	LO	CE	CE	CE
P2	MD	CE	HI	CE
P3	MH	CE	HI	CE
P4	HI	CE	HI	CE

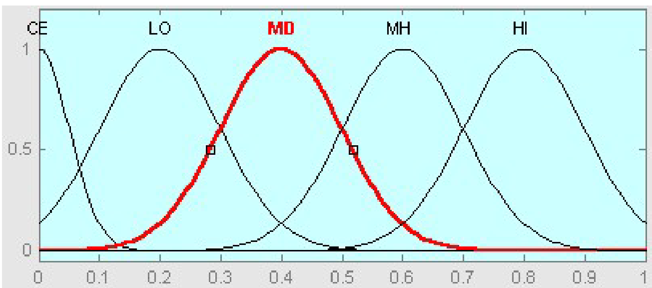

Figure 3 – Defuzzification membership function to the yaw fuzzy control

According to each input value, a different control strategy is used:

- Controller input $e > 0$

If the fuzzified input belongs to *P2*, *P3* or *P4*, then the inner frontal wheel shall be used as assistant to the inner rear wheel. The other two wheels slip must be selected as *CE* to allow the best response potential on the longitudinal direction (to improve braking) and on the lateral direction (to improve steering).

- Controller input $e < 0$

If the fuzzified input belongs to *N2*, *N3* or *N4*, then the outer rear wheel shall be used as assistant to the outer front wheel. The same as above, the other two wheels slip must be selected as *CE* to allow the best response potential.

Having the slip values determined to each wheel, the braking torque variation depends on the difference between the estimated slip and the desired slip. In the case that the estimated slip value is 1% or more above the desired slip value, the braking torque variation is given by Eq. (16):

$$\Delta T_{Br}(i) = -kT_{Br}(i)\left(\left|s_{res}(i)\right| - s_{des}(i)\right) \tag{16}$$

In the case that the estimated slip value is 1% or more below the desired slip value, the torque variation is given by Eq. (17):

$$\Delta T_{Br}(i) = +100 + T_{Br}(i)\left(s_{des}(i) - \left|s_{res}(i)\right|\right) \tag{17}$$

On Eq. (16) and Eq. (17), $s_{des}(i)$ is the desired slip value that results from the fuzzy estimator.

4. Results

To determine the efficacy of the proposed control system were simulated cornering maneuvers with braking and acceleration, constant steering angles or known generic steering inputs (such as sinusoidal or step), and with or without braking and acceleration. Several results were obtained, and three different maneuvers were chosen to demonstrate the proposed control efficient and functionality. These maneuvers are described on Tab. 4.

Table 3 – Yaw control simulations parameters

	Simulation 1	Simulation 2	Simulation 3
Initial velocity	20 *m/s*	20 *m/s*	30 *m/s*
Acceleration	No	No	6 *MPa* Step in t=3s
Terrain	Wet cobblestones	Wet cobblestones	Wet asphalt
Steering	Sinusoidal 7º and 0.7 *Hz*	0,2 *rad* Step from t = 0 to 2 *s*, -0,2 *rad* Step until t = 4 *s*.	0.1 *rad* fixed

The described simulations were made with different values for the control constant k and for the system without control. The simulations results resume can be seen on Tab. 5. The table shows the maximum obtained values for $\dot{\psi}$, $\dot{\beta}$ e β. Initially, we used these values to determine quantitatively the proposed control efficiency, according to each control constant value. After the initial quantitative analysis, the obtained graphics were the base for a qualitative analysis.

Table 4 –Yaw control simulations results (maximum $\dot{\psi}$, $\dot{\beta}$ e β.)

Simulation	1	2	3
No control	0,541 / -0.415 / 5,913	-0,549 / 0,404 / 53,462	0,385 / -0,168 / -4,143
Yaw fuzzy control			
k = 0.1	0,417 / 0,386 / 6,902	0,475 / 0,468 / 27,819	0,478 / 0,339 / -10,444
k = 0.5	0,412 / 0,271 / 5,362	0,958 / 0,954 / 60,153	0,420 / -0,208 / -4,406
k = 1	0,403 / 0,265 / 4,847	0,961 / 0,943 / 78,285	0,407 / -0,190 / -3,980
k = 5	0,365 / 0,218 / 4,485	0,953 / 0,933 / 50,505	0,390 / -0,169 / -3,569
k = 10	0,388 / 0,237 / 3,263	0,548 / 0,521 / 12,637	0,386 / -0,168 / -3,532

Taking into account the three analyzed variables, the most relevant to determine the vehicle instability is the angle β value. In an under-steering situation, the yaw rate during the maneuver is lower than the maximum possible yaw rate for the situation. Thus, when the control system enters in operation, the obtained yaw rate value is larger than that value without the control, as shown on the obtained results from Simulation 3 (on Tab. 5). In the case that the yaw rate value is the only value used to determine the control system efficiency, judgment errors could be made. In the same way one need to determine the control system efficiency or the vehicle instability taking into account the $\dot{\beta}$ value.

In such situations as when the driver needs to make a very quick maneuver to detour a sudden obstacle (as, for example, when a child or an animal runs in front of the vehicle), it is desirable a high $\dot{\beta}$ value. Without a fast variation on the vehicle attitude, it could be difficult to avert the obstacle in time. In the case of the vehicle body sideslip angle β, contrary to the other three variables, a high value always means that the vehicle moves in a direction distinct from the direction it faces. When the vehicle is not facing its movement direction, it skids laterally and is harder to control and more unstable.

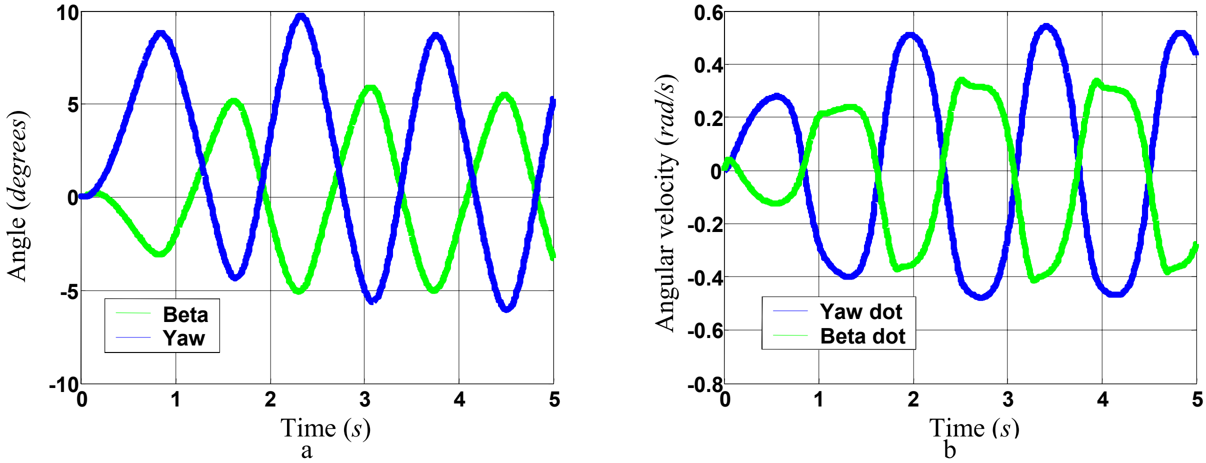

Figure 4 –Simulation 1, no control. (a) Angles beta and yaw. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

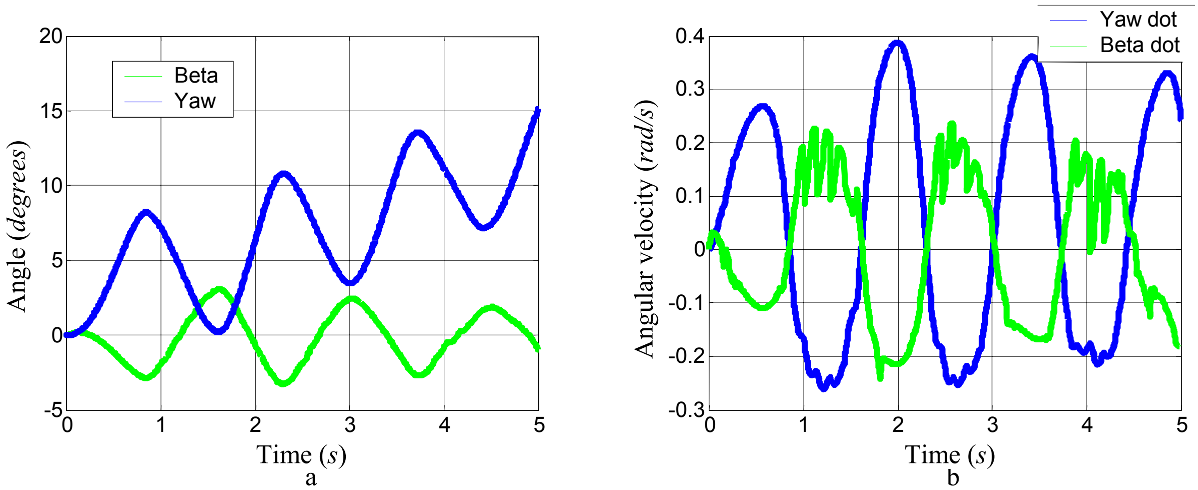

Figure 5 –Simulation 1, fuzzy logic control (k = 10). (a) Angles beta and *yaw*. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

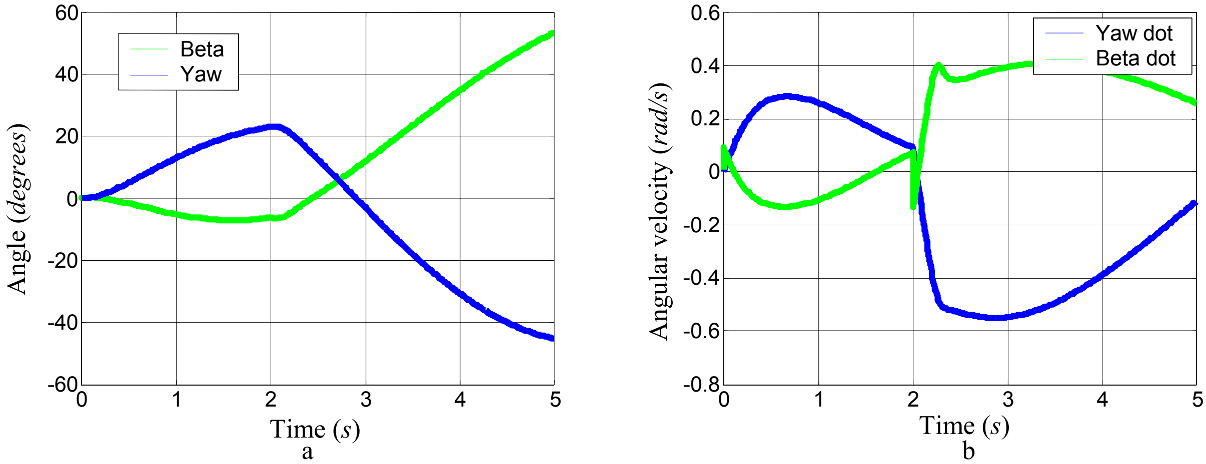

Figure 6 –Simulation 2, no control. (a) Angles beta and *yaw*. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

The first simulation results can be seen on Fig. 4, without the fuzzy control, and on Fig. 5, with the fuzzy control. In the case shown on Fig. 4 the yaw rate peak value does not diminish on each successive steering cycle. In a simulation with the same steering input but on a higher friction coefficient road, the yaw rate peak value on each successive cycle should diminish. This is the behavior obtained with the fuzzy logic control, shown on Fig. 5. With the fuzzy control the

$\dot{\psi}$ and $\dot{\beta}$ peak values diminish on each successive cycle, maintaining the vehicle stable. On the lower friction coefficient road, these variables keep on high peak values, what shows a bigger vehicular instability, with bigger β angle values.

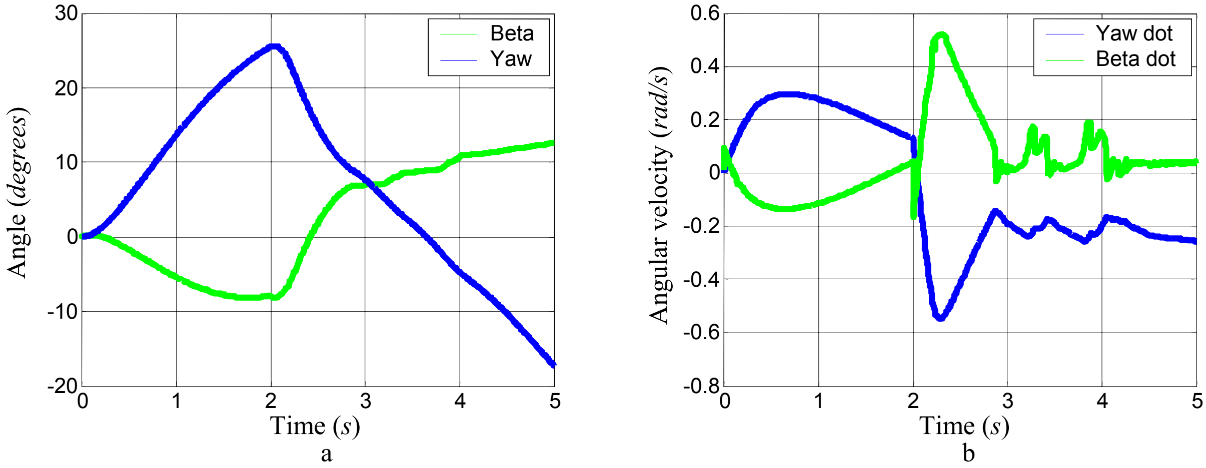

Figure 7 –Simulation 2, fuzzy logic control (k = 10). (a) Angles beta and *yaw*. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

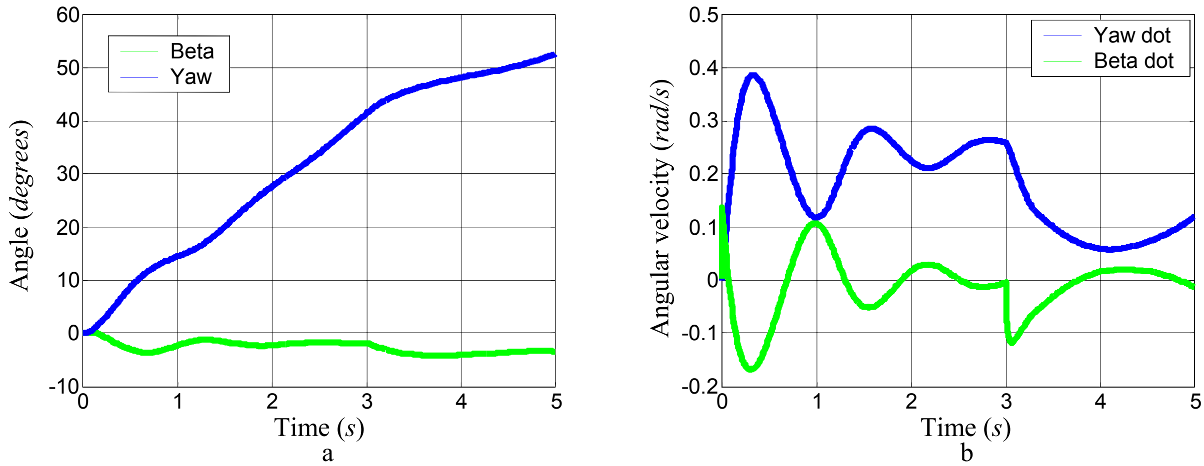

Figure 8 –Simulation 3, no control. (a) Angles beta and *yaw*. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

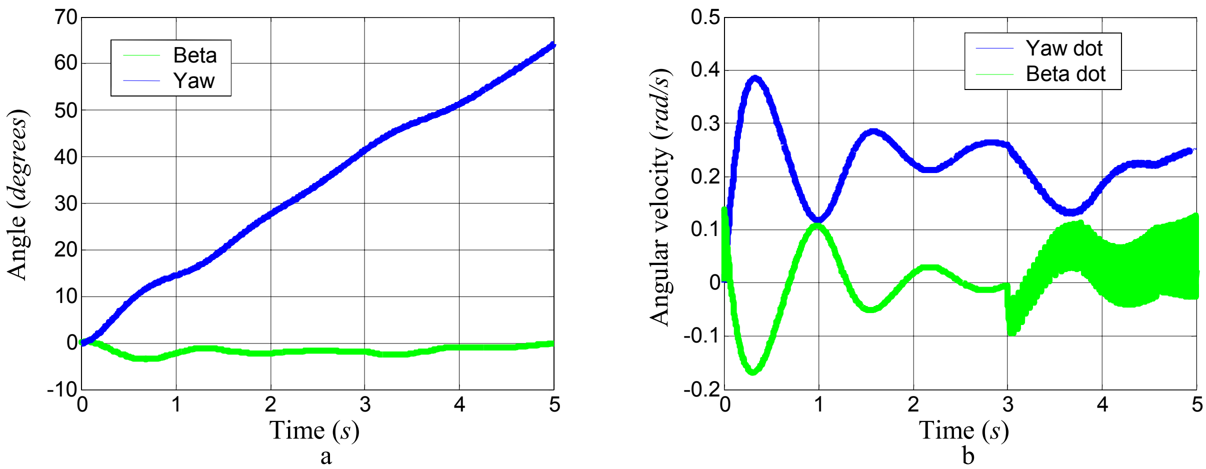

Figure 9 –Simulation 3, fuzzy logic control (k = 10). (a) Angles beta and *yaw*. (b) Angles variation ($\dot{\psi}$, $\dot{\beta}$).

The second simulation maneuver shows clearly how important is to evaluate the variables together. As shown on Fig. 6, notwithstanding the not so high $\dot{\psi}$ and $\dot{\beta}$ values, the angle β value, after the second steering step input, rises quickly. Together with a yaw angle increase of similar magnitude, but with opposite signal, this variables behavior shows that the vehicle has turned around the Z axis, with only a small change on its movement direction.

The same situation with the fuzzy logic based yaw momentum control shows a very distinct behavior, as shown on Fig. 7. The first steering step response is the same on both situation and after the second step steering input of -0,2 *rad* on both situation the $\dot{\psi}$ and $\dot{\beta}$ values change abruptly. But with the fuzzy control the eminent instability is corrected, and the $\dot{\beta}$ value approaches zero. The simulation with the fuzzy logic control shows that the vehicle on this situation follows better the desired course. Although the angle β value increases even with the control, this increase is low enough to enable the driver to control the vehicle easily.

The third maneuver simulation without the control is shown on Fig. 8. Before the braking actuation, on t = 3 *s*, the vehicle is completely stable. With the steering beginning $\dot{\psi}$ and $\dot{\beta}$ reach a peak value and then oscillate. The yaw rate value tends towards a value between 0,2 *rad/s* and 0,3 *rad/s* and the $\dot{\beta}$ value tends towards zero on steady state. When the brake is pressed, without the yaw dynamics control or some kind of antilock braking control, some wheels lock-up. Locked-up wheels do not transfer lateral forces, thus $\dot{\psi}$ and $\dot{\beta}$ diminish, tending towards zero. This way the vehicle can't keep cornering.

With the yaw dynamics fuzzy based control, shown on Fig. 9, this lock-up does not occur. Because the fuzzy control determines each tire longitudinal slip, the wheels do not lock and the vehicle keeps the desired course.

5. Conclusion

The satisfactory vehicle handling is obtained all the time that the vehicle maintains a certain path that reflects the maneuver steering angle without losing stability. The concepts of stable and steerable are very subjective, because they mainly depend on the driver ability to control the vehicle with little effort. When projecting a vehicular safety system one needs to consider the standard driver, and not the real professionals.

To test and implement the fuzzy logic based control strategy, we developed and simulated a vehicular model using a commercial software product. Due to the difficulties to model mathematically all the complex physical phenomena that occur on the tire-road contact patch, some simplifications were made to the model. These simplifications include no suspension or actuators dynamics, no movements on the Z axis direction and no pitch or roll rotations. These simplifications were meant to improve and facilitate the implementation, and they do not spoiled the obtained results. The implemented model was very consistent when compared with the reference literature and with commercial programs.

It is desired that the vehicle stays on the intended course, but the most important control objective is to obtain a safe vehicle operation and to enable the driver to easily control the vehicle. Thus, the vehicular control systems do not need to drive the vehicle, but to make its operation safer for the driver, the passengers and the pedestrians.

To the yaw momentum control, we developed and simulated a fuzzy logic based yaw dynamics control. Because the fuzzy logic based control determines individually e more precisely the desired longitudinal slip to each wheel, it is very efficient in keeping the vehicle stable and on the desired course.

The fuzzy logic system allows the vehicular control even in the cases that the vehicular model is unknown. The obtained results were satisfactory beyond expectative.

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