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BRAKING SIMULATION AND FUZZY LOGIC CONTROL OF PASSENGER CARS FOR VEHICULAR SAFETY

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Abstract. *This paper presents an assisted braking control using fuzzy logic. The intention of a braking control is to prevent wheel lock-up. These lock-ups induce vehicular stability loss and longer braking distances. The vehicle is modeled as a seven degrees of freedom body, with the yaw rotation, the angle between the vehicle X axis and his velocity vector and the Center of Gravity velocity, besides the four degrees of freedom related with the wheels rotation. The fuzzy control system have as input the slip and its instantaneous variation, and it is composed of 25 rules. The fuzzy control output is the braking torque variation constant. From the estimated vehicle velocity and the corrected wheel velocities, the slip values can be calculated. The slip values are more appropriated for the classification of the movement situations than the velocities of the wheels. The results obtained show that the presented control is very effective, and does not need an additional logic in the case of differents friction coefficients between the vehicle sides, as the actual braking controls do. Also, the presented control reaches the desired slip value really fast.*

Keywords*: Vehicular safety, fuzzy logic, braking, vehicular simulation, vehicular control*

1. Introduction

The braking system is a very challenging control problem because it is extremely non-linear with time varying parameters. Intelligent controls, with fuzzy logic or neural networks, eliminate this problem. Fuzzy controls have even more advantages because they do not need the mathematical model of the system, while still keeping robustness. Also, certain fuzzy control designs can be implemented with the ability to learn or self adapt in order to improve their performance.

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 and 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.

The vehicle brakes are responsible for important roles as decrease the vehicle velocity, stop the vehicle while in movement and keep the vehicle parked, once it has been stopped. Under normal conditions, the modern braking systems can make the vehicle brake fast and effectively. But, braking under more critical situations, as in a snow covered road, could lead to wheel lock-up, resulting in stability and dirigibility loss, as the vehicle looses traction and skids on the road.

The braking control systems hinder the wheels lock-up during braking maneuvers. This happens mostly on low friction surfaces and with high brake forces. The wheels lock-up leads to longer stopping distances and stability loss, because the locked wheels can't generate lateral forces. A braking control system must maintain the vehicle stability independent of surface conditions and changes on these conditions (Solyom & Rantze, 2002). The braking control shall also limit the yaw momentum build-up due to different friction coefficients between the vehicle sides. For this reason, the commercial braking control strategies have some additional logic, what increases the system complexity.

The braking control strategies can be divided in two groups: wheel acceleration control and wheel slip control. The first group uses the measured wheel angular velocity to determine the wheel acceleration. This control strategy is to regulate the slip indirectly by controlling the wheels acceleration/deceleration using variations to the braking pressure. According to some acceleration and deceleration value thresholds, the braking pressure is raised, kept or lowered, thus avoiding the wheels look-up during braking.

The second group needs the slip estimative, and adjusts the desired slip value as the value where the friction coefficient has its maximum value. Then the control system tries to maintain near zero the error value between the actual slip value and the desired slip value, increasing or reducing the braking pressure accordingly.

This paper presents, develops and simulates a wheel slip control using fuzzy logic. The vehicle is modeled as a seven degrees of freedom body, and the fuzzy control system have as input the slip and its instantaneous variation, and it is composed of 25 rules. The slip values can be calculated from the estimated vehicle velocity and the corrected wheel velocities. They are more appropriated for the classification of the movement situations than the velocities of the wheels. The obtained fuzzy logic output is the constant necessary for the vehicle braking control.

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.

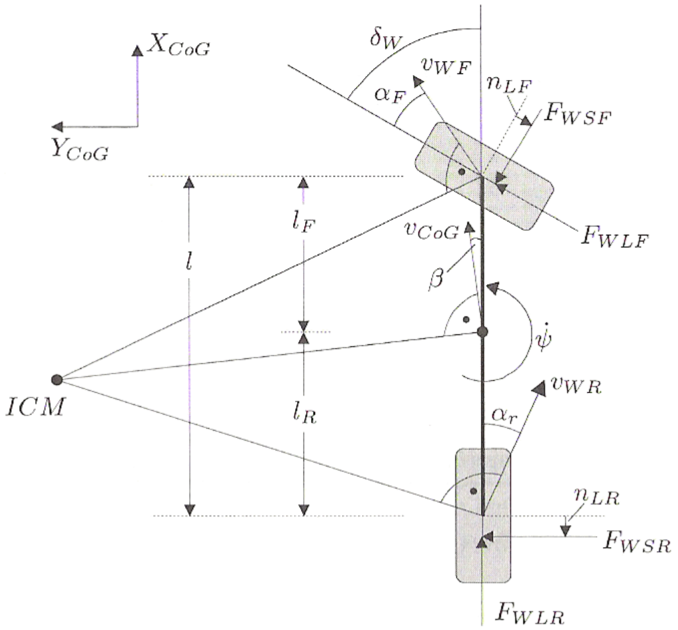


Figure 1 – Single-track representation of the vehicle model (Kiencke & Nielsen, 2000).

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 model equations, in the state space form, are:

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

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

Where, the state vector is:

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

The control inputs are:

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

And the measurement vector is:

$$\{y\} = [v_{CoG} \quad \dot{\psi}]^T \tag{5}$$

With:

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

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

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

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. It is intended on this study 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.

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	$s_S = \dfrac{v_R \sin\alpha}{v_W}$	$s_S = \dfrac{v_R \sin\alpha}{v_R \cos\alpha} = \tan\alpha$

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. Braking control

In order to stop the car on the shortest possible braking distance, the braking control shall actuate with the slip that originates the maximum friction coefficient. This optimal slip value depends on the road surface. The braking control system actuates in a range of values, near the peak of the friction coefficient versus slip curve, shown on Fig. 2. According to Fig. 2, a slip value around 20% is a good value for every surface. The shortest stop distances are obtained when the wheels have the slip that corresponds to the maximum friction coefficient.

1 Dry concrete
2 Wet asphalt
3 Snow
4 Wet black ice
Shadowed areas: desired control

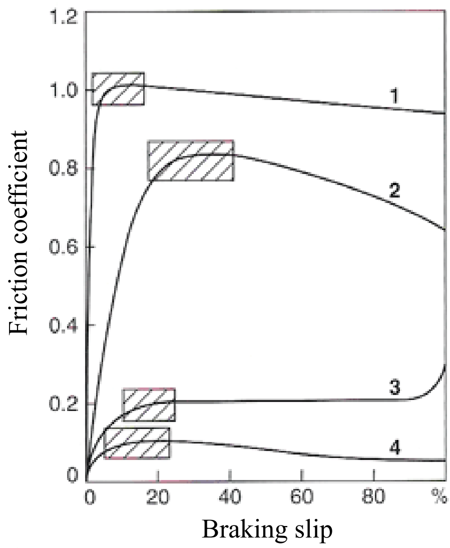


Figure 2 – Desired slip area for braking control (Bosch, 1995)

After the maximum friction coefficient point, the friction curve gradient changes its signal. Thus the system becomes unstable resulting, in the case of lack of control, in wheel rotational deceleration too high, with the final result of wheel look-up.

In order to control the tires slips, it is necessary to obtain the slips time variation. The wheel torque balance is given by:

$$J_W \dot{\omega} = r_{eff}\, \mu_L(s_L) F_Z - r_{eff}\, k_{Br}\, p_{Br} \tag{9}$$

From this equation, taking into account that the vehicle velocity varies more slowly than the other variables, we obtain the slip time variation equation:

$$\dot{s}_L = \left(-\frac{1}{v_{CoG}}\right)\left(\frac{(1-s_L)}{m} + \frac{r_{eff}^2}{J_W}\right)F_L + \left(\frac{1}{v_{CG}}\right)\frac{r_{eff}}{J_W}T_{Br} \tag{10}$$

For the present fuzzy logic control, we have two inputs. The first is the slip time variation, given by Eq. (10). The other is the relative difference (s_{Le}) between the actual estimated slip (s_L) and the desired slip value (s_{Ldes}). Thus, the second input is given by Eq. (11):

$$s_{Le} = \frac{(s_L - s_{Ldes})}{|\max(s_L, s_{Ldes})|} \tag{11}$$

Where s_L is the estimated longitudinal slip value (negative for braking) and s_{Ldes} is the desired slip value. For this work purposes, we assumed s_{Ldes} = -0,20, or 20% desired wheel slip.

To determine the rules to be used, we have taken as basis the work of Lee & Tomizuka (1995). This was necessary because, to determine the rules for a fuzzy system, a great system knowledge, experimentation and practical experience with fuzzy logic are required. Thus we used some previous known fuzzy system behavior to determine the ideal rule table for the present control.

Twenty-five rules were used, as shown on Tab. 2. For example, if the slip relative error s_{Le} is positive (PG or PP) and keeps increasing ($\dot{s}_L$ = PG or PP) then the braking input torque must be reduced (*val* = NG or NP).

Table 2 – Rules table for the slip fuzzy control

		s_{Le}				
		NG	NP	ZE	PP	PG
$\dot{s}_L$	NG	PG	PG	PP	NP	NP
	NP	PG	PG	PP	NP	NP
	ZE	PG	PP	ZE	NP	NG
	PP	PP	PP	NP	NG	NG
	PG	PP	PP	NP	NG	NG

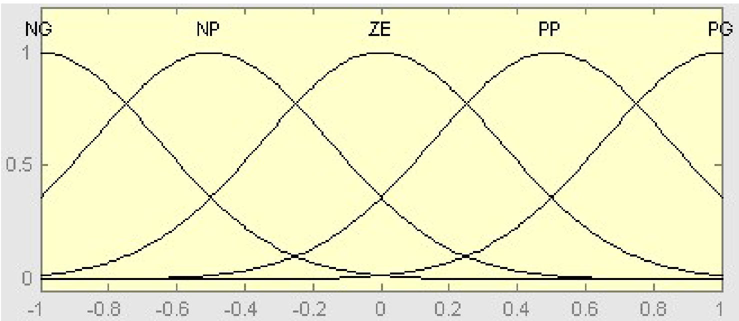


Figure 3 – Fuzzification membership functions for the slip relative error (NG = negative big, NP = negative small, ZE = zero, PP = positive small and PG = positive big)

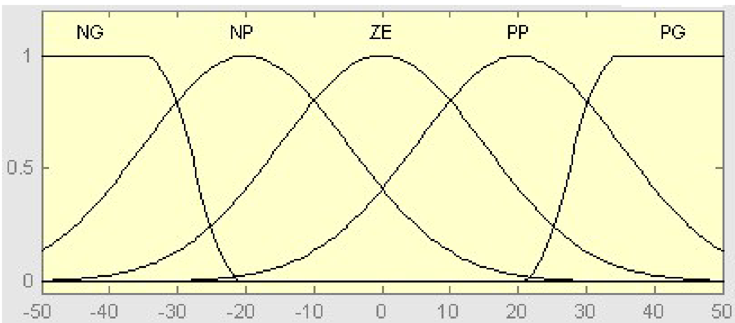


Figure 4 – Fuzzification membership functions for the slip time variation (NG = negative big, NP = negative small, ZE = zero, PP = positive small and PG = positive big)

The fuzzy logic output is used as the control constant k_I according to Eq. (12). This equation shows the braking torque variation, where T_{Br} is the current braking torque and i are indices for the individual wheels (1 for left-front, 2 for right-front, 3 for left-rear and 4 for right-rear).

$$\Delta T_{Br}(i) = -k_1(i) T_{Br}(i) \qquad (12)$$

In this braking fuzzy control, when the input values s_{Le} are negative, i. e., the estimated slip modulus is bigger than the desired slip modulus, the output is a positive value. This results in a braking torque reduction. In the case of positive values for the input s_{Le}, i. e., the estimated slip modulus is less than the desired slip modulus, the fuzzy logic estimative returns a positive value. This results in a braking pressure increase.

The membership functions for the input slip relative error are shown on Fig. 3. Figure 4 shows the membership functions for the input slip time variation and the defuzzification functions can be seen on Fig. 5.

The control surface for the described fuzzy logic braking control can be seen on Fig. 6.

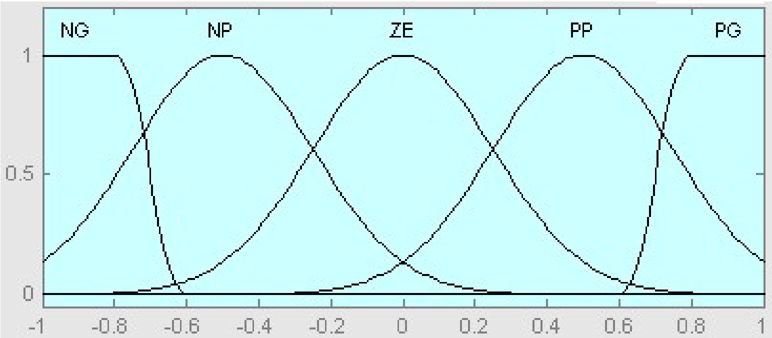


Figure 5 – Braking control defuzzification membership functions (NG = negative big, NP = negative small, ZE = zero, PP = positive small and PG = positive big)

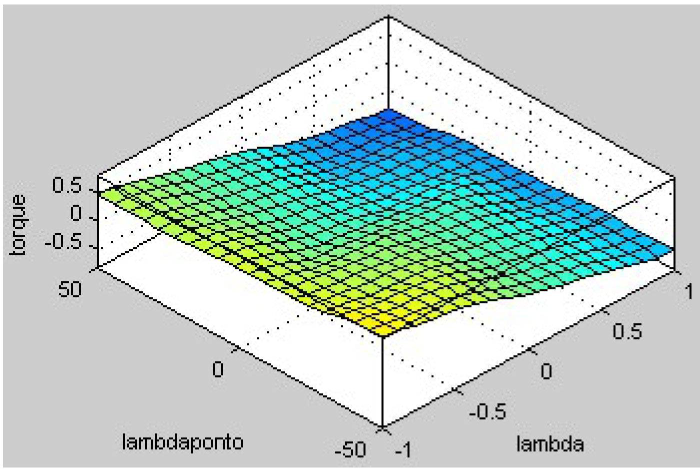


Figure 6 – Control surface for the braking fuzzy control

3.1. μ-split Surfaces braking control

Aside from preventing wheel lock-up during braking maneuvers in low friction coefficient roads, an assisted braking system shall maintain the vehicle stable during braking maneuvers in surfaces with different friction coefficients between the vehicle sides, called μ-split roads. This situation occurs, for example, when the driver pushes the brakes with the left wheels on dry road and the right wheels in a pool (wet asphalt) or on the side of the road made of a different material, as grass or sand. In such situation the vehicle loses stability and it is difficult to steer. This happens because the braking forces on the tire-road contact that have the larger friction coefficient are bigger than the forces on the low friction coefficient side of the road

The great difference in the forces generated between the vehicle sides results in a yaw momentum around the Z axis. This yaw momentum increases as the vehicle travels, and the driver needs to steer the vehicle in order to keep it stable. Figure 7 shows this situation without the braking control.

This is obviously an unstable driving situation. The angles time derivatives reach a high value. For this simulation, the left wheels are subjected to a higher friction coefficient. As expected, the vehicle turns around the Z axis following a counterclockwise movement, while still moving towards the same direction.

To make sure that the vehicular stability is maintained, even in the case of different friction coefficients values between the sides of the vehicle, some additional yaw momentum reduction logic is required. This logic is as follow:

In the beginning of the braking maneuver, the braking torque values applied on wheels on the same axle are the minimum controlled values. After the initial control cycle, the braking torque on the frontal wheel which has not yet achieved a wheel slip threshold is slowly raised, until this threshold is achieved. After this achievement, the control proceeds as normal. In the case of the rear wheels, the braking torque value is always the same. This value is always the lower possible value. To better evaluate the fuzzy logic braking control behavior, we made simulations with and without this additional logic.

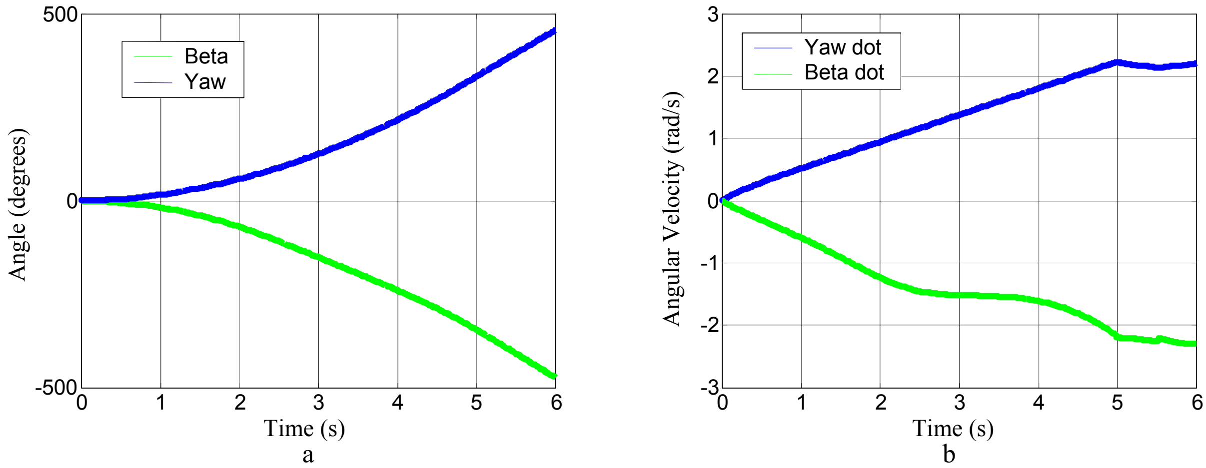


Figure 7 – Braking on μ-split road surface, no braking control. (a) Angles beta and yaw. (b) Angles variation ($\dot{\beta}$ and yaw rate – $\dot{\psi}$).

4. Results

In order to evaluate the proposed fuzzy logic braking control, we made three different simulations, according to Tab. 3. The initial velocity for the three simulations was the same, as well as the braking pressure applied. The final velocity for Simulation 1 (dry asphalt) and Simulation 2 (wet asphalt) was 0 *m/s* and in the case of Simulation 3 (snow), it was 10 *m/s*.

Table 3 – Simulation parameters for the braking control simulations

	Simulation 1	Simulation 2	Simulation 3
Initial velocity	20 *m/s*		
Braking	8 *MPa* Step in $t = 0$ *s*		
Final velocity	0 *m/s*	0 *m/s*	10 *m/s*
Terrain	Dry asphalt	Wet asphalt	Snow

The simulations numerical results can be seen on Tab. 4. This table shows the time elapsed from the beginning of the braking maneuver to the desired final velocity. The simulations graphical results can be seen on Fig. 8 for Simulation 2 (wet asphalt) and on Fig. 9 for Simulation 3 (snow), both cases with the control active.

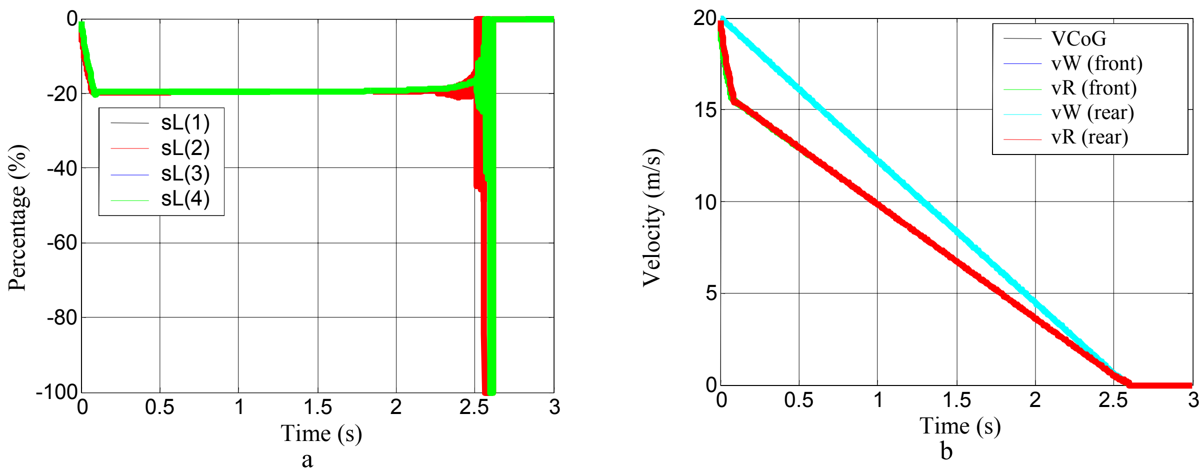


Figure 8 – Fuzzy logic braking control. Simulation 2. (a) Longitudinal slip on wet asphalt road. (b) Wheel velocities on wet asphalt road.

Table 4 – Braking control simulations results (time to final velocity)

Simulation	1	2	6
No control	2,25	3,88	7,50
With fuzzy logic slip control	2,23	2,62	5,44

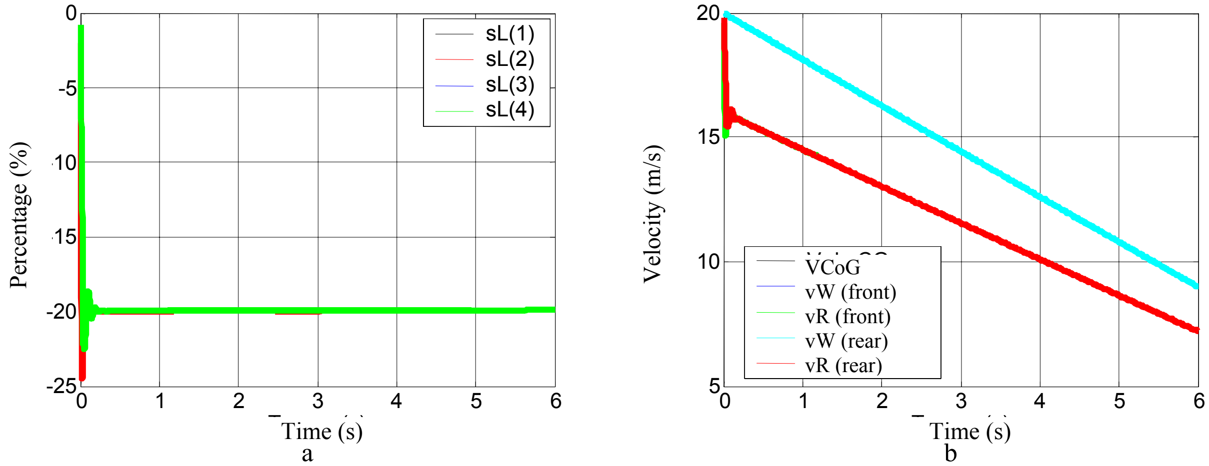


Figure 9 – Fuzzy logic braking control. Simulation 3. (a) Longitudinal slip on snow covered road. (b) Wheel velocities on snow covered road.

According to the simulations, the proposed braking control is very effective and has a very fast response time. In a few iterations the tire slip value reaches the desired tire slip value, and this value is maintained throughout the maneuver. This behavior suggests that coupled with an adaptive control strategy, which could find the optimal slip value for each road condition, the fuzzy logic braking control proposed would be even more efficient.

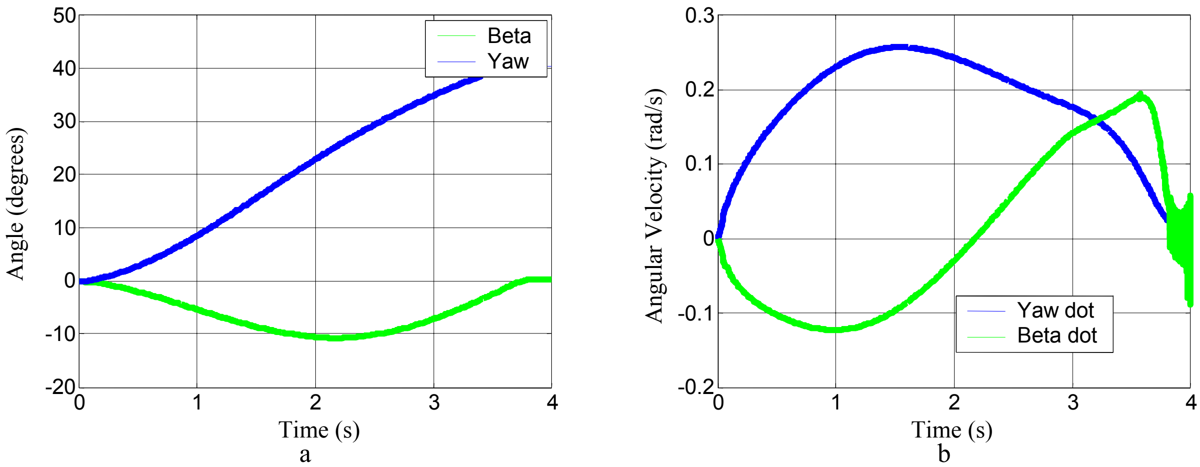
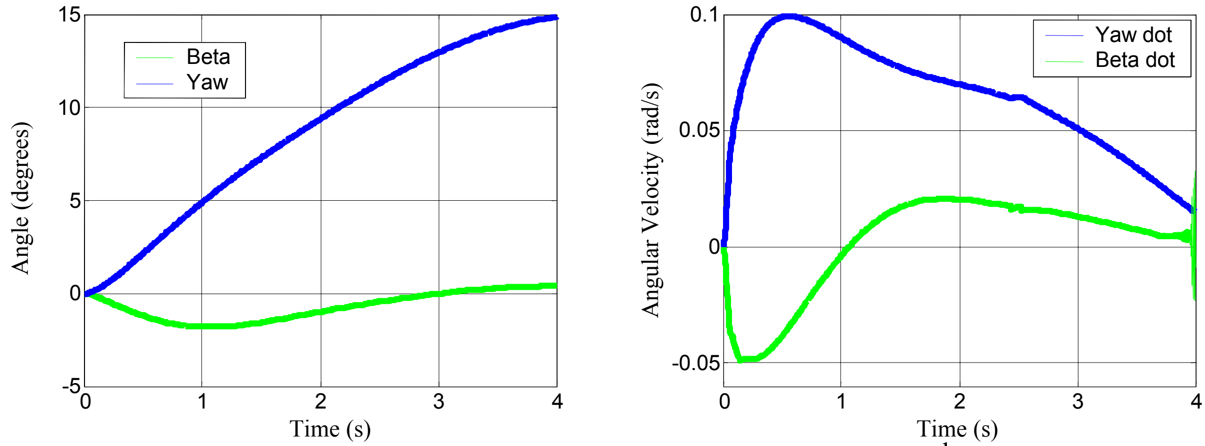


Figure 10 – Braking on μ-split road surface, fuzzy logic control without additional yaw reduction logic. (a) Angles beta and yaw. (b) Angles variation ($\dot{\beta}$ and yaw rate – $\dot{\psi}$).

Because the proposed fuzzy logic braking control controls each wheel individually, it has shown a good behavior in a braking maneuver on a μ-split road surface without the additional logic, as shown on Fig. 10. On a first instance the vehicle has turned around the Z axis, with no change on the movement direction. But the beta value does not rise beyond 10 degrees. After this point, $\dot{\beta}$ changes sign, and the vehicle goes towards the direction given by the yaw angle. This behavior shows that the driver could steer the vehicle, keeping it stable and on the desired direction.

The result for the proposed control, with the additional logic, can be seen on Fig. 11. Here the maximum $\dot{\beta}$ and $\dot{\psi}$ are well below the maximum obtained without the additional logic and the beta value decreases early. With the additional logic the control is more stable and the vehicle easier to steer, but even in the simulation without the additional logic the vehicle remains stable and steerable.



a b

Figure 11 – Braking on μ-split road surface, fuzzy logic control with additional yaw reduction logic. (a) Angles beta and yaw. (b) Angles variation ($\dot{\beta}$ and yaw rate – $\dot{\psi}$).

5. Conclusion

The braking control strategy developed exhibited more effective results than the majority of the commercial control methods. During braking, the commercial braking control methods allow some short wheel lock-up, while the proposed methods exhibited practically no wheel lock-up.

As the simulations results show, the fuzzy logic braking control is very efficacious. The desired longitudinal slip value is reached really fast. Because of this, the implemented fuzzy control could be even more efficient if some adaptive logic were used to obtain the optimum slip value for each road condition.

For a braking maneuver on a μ-split road surface, the developed additional system was based on the commercial systems. With this additional logic, better results were obtained, and the $\dot{\beta}$, $\dot{\psi}$ and β values obtained are well below the values obtained without the yaw moment reduction system, but the system is more complex. The obtained values allow the vehicle to remain stable and easily controllable by the driver.

The fuzzy logic braking control has worked well enough even without the additional logic. The results show that this control does not need an additional logic to handle a μ-split road surface situation, thus making the system less complex, what improves reliability and helps lower the implementation costs. Besides, without the additional logic, the vehicle can stop in a shorter time, because the fuzzy logic control applies the higher possible braking torque on each wheel, and the proposed additional logic in the beginning of the braking maneuver uses the lower possible braking torque.

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