# Modeling and Simulation of a Traction Control Algorithm for an Electric Vehicle with Four Separate Wheel Drives


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***Abstract*—The aim of this paper is to present a traction control algorithm for an electric vehicle (EV) with four separate wheel drives. This algorithm is necessary to improve the handling and stability of EV during cornering or under slippery road condition. It distributes the traction power among four drives and especially an independent torque reference on each wheel. The proposed algorithm is implemented in terms of a hierarchical architecture, which incorporates all new known vehicle systems ABS (Anti-lock Brake System), ASR (Anti Slip Regulation), ESP (Electronic Stability Program). To achieve good performances a correlation of the traction controller with motor performances has been implemented. Several simulation results, which show the potential of such a algorithm, are presented.**



***Keywords— EV; modeling; simulation; traction control.***


## I. Introduction

The principal constraints in vehicle design for personal mobility are the development of a non-polluting high safety and comfortable vehicle. Taking into account these constraints, our interest has been focused on the 4 X 4 electrical vehicle, with independent driving in-wheel-motor at the front and with classical motors on the rear drive shaft. This configuration is a conceivable solution, the pollution of this vehicle is strongly decreased and electric traction gives the possibility to achieve accurate and quick control of the distribution torque. Torque control can be ensured by the inverter, so this vehicle does not require a mechanical differential gear or gearbox. One of the main issues in the design of this vehicle (without mechanical differential) is to assume the car stability. During normal driving condition, all drive wheel system requires a symmetrical distribution of torque in the both sides. This symmetrical distribution is not sufficient when the adherence coefficient of tires is changing : the wheels have different speeds. So, a traction control system is needed. The proposed controller is implemented in terms of a hierarchical structure, which includes all new known vehicle systems ABS (Anti-lock Brake System), ASR (Anti Slip Regulation), ESP (Electronic Stability Program). The lateral dynamic of the vehicle can also been achieved by such control. There are many studies done in this direction [2,3,4].

## II. Modeling

### A. Vehicle modeling.

In order to improve vehicle comportment, a modeling of vehicle during cornering has been realized (Fig. 1). The following symbols are used : $\delta$ is vehicle steering angle, CG is center of gravity, $\nu$ the steering angle of the right front wheel, $\Omega_i$ the rotational wheel speed, $i\in[0\text{-}4]$, $\alpha$ the steering angle of the left front wheel, $r_w$ the radius of the wheel, r is vehicle yaw rate, e is wheel arm, lf, lr are distances of CG location between rear and front axes. For the description of the model, a virtual front wheel is assumed centered between the two actual front wheels at rotational wheel speed $\Omega_{c\_av}$ and a virtual wheel in CG of the vehicle at rotational speed $\Omega_{cg}$.

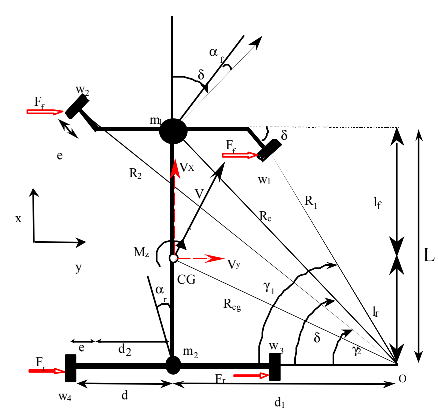


Fig 1. Vehicle modeling during right stering.

Then, speed of each wheel is given by :

$$\begin{cases} \Omega_1 = \dfrac{L + \mathrm{sign}(\delta) e \cdot \sin|\delta|}{L} \cdot \dfrac{\sin|\delta|}{\sin|\gamma|} \cdot \Omega_{c\_av} \\ \Omega_2 = \dfrac{L - sign(\delta) \cdot e \cdot \sin|\delta|}{L} \cdot \dfrac{\sin|\delta|}{\sin|\alpha|} \cdot \Omega_{c\_av} \\ \Omega_3 = \dfrac{L \cdot \cos|\delta| + sign(\delta) \cdot d \cdot \sin|\delta|}{L} \cdot \Omega_{c\_av} \\ \Omega_4 = \dfrac{L \cdot \cos|\delta| - sign(\delta) \cdot d \cdot \sin|\delta|}{L} \cdot \Omega_{c\_av} \end{cases} \quad (1)$$

and steering angles are given by :

$$\begin{cases} \alpha = sign(\delta)\cdot arctg\left(\dfrac{(L - sign(\delta)\cdot e\cdot \sin|\delta|)\cdot \sin|\delta|}{L\cdot\cos|\delta| - sign(\delta)\cdot(e\cdot\cos|\delta| + d_2)\cdot\sin|\delta|}\right) \\ \gamma = sign(\delta)\cdot arctg\left(\dfrac{(L + sign(\delta)\cdot e\cdot \sin|\delta|)\cdot \sin|\delta|}{L\cdot\cos|\delta| + sign(\delta)\cdot(e\cdot\cos|\delta| + d_2)\cdot\sin|\delta|}\right) \end{cases} \qquad (2)$$

The speed of front virtual wheel is used given by :

$$\Omega_{c\_av} = \frac{\cos(arctg(\frac{l_r}{L}\cdot tg(\delta)))}{\cos(\delta)}\cdot\Omega_{cg} \qquad (3)$$

From (1), it can be seen the wheels have different speeds value during cornering. This is a consequence of different resistant (load) torque variation $T_{ri}$ for each wheel. The drive torque $T_{di}$ given by wheel "$w_i$" $i\in[1\text{-}4]$ is :

$$T_{di} = T_{mi} - T_{ri} = (J_{wheel} + J_v)\cdot\frac{d\Omega_i}{dt} + f_r\cdot\Omega_i \qquad (4)$$

with $J_v$ ,$J_{wheel}$ vehicle and wheel inertia, $T_{mi}$ is motor torque, $f_r$ is friction coefficient, $\Omega_i$ is angular speed of wheel "i". In normal case, an uniform distribution of $T_{ri}$, is considered

$$T_{ri} = \frac{T_r}{4} \text{ and } T_r = r_w(F_a + F_c + F_r) \qquad (5)$$

with $F_a$ aerodynamic drag, $F_c$ climbing forces and $F_r$ rolling force. During cornering ($\delta$<0 for right steering) the different load repartition is expressed in function as proposed in [5].

$$\begin{cases} T_{r1} = \dfrac{T_r\cdot R_1}{4\cdot R_c} - sign(\delta)\cdot\dfrac{r_w^2\cdot d}{k_e\cdot R_c} \\ T_{r2} = \dfrac{T_r.R_2}{4\cdot R_c} + sign(\delta)\cdot\dfrac{r_w^2\cdot d}{k_e.R_c} \\ T_{r3} = \dfrac{T_r.(d_1 - d)}{4\cdot d_1} - sign(\delta)\cdot\dfrac{r_w^2\cdot d}{k_e.d_1} \\ T_{r4} = \dfrac{T_r.(d_1 + d)}{4\cdot d_1} + sign(\delta)\cdot\dfrac{r_w^2.d}{k_e.d_1} \end{cases} \qquad (6)$$

where $k_e$ [m/N] is tangential elasticity coefficient of the tire.

### B. *Motor modeling.*

A DC motor model is used [6,7]. It describes satisfactory the dynamical behavior of the wheel-motor.

## III. Traction Control

The traction control algorithm is implemented in hierarchical architecture structure as shown in Fig. 2 :

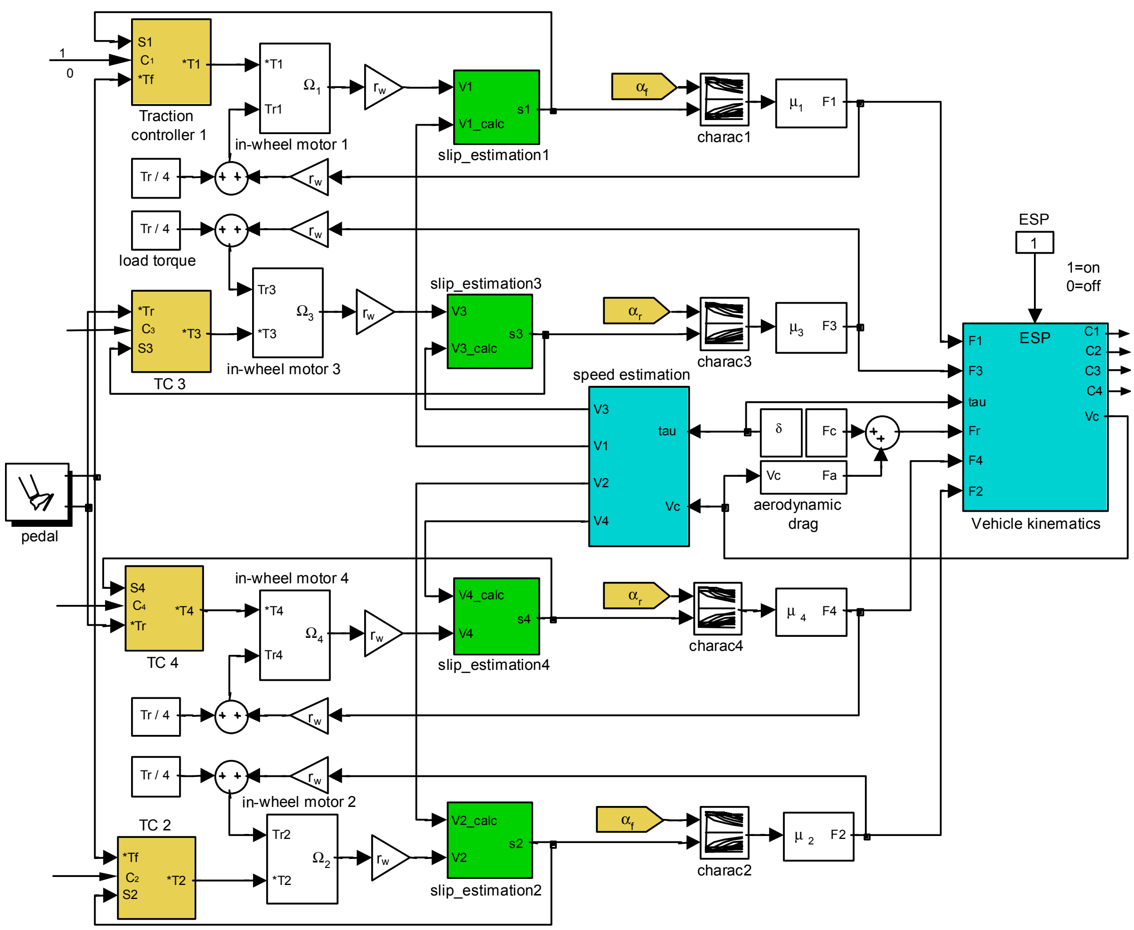


Fig 2. General configuration of traction control algorithm.

This configuration can be divided in two parts, the first integrates the control algorithm and the second, vehicle kinematics and ESP system. In order to achieve a suggestive figure, just a few internal signals are presented in the explicit implementation of the blocks.

*A. Hierarhical control program*

For implementation of control algorithm a modular structure has been realized. The control development is explicated in following paragraphs.

*1) Pedal reference*

In order to ensure the stability of the vehicle the four driving independent wheels systems requires a symmetrical distribution of torque between the wheels of the same shaft (right and left). This symmetrical distribution can be assured only if the reference torque applied to the motors is correlated with there maximal speeds ( $T_{mi} = f(\Omega_i)$ ). So to achieve good results, the pedal reference must take into account the motor performance [8]. A restriction of motor reference inside of the torque speed curve is necessary. In the case of different electric motor (EM) used in the front and rear vehicle drive shaft the motor characteristic that has the worst torque must be used as reference by pedal block. This model gives the same reference torque for the front and rear drive shaft ( $^*T_r = {}^*T_f = {}^*T_{mi}$ ).

An advanced development can be obtained by integration of energy availability in reference torque system [9].

*2) Slip control*

During normal moving condition, the vehicle stability can be assured if the wheels system require a symmetrical distribution of torque (right and left). This symmetrical distribution is not sufficient when the adherence coefficient of tires vary. So, a traction controller is integrated in order to adjust the slip coefficient for each wheel. A limitation of slip coefficient in the both sides (negative and positive) is the strategy used for implementation of ABS and ASR systems. As it is shown in [10], the value of the adherence coefficient is directly related to the wheel slip "*s*"and defined by : $s=(V_w-V)/Max(V_w,V)$, with $V_w$ is the wheel velocity and $V$, the vehicle velocity. A typical adhesion characteristic is shown in Fig. 3.

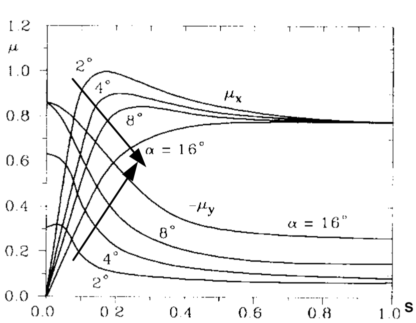


Fig. 3. Longitudinal and lateral adhesion coefficient during cornering for given values of sideslip angle α.

The best values of the adherence coefficient is obtained for $s \in [0.05..0.25]$,[3]. This can be done only if the slip coefficient "s" is correctly estimated. In case of a straight line, a limitation of a slip coefficient in the stable area (s<0.25) is sufficient. During a turn, a lateral force is exerted by the tire and the adhesion coefficient "$\mu_i$" is also influenced by the wheel sideslip angle $\alpha_f$, $\alpha_r$. By reference to Fig. 3, it is possible to see that the peak value of longitudinal characteristic decreases with increasing of sideslip angle. The measurement of sideslip angles is difficult, so they are estimated as :

$$\alpha_f = \frac{V_y + r \cdot l_f}{V_x} - \delta, \quad \alpha_r = \frac{V_y - r \cdot l_r}{V_x} \qquad (7)$$

The lateral velocity $V_y$ of the vehicle is obtained by integration of the lateral acceleration $a_y$.

$$V_y = \int_0^t a_y \, dt - r \cdot V_x \qquad (8)$$

To modify the slip wheel reference during turning, following approach is used:

$$s = s_{max} + \frac{s_{min}}{\lambda \cdot \alpha_{max}} \cdot \min(\alpha_f, \alpha_{max}) \qquad (9)$$

where $s_{max}$ =5%, $s_{min}$=4%, and λ is the characteristics gradient during cornering defined by : $\lambda = \Delta\mu/\Delta s$.

During the turn, the correct estimation of wheel slip coefficient "s" is done by equations (1--3) in speed estimation block.

*3) Traction controller*

The torque reference given by acceleration or brake pedal is limited by pedal block in order to maintain the reference between the performance system boundaries. The output reference is the torque reference applied to traction controller block. This block corrects the torque reference in order to stabilize the vehicle if a critical case is detected (skating of the wheel). This block can be divided in two parts. The first, with priority in the modification of torque reference, prevents the skating of the wheels (maintaining the wheel slip coefficient in the left side of longitudinal characteristic peak, Fig.3). The second modify the torque reference in order to correct vehicle yaw rate "r". The same block is used for each motor. With reference to (9) the maximal value of sideslip angle has been limited to 12°.

*B. Vehicle kinematics*

For vehicle moving in cruise mode on a straight and flat road with equal traction torque on the both sides, the cornering force given by (10) is zero.

$$\sum F_y = 2F_f + 2F_r = -2C_{f\alpha}\alpha_f - 2C_{r\alpha} \cdot \alpha_r = 0 \qquad (10)$$

with $C_{f}\alpha$ ,$C_{r}\alpha$ front and rear cornering stiffness and $F_f$, $F_r$, lateral forces applied of front and rear wheels. For simplicity reasons a two-degree-of-freedom model with front wheels steering is used. In turn moving (Fig. 1), the cornering force is not nil and to maintain a good trajectory a control vehicle moment is necessary. The vehicle moment can be given as :

$$\sum M_Z = 2l_f F_f - 2l_r F_r + d\left(\frac{T_{d2} + T_{d4} - T_{d1} - T_{d3}}{r_w}\right) \quad (11)$$

$\Sigma M_z = J_z . dr/dt$ with $r$ vehicle yaw rate and $J_z$ yaw inertia moment.

In this case, the cornering force, which equilibrates centrifugal force, can be expressed as

$$\sum F_y = 2F_f + 2F_r = (m_1 + m_2)(\dot{V}_y + V_x r) \qquad (12)$$

with $m_1+m_2 = m$ vehicle mass, and $V_x$, $V_y$, longitudinal and lateral velocities.

So during cornering, to assure a correct yaw moment, an ESP system is implemented. This system uses a measured yaw rate "r" and an estimated yaw rate of vehicle. In the presented model, it is obtained from differential system :

$$\begin{cases} \dot{V}_x = V_y r + \dfrac{F_1 + F_2 + F_3 + F_4}{M_v} - \alpha_f C_f \delta_f \\ \dot{V}_y = -\dfrac{C_r + C_f}{M_v} V_y + \left(\dfrac{C_r l_r - C_f l_f}{M_v} - V_x\right) r + \dfrac{C_f}{M_v} \delta_f \\ \dot{r} = \dfrac{C_r l_r - C_f l_f}{J V_x} V_y - \dfrac{C_r l_r^2 + C_f l_f^2}{J V_x} r + \dfrac{C_f l_f}{J} \delta_f + \dfrac{d}{J}(F_1 + F_3 - F_2 - F_4) \end{cases} \qquad (13)$$

To compute the yaw rate, we use the following expression $r = signe(\delta) . V_{cg}/R_{cg}$ with Vcg, $R_{cg}$ vehicle speed and radius in CG.

The driver, by changing vehicle steering angle, can correct a low yaw rate difference. For high difference the correction is realized by the implemented ESP system, which authorizes different torque references to the both sides of vehicle. The output signals $C_i$ are logic (0 or 1) integrated in Traction controller blocks. See [11] for more details.

## IV. Simulation Results

To evaluate the performances of the proposed control a combined moving cycle has been considered. It starts with an acceleration interval (Fig 4.a). In this interval, a variable adherence coefficient is assumed (Fig 4.b). So, the traction system gives different torque references to wheels, in order to improve vehicle lateral stability.

Fig. 5 shows the variation of vehicle yaw rate during supposed variable adherence between tire $w_1$ (front, right) and road. The case without ESP (Fig.5.a) and with ESP intervention (Fig.5.b) has been considered. With reference to Fig 5.a, we can see that difference between desired yaw rate and vehicle yaw rate. The variation adherence of wheel $w_1$ gives in this case an oversteer behavior.

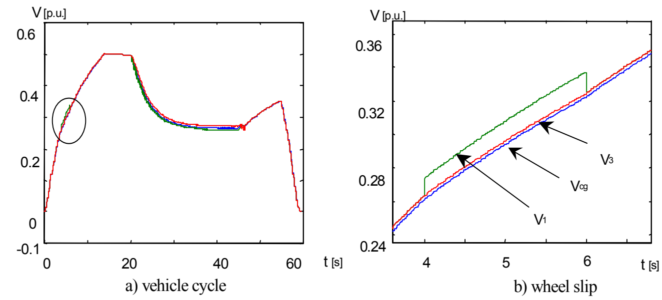


Fig. 4 - Vehicle and wheel speed during slip of the wheel $w_1$

The intervention of ESP system (Fig 5.b) improves the lateral dynamic stability of the vehicle. Low difference ( $|r\text{-}r_r| < \xi$ ) can be controlled changing vehicle steering angle.

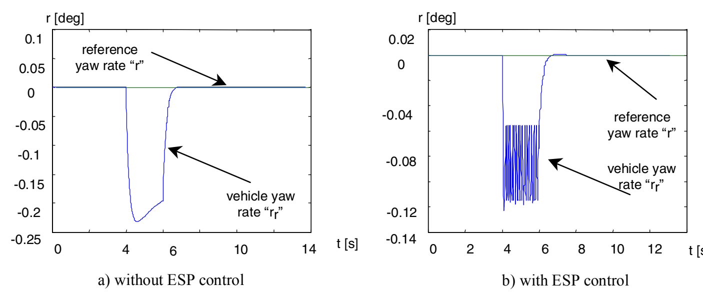


Fig. 5 - Vehicle yaw rate during a wheel sliping

Consequently, the ESP system intervenes only if a high difference ($|r\text{-}r_r| > \xi$) is detected. The correction of vehicle yaw moment is not possible during skating of the wheel, the slip regulation must have priority. Fig.6 shows the regulation of wheel slip in order to maintain the slip coefficient "$s_1$" in the stable area. We can see that the slip value is maintained inferior to 5%.

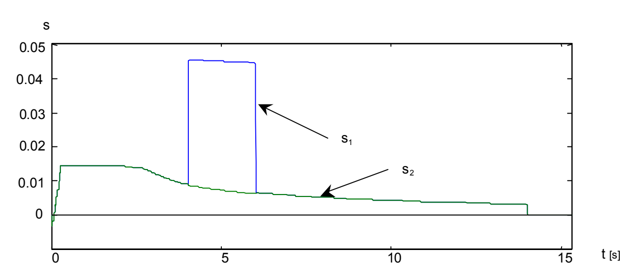


Fig. 6 - Slip regulation during variable adherence for $w_1$.

The cycle continues with vehicle moving in cruise mode and beginning a right turn ( Fig 7.a, t ∈ [20- 45]s). In this case the wheels of vehicle do not follow the same trajectory and so the wheels have different speeds.

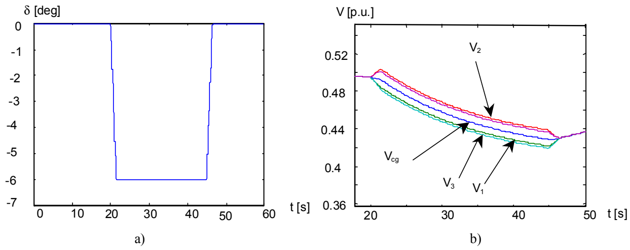


Fig. 7 - Steering angle and wheel speed during cornering.

Let's consider a different mass repartition between front and rear shaft ( $m_1>m_2$). Owing to superior mass in the front shaft, during a turn the vehicle has understeer behaviour ( Fig.8.a). In this case the ESP system brakes the rear inside wheel of curve, to stabilize vehicle trajectory (Fig.8.b).

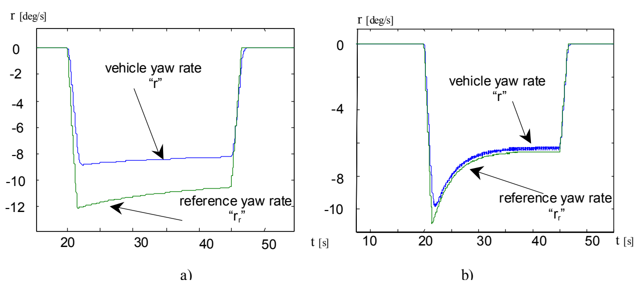


Fig. 8 -Vehicle yaw rate during right stering and critical case.

The variation of motor current imposed by ESP in order to correct vehicle understeer behavior is shown in Fig.9.

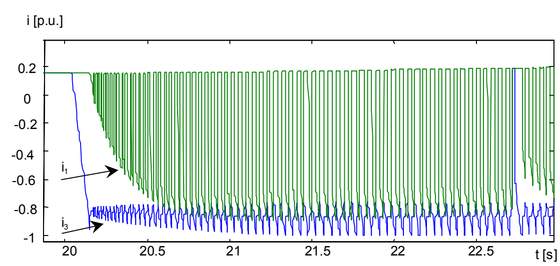


Fig. 9 - Motors current of wheels $w_3$ and $w_1$ during ESP control.

It is possible to remark that the wheel $w_3$ is braked but as consequence of the low adhesion coefficient (a snowing road surface for example) this torque is not sufficient to correct vehicle yaw motion. In this case a complementary braking torque is given by wheel $w_1$ (in the same shaft).
After the return of the vehicle in cruise mode the simulation cycle is finished by a regenerative braking (Fig 10.a).

We can see that the regenerative torque (current) is limited by motor performances. If the energy management is integrated, this regenerative torque must be also correlated with the maximal recharge current accepted by the battery.

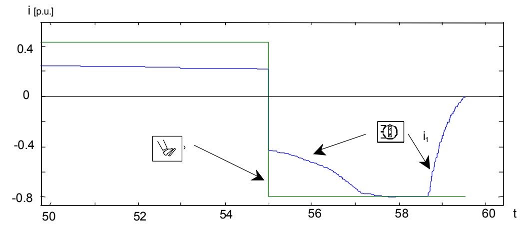


Fig. 10 - Pedal reference and motor current during regenerative braking

## V. Conclusion

In this paper a modeling and simulation of a new traction control algorithm for an electrical vehicle with four separate wheel drives has been proposed. Using the most advantage of electric traction (quick and precise torque control) this algorithm which incorporates all new known vehicle systems ABS, ASR, ESP is our solution of new constraints imposed in vehicle design : high safety and non polluting personal vehicle. The proposed control is necessary however if the road conditions vary to guarantee the lateral dynamic stability. During traction and regenerative braking, a correlation of traction control with motor performances has been realized. In the turn for providing a good traction control, this algorithm must take into account different wheel speeds. In this paper a modular and hierarchical implementation is presented. The results obtained by simulation with this model are satisfactory but in fact, control performances depend on the quality of the reference vehicle velocity. As a solution, a fuzzy estimation speed is proposed [10].

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