ELECTRIC VEHICLE WITH TWO INDEPENDENT WHEEL DRIVES - IMPROVING THE PERFORMANCE WITH A TRACTION CONTROL SYSTEM

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ABSTRACT

This paper presents a detailed dynamic model of an electric vehicle with two independent wheel drives and an improved traction control system. Using electric motors it is possible to have a torque control in each wheel drive, that enables the implementation of a traction control. This control level improves the stability and the safety of the vehicle. Analysis, design and simulation results of this system will be presented.

1 - INTRODUCTION

One of the main problems related with our society evolution are the impacts in the population quality of life, in consequence of traffic evolution and related environmental impacts. This situation imposes the search of alternative solutions to the use of internal combustion engine vehicles (ICV). In this perspective, electric vehicles (EV) are an interesting alternative to solve the environmental and energy consuming problems.

However, the advantages of electric road vehicles are not limited to environmental impact benefits. Electric vehicles can be more interesting if we increased the most remarkable advantage of EV’s: when compared with internal combustion engines, electric motors have the capability to control the generated torque with a better and precise dynamic performance. This major aptitude of electric drives can be applied in the control of the effective traction force applied between tire and road surface. This could result in improvements on the vehicle stability and safety conditions, when compared with traditional cars, allowing a better performance in limit conditions. From this point of view, a traction control system (usual only at top level cars) could be introduced more often without the necessity of turn to a significant “hardware”, but only with some appropriate “software” and, without an increased price.

In this paper it is introduced a project for developing an electric vehicle with a two independent front wheels propulsion system and some contributions and preliminary results of a traction control system by reference model are presented. As the torque in each wheel is controlled and knowing the vehicle dynamics, the traction control system will allow a maximum torque applied to the road surface independently from the adhesion conditions, inhibiting wheel-spin and assuring the steerability and stability of the vehicle.
2 – Vehicle Structure Whit Two Independent Front Wheel Drives

The vehicle considered in the analysis and target for the implementation of the proposed traction control system is a Renault Clio (fig. 1). Starting from an usual vehicle structure, some adaptations are in course with the objective of introducing a two independent front wheels propulsion system using electric drives.

Fig. 1 – Experimental electric vehicle with two independent wheel drives.

In fig. 2 is showed the implemented system (electric and mechanical components) in the experimental vehicle, with a control traction system of independent drive wheels.

Fig. 2 – Propulsion and control system of the Renault Clio.

Each wheel is coupled to a 8,5kW permanent magnet DC machine. A compact power electronics converter with two independent 4Q choppers, handling each one with 48V, 200A and a maximum switching frequency of 33 kHz, was developed in the team.

In order to reduce the switching frequency a 4Q chopper with 3 level output voltage was developed. Figure 3 shows some simulation results of the proposed methodology, where a sliding-mode current control was assumed.

Fig. 3 – Voltage and current output of 3 level chopper - simulation results.

a) 3 level output voltage

b) sliding-mode current control
Like the expected from simulations, experimental results were obtained (figure 4).

From the system analysis and experimental results, the implemented electric drive will assure the effective (precise and fast performance) torque control, needed for any traction control system to be realised.

3 – VEHICLE DYNAMIC ANALYSIS

3.1 - Resistant Forces

Equation (1) presents the road load $F_{RT}$ including all the resistant forces opposing to the vehicle motion,

$$F_{RT} = F_{RR} + F_{ST} + F_{DA} + F_I$$

where $F_{RR}$ is the rolling resistance, $F_{ST}$ is a Stokes force, $F_{DA}$ is the aerodynamic drag force, $F_I$ is a climbing resistance.

The rolling resistance is obtained by (2), where $\mu$ is the rolling resistance coefficient (caused by the tire deformation and contact with the road), $m$ is the vehicle mass and $g$ the gravitational acceleration constant.

$$F_{RR} = \mu m g$$

$F_{ST}$ is the Stokes force or viscous friction, given by (3), where $k_A$ is the Stokes coefficient and $v$ is the speed of the vehicle.

$$F_{ST} = k_A v$$

The resistance of the air acting upon the vehicle is the aerodynamic drag, which is given by (4), where $\rho$ is the air density, $C_D$ is the aerodynamic drag coefficient, $A_f$ is the vehicle frontal area and $v$ is the vehicle speed [2],[3].

$$F_{DA} = \frac{1}{2} \rho C_D A_f v^2$$
The climbing resistance ($F_I$ is positive) or the downgrade force ($F_I$ is negative) is given by (5).

$$F_I = P \sin \Theta = m \ g \ \sin \Theta$$  \hspace{1cm} (5)

### 3.2 - Wheel Dynamics

The mechanical equation (in the motor referential) used to describe each wheel drive is expressed by (6).

$$J_m \frac{d\omega_m}{dt} = T_m - T_r$$  \hspace{1cm} (6)

In this equation, $\omega_m$ is the angular motor speed and $T_m$ the produced motor torque. Due to the use of a reduction gear, with transmission ratio, $i$, relations presented by equations (7) and (8) can be defined. In equation (8) $\eta_i$ is the transmission efficiency.

$$\omega_{wheel} = \frac{\omega_m}{i}$$  \hspace{1cm} (7)

$$T_{wheel} = T_m \ i \ \eta_i$$  \hspace{1cm} (8)

The load torque at the motor referential is defined by (9), where $R$ is the tire radius.

$$T_r = \frac{T_{wheel}}{i} = \frac{R}{i} \ F_{gy}$$  \hspace{1cm} (9)

The global moment of inertia of the vehicle from the motor referential ($J_m$), can be defined as a sum of shaft inertia moment including the motor and wheel inertia ($J_{wheel}$) and the factor corresponding to the vehicle mass ($J_V$) (equation (10)).

$$J_m = J_{wheel} + J_V$$  \hspace{1cm} (10)

The shaft inertia moment $J_V$ is defined by (11).

$$J_V = \frac{1}{2} m \ \frac{R^2}{i^2} (1 - \lambda)$$ \hspace{1cm} (11)

If the adhesion coefficient of the road surface is high, then the slip $\lambda$ (i.e. the relative differences between wheel and vehicle speeds) is usually low, and can be neglected. However this coefficient must be considered in any traction control system, as it will be shown in 3.4 and defined by equation (18)[1].

### 3.3 - Longitudinal and Lateral Vehicle Motion Equations

In literature [8] it is possible to obtain several descriptions and models for the dynamics of lateral and longitudinal vehicle motion. These dynamics are described on the figure 5. This model uses chassis-based coordinates and the variables are depicted in the figure.
The differential equations of motion from Newton’s second law are:

\[ F_x = m a_x \]  \hspace{2cm} (12)

\[ F_y = m a_y \]  \hspace{2cm} (13)

\[ N = I_z \frac{d \dot{\gamma}}{dt} = I_z \dot{\gamma} \]  \hspace{2cm} (14)

where \( F_x \) and \( F_y \) are the longitudinal force and lateral force, \( N \) is the resultant yawing moment that the tires apply to the vehicle, \( I_z \) is the moment of inertia about the Z-axis, \( \dot{\gamma} \) is the angular acceleration, \( a_y \) the lateral acceleration and \( m \) is the mass of vehicle.

The dynamic model of the vehicle related to its referential gravity centre is:

\[ m (\dot{V}_x - V_y \dot{r}) = F_{x1} \cos \delta + F_{x2} \cos \delta - F_{y1} \sin \delta - F_{y2} \sin \delta \]  \hspace{2cm} (15)

\[ m (\dot{V}_y + V_x \dot{r}) = F_{y1} \cos \delta + F_{y2} \cos \delta + F_{y3} + F_{y4} + F_{x1} \sin \delta + F_{x2} \sin \delta \]  \hspace{2cm} (16)

\[ I_z \dot{\gamma} = l_1 (F_{y1} + F_{y2}) \cos \delta - l_2 (F_{y3} + F_{y4}) + l_1 (F_{x1} + F_{x2}) \sin \delta + \frac{d}{2} (F_{x1} - F_{x2}) \cos \delta + \]  \hspace{2cm} (17)

\[ + \frac{d}{2} (-F_{x1} + F_{x2}) \]

The global model of the vehicle dynamics is presented on fig. 6.
3.4 – Tire Model

In order to make traction control it is necessary to know the speed of each drive wheel ($v_W$) and also the real speed of the vehicle on both sides ($v_v$). Considering those speeds it is possible to calculate the slip (i.e. the relative speed differences, as defined by equation (18) [4].

$$\lambda = \frac{v_W - v_v}{\max\{v_W, v_v\}} = \frac{\omega R - v_v}{\max\{v_W, v_v\}}$$  \hspace{1cm} (18)

The slip value depends on the generated motor torque and also on the road conditions. The adhesion coefficient or friction coefficient is defined by equation (19) and is illustrated in figure 7. In equation (19), $F_d$, is the force longitudinal that each wheel drive can transmit to the road surface.

$$\mu = \frac{F_d}{mg}$$  \hspace{1cm} (19)

3.4.1 – Longitudinal Force in the Tire

In the simulation program the friction coefficient is a function of the slip $\lambda$, approximated by equation (20).

$$\mu = C_1 (1 - e^{-C_2 \lambda}) - C_3 \lambda$$  \hspace{1cm} (20)

![Fig. 7 – Friction coefficient as a function of the slip.](image-url)
The component along the longitudinal axis of the absolute acceleration in the gravity centre has two components: one according to direction X and another according to direction Y [8] (equations (21) and (22)).

\[
a_x = (\dot{V}_x - V_x r)
\]

\[
a_y = (\dot{V}_y + V_x r) = V (r + \dot{\beta})
\]

3.4.2 – Lateral Force in the Tire

In agreement with fig. 5, the side slip angle is given by (23).

\[
\dot{\beta} = \tan^{-1} \frac{V_x}{V_y}
\]

The slip angle \(\alpha\) is the angle formed between the direction of wheel travel and the line of the intersection of the wheel plane with the road surface. For a small side-slip angle \(\alpha_i\) equations (24) and (25) can be achieved.

\[
\alpha_i = \frac{V_y - l_2 r}{V} = \beta - \frac{l_2 r}{V}
\]

\[
\alpha_f = \frac{V_y + l_1 r}{V} = \beta + \frac{l_1 r}{V} - \delta
\]

The lateral forces acting on the front and rear tires are a function of the corresponding slip angle and the cornering stiffness, using the linear tire model are expressed by (26) and (27), as showed in the fig. 8.

\[
F_{yfi} = C_{a_f} \alpha_f
\]

\[
F_{yri} = C_{a_r} \alpha_r
\]

The cornering stiffness in the tire is defined by equation (28).

\[
C_a = \frac{\partial F_y}{\partial \alpha} \bigg|_{\alpha = 0}
\]

Typical lateral force characteristic in the tire as function of vehicle speed is show in fig. 8.
3.4.3 – “Magic formula tire Model”

The tire model describes the forces and moments generated at the interface between the vehicle and road. Extensive studies and experiments on pneumatic tire models have both led to empirical models. A “Magic formula tire Model” proposed by [7],[8] (Bakker, Nyborg and Pacejka, 1987) is used to capture the nonlinearities of tires.

The tire longitudinal and lateral forces for the tire i are calculated from:

$$F_x = D_x \sin(C_x \tan^{-1}(B_x \Phi_x)) + S_{vx}$$  \hspace{1cm} (29)

$$F_y = D_y \sin(C_y \tan^{-1}(B_y \Phi_y)) + S_{vy}$$  \hspace{1cm} (30)

Where:

$$\Phi_x = (1-E_x)(\lambda_S + S_{hx}) + \frac{E_x}{B_x} \tan^{-1}(B_x(\alpha + S_{hx}))$$  \hspace{1cm} (31)

$$\Phi_y = (1-E_y)(\lambda_S + S_{hy}) + \frac{E_y}{B_y} \tan^{-1}(B_y(\alpha + S_{hy}))$$  \hspace{1cm} (32)

In equations (29) to (32) $B_x, C_x, D_x, E_x, B_y, C_y, D_y, E_y, S_{vx}, S_{vy}$ are constants depending of the tire normal forces [8].

4 - Traction Control Algorithm

4.1 - First approach

In [1] a first approach to a traction controller it was proposed. Fig. 9 presents the vehicle model and the suggested algorithm that control the motor torque at each wheel, in order to maintain the steerability and stability of the vehicle.
The command torque $T_{ref}$ is proportional to the acceleration pedal angle. The traction controller is implemented at the central block of the figure, having as inputs the wheel speeds ($v_W$), the real speed of the vehicle on both sides and the steering angle $\delta$. The outputs of the traction controller are $T_{cR}$ and $T_{cL}$ that will change the torque reference in each motor ($T_{refR}$ and $T_{refL}$). This control algorithm compensates the difference between the forces that each wheel applies to the road surface and so the real speed of the left and right side of the vehicle. In the controller implementation the velocity can be measured directly in a wheel without traction function or can be estimated. In the simulation program it was considered that the wheel speeds were measured.

Simulation results were developed in order to test the effectiveness of this first approach to the traction control algorithm. Figures 10 and 11 presents one of the situation tested.

Assuming equal road adhesion conditions for each wheel, the control system impose equal values for the generated torque on the left and right wheel drives. Consequently, the right and left side wheels rolls at the same speed and vehicle trajectory is only in x axis direction.

However, if the road conditions under the left and right side wheels are different, for the same generated torque in the left and right side motors different slip occurs in each wheel. In this situation, the traction forces applied to the road surface under the left and right side wheels are different. Consequently, the vehicle describes a curve, even if the steering angle is maintained at zero (fig. 10).

To correct the situation simulated, on fig. 11 the control traction algorithm is activated. When different vehicle speeds on left and right side are detected, the control algorithm changes the reference torque for each motor, in order to produce the same traction force on both sides and limiting the slip to the stable region (fig. 11).
As the road conditions under the left and right side wheels are different, for the same traction force correspond different slips on both sides, as represented on fig. 11a).

The results presented where obtained from a basic control approach based in a PID controller related to the error between the space values travelled by each wheel, (fig. 11b)).

Figure 11b) shows the effectiveness of the control to assure the return of the vehicle motion to a trajectory under the \( x \)-axis direction, but it is also verified that a static error in \( y \)-axis position occurred.

From an exhaustive analysis of this and other simulation results it was concluded the need of developing an alternative and more global approach to the traction algorithm control.

### 4.2 Global approach

A more global approach to the traction algorithm control was assumed. Various controllers philosophy can be found proposed in literature to be applied to vehicles traction control [6].

Based in the vehicle dynamics model presented in fig. 6, it is possible to develop a control algorithm supported in a reference model approach, fig. 12.

Fig. 12 – Global approach to the traction control algorithm

Considering the output from the vehicle dynamic model as the desired (expected) vehicle performance, the comparison with the real vehicle performance gives the input to the traction control algorithm, deciding what are the correct action to be developed in order to assure a correct vehicle control.

When a difference in performance is detected, the controller should act on the torque (current) that each motor imposes on the road surface. Also, an on-line analysis about the
reason that justifies this difference must be developed, imposing an adaptation of the reference model to the road real conditions.

## 5 - Conclusions

In this paper is introduced a project for developing an electric vehicle with a two independent front wheels propulsion system and it is presented contributions and preliminary results of a traction control system by reference model.

In order to assure an effective control of torque developed by each wheel and the force applied to road surface, a special attention has been given to the electric drive development. A power electronics converter with a sliding-mode current control was implemented and simulation and experimental results were presented.

Traction control systems imposes a very precise knowledge of the vehicle dynamics. A vehicle dynamics model is exhaustively detailed.

When the torque in each wheel is controlled and knowing the vehicle dynamics, the traction control system will allow a maximum torque applied to the road surface independently from the adhesion conditions, inhibiting wheel-spin and assuring the steerability and stability of the vehicle.

Results from a first approach and the basis for a more global traction control algorithm in development are presented.

## References:


