ANALYSIS AND DESIGN OF A TRACTION CONTROL ALGORITHM FOR AN ELECTRIC KART WITH TWO INDEPENDENT WHEEL DRIVES

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Abstract: This paper presents a detailed dynamic model of an electric vehicle with two independent wheel drives and a 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 algorithm. 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

The consequences of traffic evolution and related environmental impacts in the population quality of life impose the search of alternative solutions to the use of internal combustion engine vehicles (ICV). Electric vehicles (EV) are an interesting alternative.

Electric road vehicles are not a new subject in the research & development field. Nowadays, the major companies of carmakers propose battery based electric vehicles with enough performance even in practical use and the major limitation is related to the battery storage capability. Alternative solutions using hybrid technologies and fuel cells are in development or already in the market.

However, advantages of electric road vehicles are not limited to environmental impact benefits. The attractiveness of electric vehicles can be increased if the most remarkable advantage of EV's is applied: electric motors can control the generated torque with a better and precise dynamic performance, when compared with internal combustion engines. This major capability of electric drives can be applied in the control of the effective traction force applied between tire and road surface. The vehicle stability and safety can be improved, allowing a better performance in limit conditions when compared with traditional cars.

Some expensive models of ICV's present traction control systems combined and/or antilock (or ABS or antiskid) systems. Traction control systems are designed to prevent spinning of drive wheels when an excessive throttle is applied.

Meanwhile, antilock systems are included in order to assure that wheels do not lock in braking actions. When it is detected that a wheel will lock, this system reduces the brake pressure, allowing the wheel speed return to the slip level range necessary for near-optimum braking performance. These systems are fundamental in order to improve the vehicle steerability and stability and to reduce stopping distances. However, they are expensive and bulky and, sometimes, the performance is not the expected.

These systems objectives can be implemented in EV's in a much more easier and adapted way. The natural ability of electric drives to control the generated torque and the introduction of an independent control of the traction wheel drives (two or four) can allow an high-performance traction control with a low cost, quick response and easy to design implementation. A vehicle topology like the proposed one allows a simplified mechanical structure of the vehicle and an effective traction control will allow to reduce the energy consumption, namely by diminishing energy losses from the friction between the tires and the road surface during sliding, improving the tires lifetime.

In another paper [1] we show that with two wheel drives and two independent motors it is possible to eliminate the mechanic differential and implement a electric differential.

Now, we will assume that the adhesion coefficients on the left and right side of the vehicle could be different. So, it is necessary to read the wheels speed and the real speed of the vehicle in both sides, in order to analyse the possible sliding of the vehicle. With the acquisition of these values, the torque in each wheel could be controlled, making possible the steerability and stability of the vehicle. This is the task of the traction control algorithm to be introduced.

2 - KART STRUCTURE WITH TWO INDEPENDENTE WHELL DRIVES

The proposed traction control algorithm will be applied to a kart with two independent wheel drives (rear axle) that has been projected and constructed in our group during the last year (figure 2).

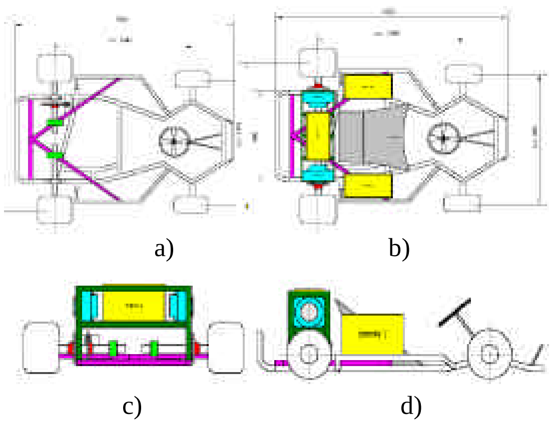


Fig. 1 – Schematics of the developed kart. a) Structure strengthen and rear axle with two independent wheels. b), c), d) Perspectives from the vehicle final design.

Fig. 2: Electrical kart with two independent wheels drives

Each wheel is coupled to a 10kW permanent magnet DC machine. A compact power electronics converter with two independent 4Q choppers, handling each one with 36V, 200A and a maximum switching frequency of 33 kHz, was developed in the team (for 36V, the motor power output is only 5.5kW).

3 - DYNAMIC ANALYSIS

3.1 – Forces Applied to the Vehicle Structure

In the dynamic analysis of the vehicle behaviour, we consider the front wheels are free. This is the worst situation to be considered for the traction control algorithm. At the beginning it is assumed that at each drive wheel is applied an equal influence from the vehicle mass and from the external forces. If the DC motors have the same armature current, the produced torques are the same and also the applied forces to the ground. In this situation the left and right wheels rolls at the same speed and the vehicle has a straight trajectory.

However, if the force that the right wheel (W_R) applies to the ground is different from the force applied by the left wheel (W_L), the vehicle trajectory describes a curve, like the presented at figure 3.

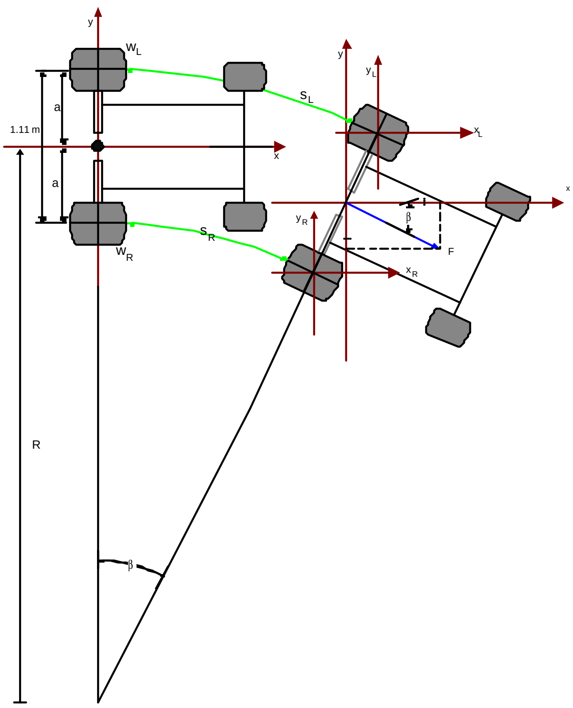


Fig. 3 – Vehicle trajectory with different forces applied on left and right sides.

In figure 3, is considered that the applied force to the road surface generated from the left wheel drive is higher than the applied force from the right wheel drive, and the vehicle turn right. The vehicle trajectory describes a curvature with a radius R, making an angle β(t) with the y-axis.

$$\begin{cases} b*(R-a)=S_R \\ b*(R+a)=S_L \end{cases} \begin{cases} R=\dfrac{S_R}{b}+a \\ b=\dfrac{S_L-S_R}{2a} \end{cases} \tag{1}$$

Note that this situation can be the result of a variation in the steering angle command, but also could appear as the result of different road conditions under the left and right side wheels.

To understand the dynamic behaviour of the vehicle, two consecutive time intervals are analysed: the first one (before t_0) and the second one (after t_0).

<u>First time interval</u> (before t_0): in this time interval the applied forces from each wheel drive are the same ($F_L=F_R$).

The resultant force applied to the vehicle is in the x-direction, with value F_X=F and F_y=0

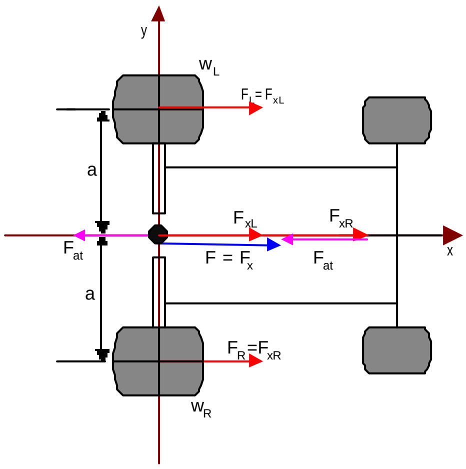


Fig. 4 – Vehicle trajectory before t_0.

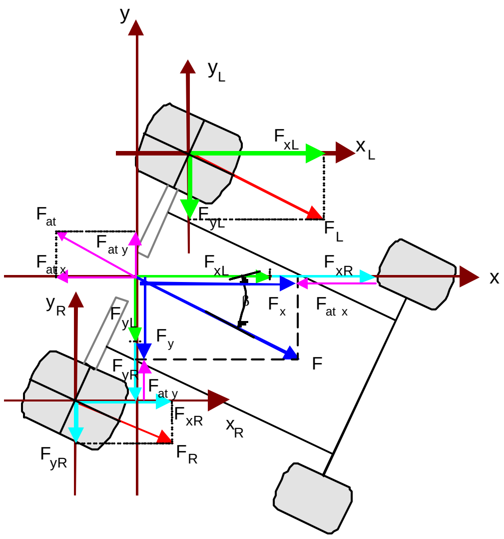


Fig.5–Vehicle trajectory in the second time interval (after t_0).

<u>Second time interval</u> (after t_0): in this situation the applied forces at the road surface are different at the left and right sides ($F_L \neq F_R$), resulting a total force with components in x and y directions (figure 5). In this time interval the left wheel, W_L, travel a distance S_L and the right wheel, W_R, travel a distance S_R.

Total Load Force, F_{at}: This force it is the sum of the rolling resistance, Stokes force or viscous friction, aerodynamic drag and climbing resistance (2).

$$F_{at} = F_{ae} + F_{ad} + F_{a\,ad} + F_i \qquad (2)$$

In equation (2), the rolling resistance, F_{ae} , is obtained by (3), where m 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.

$$F_{ae} = m M \ g \qquad (3)$$

F_{ad} is the Stokes force or viscous friction, given by (4), where k_A is the Stokes coefficient and v is the speed of the vehicle.

$$F_{ad} = k_A \ v \qquad (4)$$

The resistance of the air acting upon the vehicle is the aerodynamic drag, which is given by (5), [2], [3].

$$F_{a\,ad} = ½ * \boldsymbol{r} * C_w * A.v^2 = k_D.v^2 \qquad (5)$$

The climbing resistance (F_i is positive) or the downgrade force (F_i is negative) is given by (6).

$$F_i = P.\sin \boldsymbol{a} = Mg.\sin \boldsymbol{a} \qquad (6)$$

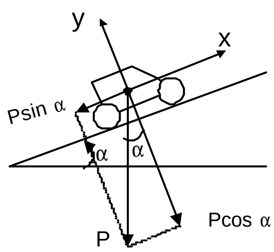


Fig. 6 – Vehicle submitted to the action of the climbing force.

3.2 - Speed of the Vehicle

If the left side speed of the vehicle (v_L) is equal to the right side speed (v_R), then the vehicle speed (v_V) can be considered equal to those speeds. However, if the vehicle is describing a circle, then the left and right side speeds are different ($v_L \neq v_R$). Therefore, the vehicle speed can be considered the average of both speeds (equation (7)).

$$\vec{v}_V = \frac{\vec{v}_L + \vec{v}_R}{2} \qquad (7)$$

To calculate the left and right side vehicle speeds, it is considered that the vehicle mass (M) is equally distributed on both sides and also the total resistant force. Considering (8), the right side speed of the vehicle is expressed by (9), where a_R is the respective acceleration.

$$\vec{F}_R - \frac{\vec{F}_{at}}{2} = \frac{M}{2} * \vec{a}_R \Rightarrow \vec{a}_R = \frac{2}{M} * (\vec{F}_R - \frac{\vec{F}_{at}}{2}) \qquad (8)$$

$$\vec{v}_R = \int_0^t \vec{a}_R dt \qquad (9)$$

Similarly for the left side:

$$\vec{F}_L - \frac{\vec{F}_{at}}{2} = \frac{M}{2} * \vec{a}_L \Rightarrow \vec{a}_L = \frac{2}{M} * (\vec{F}_L - \frac{\vec{F}_{at}}{2}) \quad (10)$$

$$\vec{v}_L = \int_0^t \vec{a}_L dt \quad (11)$$

3.3 - Wheel Dynamics

The mechanical equation (in the motor referential) used to describe each wheel drive is expressed by (12).

$$J_m \frac{d\omega_m}{dt} = T_m - T_r \quad (12)$$

In this equation, ω_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 (13) and (14) can be defined. In equation (14) η_t is the transmission efficiency.

$$\omega_{wheel} = \frac{\omega_m}{i} \quad (13)$$

$$T_{wheel} = T_m \, i \, \eta_t \quad (14)$$

The load torque at the motor referential is defined by (15), where r is the tire radius.

$$T_r = \frac{T_{r\,wheel}}{i} = \frac{r}{i} F_{at} \quad (15)$$

The global moment of inertia of the vehicle from the motor referential (J_m), can be defined as a sum of shaft inertia moment (J_{wheel}) and the factor corresponding to the vehicle mass (J_V) (equation (16)).

$$J_m = J_{wheel} + J_V \quad (16)$$

The shaft inertia moment (J_{wheel}) is defined by (17), and J_V is defined by (18).

$$J_{wheel} = \frac{1}{2} \frac{r^2}{i^2} M_{wheel} \quad (17)$$

$$J_V = \frac{1}{2} M \frac{r^2}{i^2} (1 - slip) \quad (18)$$

The function *slip* in equation (18) will be defined in section 4 (equation (22)). If the adhesion coefficient of the road surface is high then the slip is usually low. If this coefficient can be neglected, the contribution of the vehicle mass to the global inertia moment can be calculated as follow:

$$\Delta W_c = 0 \Leftrightarrow W_{c_{in}} = W_{c_{out}} \quad (19)$$

$$2 \times \frac{1}{2} J_V \omega_m^2 = \frac{1}{2} M v^2 \quad (20)$$

$$J_V = \frac{1}{2} \frac{r^2}{i^2} M \quad (21)$$

4 - TRACTION CONTROL

The capability of a better dynamic performance that electric motors have in controlling the developed torque, when compared with the internal combustion engines, enable us to implement a traction control algorithm [5], [6].

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 (22) [2].

$$slip = \left| \frac{v_W - v_V}{\max\{v_W, v_v\}} \right| \quad (22)$$

The slip value depends of the generated motor torque and also of the road conditions. The adhesion coefficient or friction coefficient is defined by equation (23) and is illustrated in figure 7. In equation (23), F_d, is the force that each wheel drive can transmit to the road surface.

$$\mu = \frac{F_d}{\frac{M}{2} g} \quad (23)$$

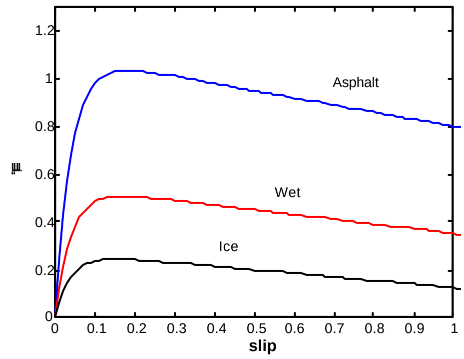


Fig. 7 – Friction coefficient as a function of the slip

In order to give physical meaning to the function coefficient, μ, it is necessary to guarantee the condition expressed by equation (24).

$$m = \begin{cases} m & if \quad v_w > v_V \quad and \quad v_V > 0 \\ m & if \quad v_w < v_V \quad and \quad v_V < 0 \\ -m & if \quad v_w < v_V \quad and \quad v_V > 0 \\ -m & if \quad v_w > v_V \quad and \quad v_V < 0 \end{cases} \qquad (24)$$

In the simulation program the friction coefficient as a function of slip is approximated by equation (25).

$$\mu = C_1(1 - e^{-C_2 * slip}) - C_3 * slip \qquad (25)$$

It is important to verify that if the slip is high, each wheel drive "see" the mass of the vehicle with a lower value, as is expressed by equation (18). In this situation, an increase on the generated motor torque can contribute in the wrong way, increasing the slip and decreasing the traction force applied to the road surface.

Figure 8 presents the vehicle model used in the analysis and the design of the proposed traction control algorithm.

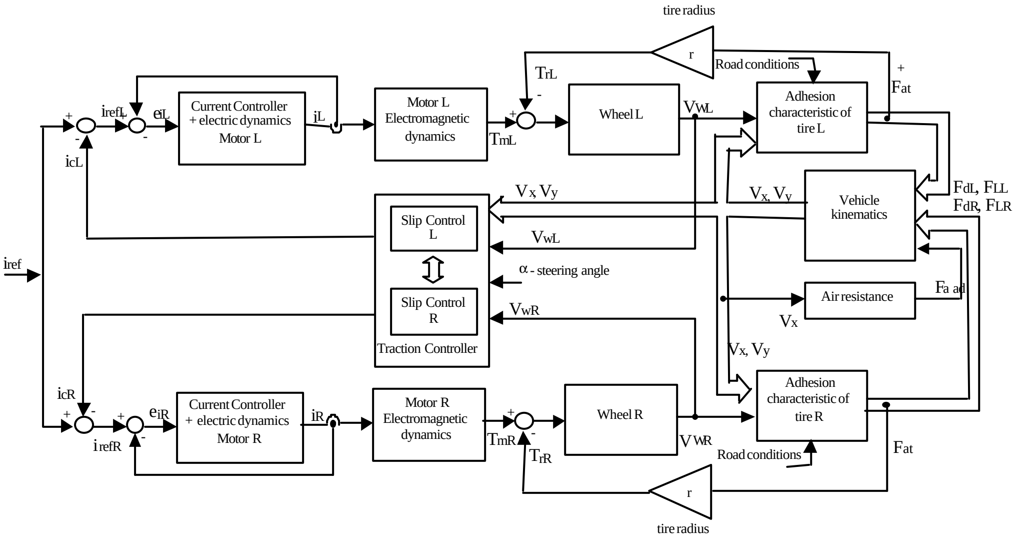


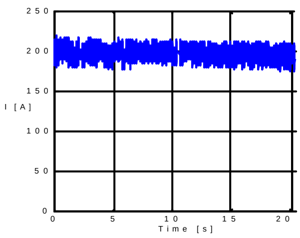
Fig.8 – Global model system, with traction control algorithm

The command current i_{ref} is proportional to the acceleration pedal angle. The traction controller is implemented at the central block of the figure, having as inputs the wheels speeds (v_W), the real speed of the vehicle on both sides and the steering angle α. The outputs of the traction controller are i_{cR} and i_{cL} that will change the reference current in each motor (i_{refR} and i_{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 this speed was measured.

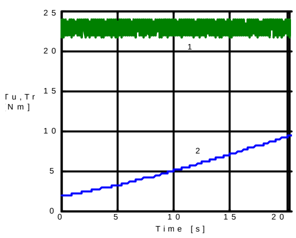
The adhesion characteristics of each tire, considering road conditions and the linear speed of wheels and vehicle will define the driving forces, F_{dL} and F_{dR}, and the side forces, F_{LL} and F_{LR}, effectively applied in the contact between tires and road surface.

5.-SIMULATION RESULTS

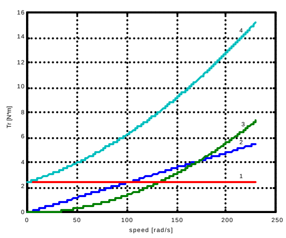
In the first simulated results (figure 9), is assumed that the generated torque on the left and right wheel drives are equal and also the load torque applied to each motor. Consequently, the right and left side wheels rolls at the same speed and the vehicle has a straight trajectory.



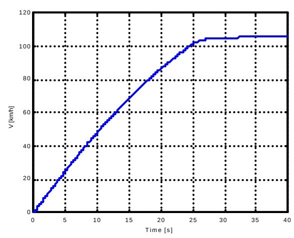
a) Motor current



b) 1-Motor torque 2-Load torque



c) 1- Rolling friction; 2-Stoks friction; 3-Aerodynamic friction; 4-Total torque resistor

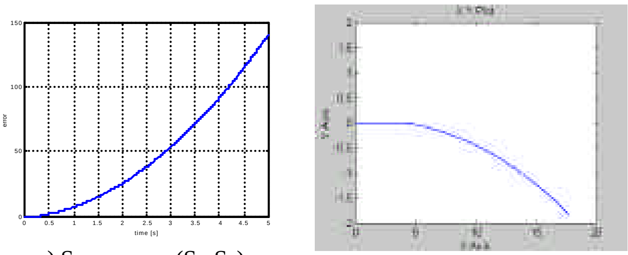


d) Motor speed

Fig.9 – Simulation results for a straight trajectory (same road conditions under the left and right side wheel drives)

In figures 9-a) and 9-b) it is visible the current and motor torque oscillations, due to the strategy used to control these variables (sliding mode control). Figure 9-c) shows the different contributions to the total load torque, and on figure 9-d) the resultant vehicle speed is visualised.

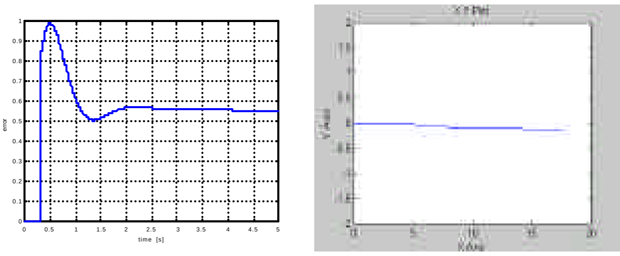
However, if the road conditions under the left and right side wheels are different, for the same generated torque in the left and right side motors different slip occurs in each side wheels. In this situation, the traction forces applied to the road surface under the left and right side wheels are different.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



a) Space error (S_L-S_R) b) Vehicle trajectory

Fig10 – Simulation results without traction control and when the road under the right wheel changes from asphalt to ice

To correct the situation simulated on figure 10 the control traction algorithm is activated. When different vehicle speeds on left and right side is detected, the control algorithm changes the reference currents for each motor, in order to produce the same traction force on both sides and limiting the slip to the stable region (figure 11).



a) Space error (S_L-S_R) b) Vehicle trajectory

Fig11 – Simulation results with traction control and when the road under the right wheel changes from asphalt to ice

It is important to verify that if 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 figure 12.

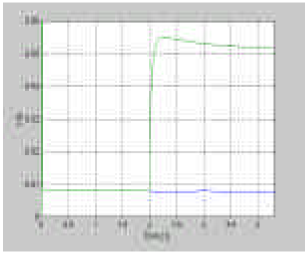


Fig12 – Slip variation when the road under the right wheel changes from asphalt to ice

6.- CONCLUSIONS

This paper presented an electric vehicle already in development with two independent rear wheel drives. An electrical differential was implemented assuring that, in straight right trajectory, the two rear wheels roll exactly at same speed (for a steering angle α=0) and for the same road conditions under the left and right side wheels. The system analysis, modelling and simulation is presented.

However, if the road conditions changes, a simple torque motor control is not sufficient. In this situation it is necessary, to implement a traction control algorithm, without any substantial extra hardware incorporation. With the information about the real speed of the vehicle, on left and right sides, the control algorithm changes the generated torques in each wheel drive, in order to produce the same traction force applied to the road surface.

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