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Electrical Vehicle Modeling and Backstepping Control

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Abstract

Nowadays, the development of electric vehicles has become a general trend. Electrical vehicles have improved their performance, and have been made suitable for commercial and domestic use during the last decades. The proportional–integral–differential (PID) controller has been widely used in the industrial field. It has a simple structure, and can be easily realized. The recursive backstepping design methodology is originally introduced inadaptive control theory to systematically construct the feedback control law, the parameter adaptation law and the associated Lyapunov function for a class of nonlinear systems satisfying certain structured properties. The backstepping control (BKC) is used to improve the robustness and real-time performance of the electrical vehicle system. Numerical simulation results show the effectiveness of this approach.

Keywords*: Electrical vehicle, backstepping controller, kinematic model, dynamic model*

1. Introduction

In this dissertation the effort is to explore new aspects, from both theoretical and application point of view, in the design of the backstepping control systems with applications to vehicle lateral control in Automated Highway System (AHS).

Automated systems were originally designed to replace humans in the performance of tasks tedious, dangerous or beyond their physical capabilities. The automotive field is a good example of this trend, witnessing a big number of automatic control systems applied for different application aspect. Since the 1980s, various active chassis vehicle control approaches have been investigated. In particular, research into vehicle dynamics control (VDC) or vehicle stability control systems has become very active, attracting intensive efforts from both the academic community and industry [1-4]. The main goals of VDC include improvements in vehicle safety, steerability, maneuverability, passenger comfort, and reduced driver workload, especially in adverse driving situations. For VDC systems, control states typically are longitudinal velocity, lateral velocity, and the yaw rate, while the actuation could include individual wheel drive/brake and four-wheel steering, comprising a redundantly actuated system.

ABS optimal control problem to maximize brake force without a priori knowledge of road friction coefficient has been presented in [6]. Using linearized lateral forces, the vehicle yaw dynamics have been regulated by a sliding mode wheel slip controller in [7]. Vehicle Dynamics Controller (VDC) supporting the drivers in controlling laterally critical and emergency situations has been proposed in [8]. Considerable research efforts have targeted coordinated VDC systems. Guvenc, et al. [5] proposed a vehicle yaw stability control approach coordinating steering and individual wheel braking actuations by using a simple coefficient to distribute the control effort between steering angle and braking torque.

In the present paper, we are most interesting in the automatic control of the vehicle. Then it is necessary to combine both longitudinal and lateral control of the vehicle. Hence, we develop control strategies improve the performances vehicle lateral dynamics as well as its robustness with regards to external disturbances such as varying road state.

This paper is organized as follow. First, the vehicle dynamic model is explicitly introduced. Then, based on the lateral model, backstepping control applied to the vehicle is described. Finally, the performances of the proposed control algorithms are evaluated by comparisons performed on the simulation results for longitudinal and lateral control of the vehicle.

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2. Modeling Of The Vehicle

The mathematical model of the vehicle is known to be very a complex and challenging problem. The vehicle is of 6 degrees of freedom (3 translations and 3 rotations). However, for our application, we consider only the lateral behavior of the vehicle in response to a solicitation from the steering angle, with only three involved degrees of freedom: rotation around the vertical axis (cape angle), translations along X and Y axis (longitudinal and lateral).

We focus in this section on the various steps to derive the positions, speeds and accelerations of the vehicle in translation and in rotation. The fundamental principles of dynamics relate on the one hand, the equilibrium of external forces acting on the vehicle and on the other hand, the moment equilibrium dynamics of the vehicle relative to the external moments.

Newton's principle $$\sum \vec{F}_{ext} = m\vec{\gamma} \quad (1)$$

$$\sum \vec{M}_o = I \frac{d}{dt}\vec{\Omega} \quad (2)$$

Hypothesis

For simplification reasons, several hypotheses are necessary:

1. The road is considered flat without pant nor quotation, and without slope.
2. The translation movements is reduced to two degrees of freedom: the longitudinal translation and the lateral translation.
3. The rotation movement is reduced to a single degree of freedom which corresponds to the movement yaw Ψ.
4. The gravity center of the vehicle is confused with the origin of bound (connected) had mark convey R_V.

$$C_G = O_V \quad (3)$$

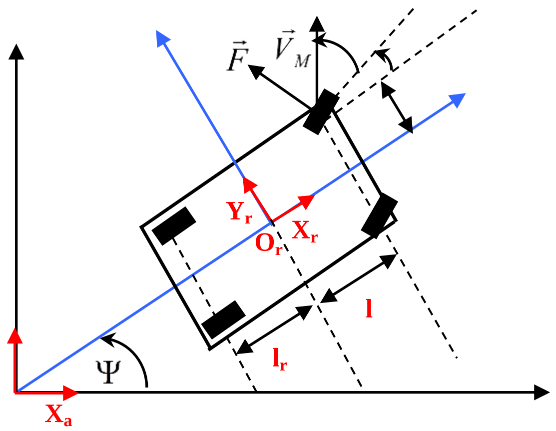

Figure 1. Angle of drift of the left front wheel

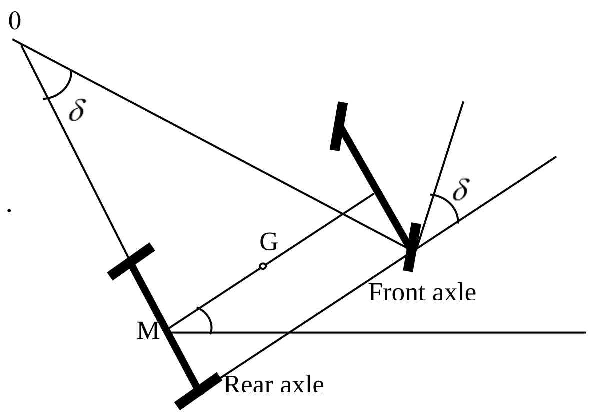

Figure 2. Geometrical model of the vehicle

2.1. Dynamique Model of the Vehicle

Equation below, give the general form of the kinematic model vehicle's.

$$\sum(F_x) = \dot{v}_x - \dot{\Psi}v_x \quad (4)$$

$$\sum(F_y) = \dot{v}_y + \dot{\Psi}v_y \quad (5)$$

$$\sum(M_z)=I_z\ddot{\Psi} \quad (6)$$

F_x : external forces along X axis.
F_y : external forces along Y axis.
M_Z: external moments.
I_Z : the moment of inertia.
$\dot{V}_X$: acceleration along X axis.
$\dot{V}_y$: acceleration along Y axis.
$\dot{\Psi}$: rotational speed.
$\ddot{\Psi}$: rotational acceleration.

From the previous relations and simplifications were allowed by our hypotheses and in linearization around a steering I considered small on highway, we obtain the expressions of the dynamic model in a mark bound (connected) to the vehicle:

$$\left(\frac{m.v_x}{2.C}\right).\dot{v}_y+v_y=\left(\frac{v_x}{2}\right).\delta-\left(\frac{m.v_x^2}{2.C}\right).\dot{\Psi} \quad (7)$$

$$\left(\frac{2.I_z.v_x}{L^2.C}\right).\ddot{\Psi}+\dot{\psi}=\left(\frac{v_x}{L}\right).\delta \quad (8)$$

$$\dot{y}=v_x\sin\Psi+v_y\cos\Psi \quad (9)$$

In the with m the mass of the vehicle and I_z the moment of inertia with compared with the vertical axis. Seen a symmetry of the vehicle with compared with center of gravity C_G we can note:

$$L_f = L_r = L/2 \quad (10)$$

Where L is the distance in axles. Generally, the pneumatics of front and back trains are identical, that is, the coefficient of stiffness are identical.

$$C_f = C_r = C \quad (11)$$

In a previous work if we considered that the speed *Vx* is constant which makes the system or the dynamic but simple system in this work we introduced the variation of this variable *Vx* so we can speak in this case of a system with a very complicated dynamic or strongly coupled. In this case the control problem is very difficult and in the same way we did design a controller that is based on the backstepping technique

2.2. Kinematic Model

In this section we consider a simple model of the vehicle called in classic mechanics kinematic model. In this case, that the inertia and mass of the vehicle are negligible. The figure 2 represents the geometrical model of the vehicle, where C_G is the center of gravity, L is the distance between the front and rear axles of the vehicle and M is anointed environment (middle) of the back axle.

We are interested in the lateral to define the equations of the model. We extract the following non linear kinematic model:

$$\dot{X}=v_x\cos\Psi-(L/2).\sin\Psi.\dot{\Psi} \quad (12)$$

$$\dot{Y}=v_x\sin\Psi+(L/2).\cos\Psi.\dot{\Psi} \quad (13)$$

$$\dot{\Psi} = (v_x / L).tng\delta = (v_x / L).\delta \quad (14)$$

2.3. Kinematic Model

So far we have considered that the road was perfectly straight, that is, with zero curvature. In the case of highway driving, the road is not always a straight line. [9] found that the disturbance due to the curvature can be modeled by an equivalent input on the steering angle directly proportional to curvature (Kc).

So the curved line model is the same model in a straight line with increasing the steering angle by L / R, where R the radius of curvature.

$$\delta_{curvature} = \delta_{straight} + L / R \quad (15)$$

3. Control Of The Vehicle

Figure 3 shows the bloc diagram control of the vehicle and the different variables are described below.

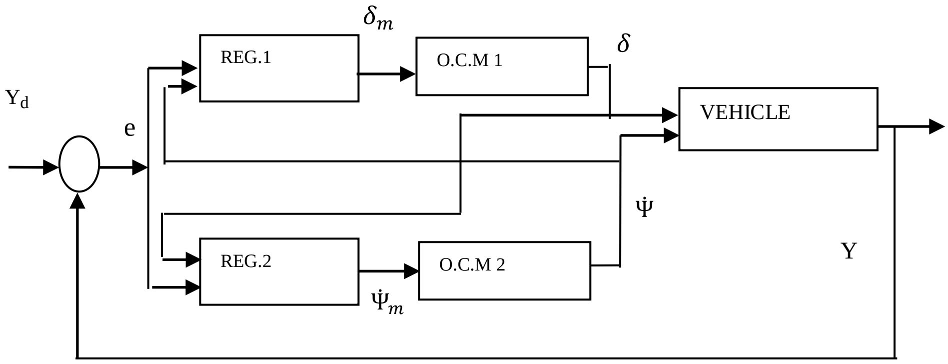

Figure 3. Block diagram of the control

Y_d: The desired distance.
E: The error.
O.C.M 2: a control device of $\dot{\Psi}$.
REG2: Controller 2.
δ : The controlled variable 1.
Ψ: The controlled variable 2.

Y: The real distance (the measure).
O.C.M 1: a control device of δ.
REG1: Controller 1.
δ_m : the control signal supplied by the REG1
$\dot{\Psi}_m$: the control signal supplied by the REG2
Vehicle

3.1. Backstepping Controller

The backstepping control design procedure has been used to develop stabilizing controllers for time invariant plants that are linear or belong to some class of nonlinear systems. The backstepping design technique is utilized to design the lateral controller in this study. Firstly, we will regard that the relative yaw angle be the control input in Eq (17). Then, we 'backstep' to determine the steering angle in Eq (18).
From the dynamics we have the following equations:

$$\dot{v_x} = v_y\dot{\psi} + \frac{c}{m}.\frac{v_y}{v_x} + \frac{L.c}{2.m}.\frac{\dot{\psi}}{v_x}\delta \quad (16)$$

$$\dot{v_y} = -v_x\dot{\psi} + \frac{c}{m}.\delta - \frac{2.c}{m}.\frac{v_y}{v_x} \quad (17)$$

$$\ddot{\psi} = \frac{Lc}{2I_Z}\delta - \frac{L^2}{2I_Z}.\frac{\dot{\psi}}{v_x} \quad (18)$$

We take the equation (17): It is equivalent to the following equation:

$$\ddot{y} = -v_x\dot{\psi} + \frac{c}{m}.\delta - \frac{2.c}{m}.\frac{\dot{y}}{\dot{x}} \quad (19)$$

We extract $\dot{y}$ from equation (19), we obtain:

$$\dot{y} = (\frac{\dot{x}}{2})\delta - (\frac{m\dot{x}^2}{2C})\dot{\psi} - (\frac{m\dot{x}}{2C})\ddot{y} \quad (20)$$

And we have the equation of exit (21):

$$\dot{y} = \dot{x}\sin(\psi) + \dot{y}\cos(\psi) \quad (21)$$

Thus we replace the equation (20) in the equation (21):

$$\dot{y} = \dot{x}\sin(\psi) + \left[\left(\frac{\dot{x}}{2}\right)\delta - \left(\frac{m\dot{x}^2}{2C}\right)\dot{\psi} - \left(\frac{m\dot{x}}{2C}\right)\ddot{y}\right]\cos(\psi) \quad (22)$$

The first error variable is defined as:

$$e = y_r - y \quad (23)$$

Or both equation (23), is associated to the following Lyapunov function:

$$V = \frac{1}{2}e^2 \quad (24)$$

Its derivative is written:

$$\dot{V} = e\dot{e} \quad (25)$$

With:

$$\dot{V} \leq 0 \quad (26)$$

Derivative also wrote:

$$\dot{V} = -Ke^2 \quad (27)$$

With:

$$K > 0 \quad (28)$$

From equation (25) and (27) we obtain:

$$e\dot{e} = -Ke^2 \quad (29)$$

So:

$$\dot{e} = -Ke \quad (30)$$

From equation (23) and (30) we obtain:

$$\dot{y} = \dot{y_r} + Ke \quad (31)$$

Taking equation (21)

$$\dot{y} = v_x\, sin\,\psi + v_y\, cos\,\psi \quad (32)$$

From equation (17) we deduce v_y:

$$v_y = \frac{v_x}{2}\delta - \frac{m}{2C}v_x{}^2\dot{\psi} - \frac{m}{2C}v_x\dot{v}_y \quad (33)$$

We replace the equation (33) in (32), we obtain:

$$\dot{y} = v_x \sin\psi + [\frac{\dot{x}}{2}\delta - \frac{m}{2C}v_x{}^2\dot{\psi} - \frac{m}{2C}v_x\dot{v}_y] \cos\psi \quad (34)$$

We replace the equation (34) in (31), we obtain:

$$v_x \sin\psi + \left[\frac{v_x}{2}\delta - \frac{m}{2C}v_x{}^2\dot{\psi} - \frac{m}{2C}v_x\dot{v}_y\right] \cos\psi = \dot{y_r} + Ke \quad (35)$$

Note that in equation (35) that there are two inputs control δ and $\dot{\psi}$.
-The first input control δ:

$$\delta = \frac{2}{V_x COS\Psi}[\dot{y}_r + K_1 e - v_x \sin\psi + \frac{m}{2c}v_x{}^2\,\dot{\Psi}\cos\,\Psi + \frac{m}{2c}v_x\dot{v}_y\cos\,\Psi \quad (36)$$

-The second input control $\dot{\psi}$:

$$\dot{\psi} = \frac{2c}{m\,v_x{}^2\cos\psi}[-\dot{y}_r - K_2 e + v_x \sin\psi + \frac{v_x}{2}\delta\cos\psi - \frac{m}{2c}v_x\dot{v}_y\cos\psi] \quad (37)$$

With K1, K2 are positive constants design which determines the dynamics of the feedback.
We developed in this work the backstepping controller for the dynamic model for both coupled control inputs δ (steering) and $\dot{\psi}$ (cape).

4. Simulation

Numerical simulations have been done using SIMULINK software under MATLAB environment. The vehicle parameters considered in simulation are:

Table 1. The Performance of vehicle

Y_d (m)	M (kg)	I_z (kg/m)	C (N/rad)	L (m)
1.5	1200	204	45000	3

In the first place, we consider the state initial of the longitudinal speed of vehicle V_x as 30 m/s is equal at 108km / hour.
The initial position of the vehicle is considered as zero (0).

4.1. Staight line

After several trials one noticed that: if decreased k goes away to decrease the transitory regime as well as the overtaking is eliminated. Thus we are going to fix (λ=1.2, k=0.001), we obtain the following result.

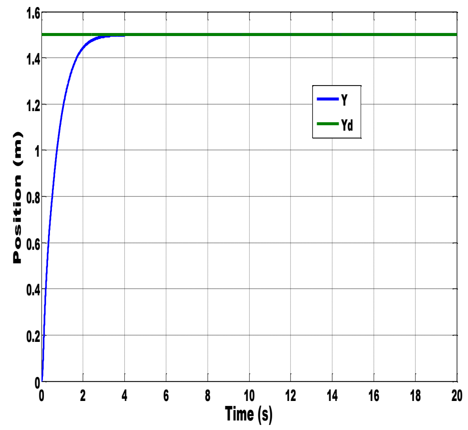

Figure 4. Variation of lateral position

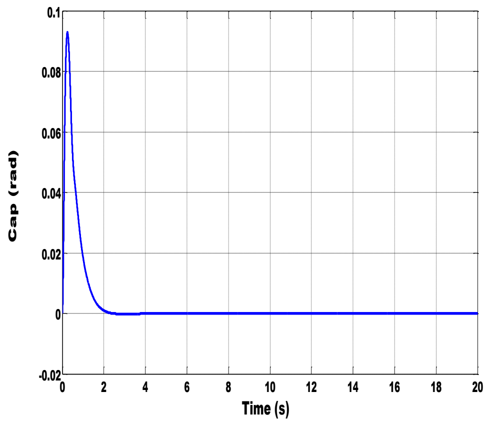

Figure 5. The cape angle

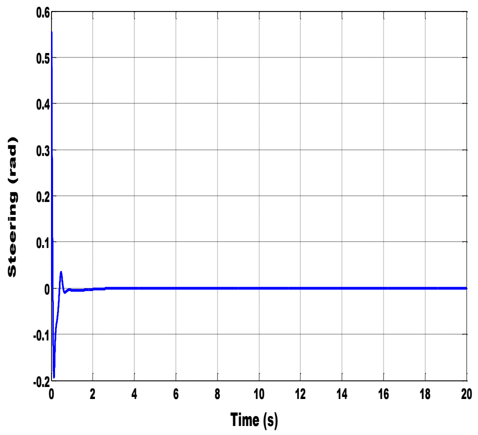

Figure 6. The steering angle

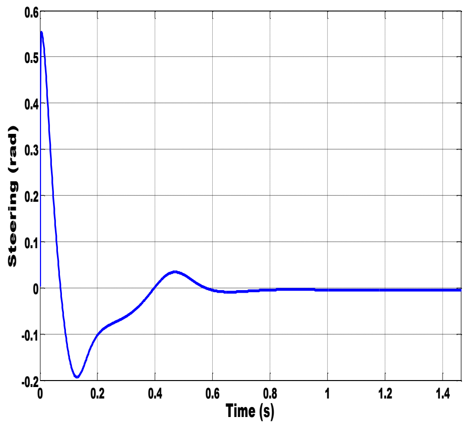

Figure 7. The steering angle

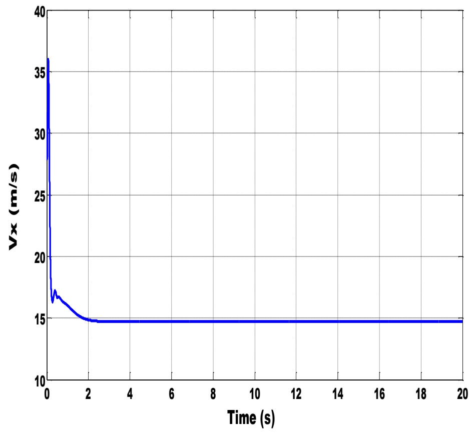

Figure 8. Variation of the speed V_x

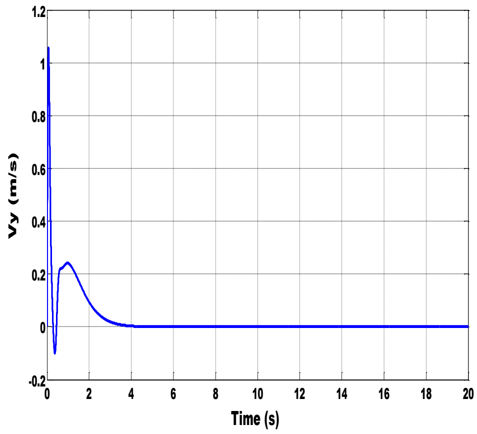

Figure 9. Variation of the speed V_y

In the Figure 4 shows that the transient time is equal to 4 sec, so the vehicle is on the desired path from the second fourth. The overshoot is removed and the steady state error is zero. The same goes in Figures 5 and 6 for the steering and the cape, they tend to zero after the fourth second.

It was also noted in Figure 7 the maximum steering angle of the wheels is equal = 0.47rad and a zero steady state error.

δ_{max}=(0.6*180)/pi with pi=3.14. δ_{max}=26.94°

δ_{max} is the acceptable values of steering because the maximum steering in the real vehicule tend to 45°.

We can visualize also in figures (8) and (9) the longitudinal speed (Vx) and the lateral speed (Vy). We notice crossing the speed V_x goes to 15 m/sec and the speed Vy tends to 0 it means that when the vehicle is on the distance desired both steering angle and speed Vy are equal to 0.

For a straight line behavior testing, we noticed that our controller has responded well, he gave satisfactory responses with no overshoot, a reasonable settling time, and the steering angle does not exceed the maximum turning wheels possible error in the steady state zero

4.2. Line Change

The lane change is a test consists in given a distance wished y_d (a level of 1.5m) then in 15sec we change the way by application of a level of 0m.

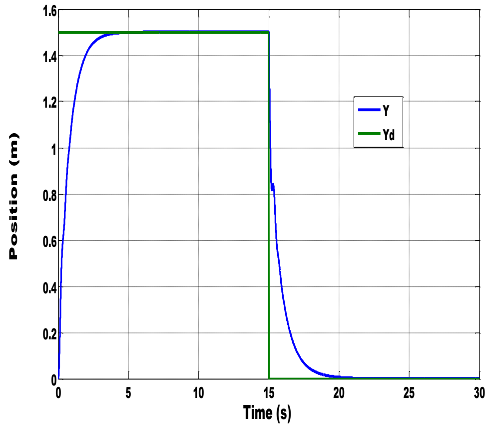

Figure 10. Variation of lateral distance

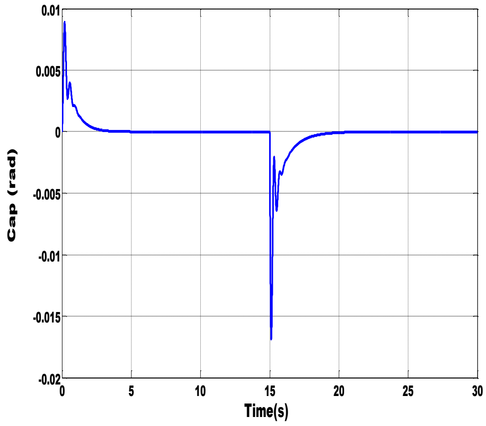

Figure 11. The cape angle

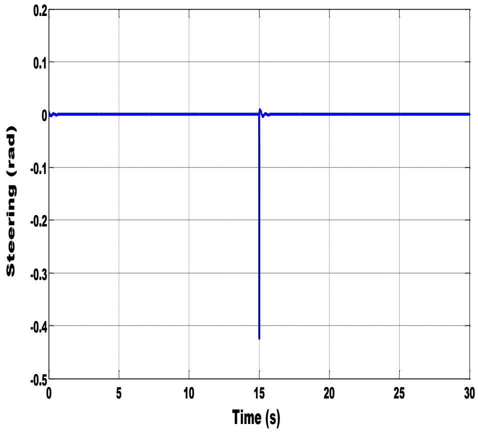

Figure 12. The steering angle

We remark in the figure (10) that the variation of the measured position follows the desired position. The same thing the previous figure shows that the transient time is equal to 5 sec, the overshoot and the steady state tend to zero.

Figures 11 and 12 present the variation of the cape angle and the steering angle for lane change.

4.3. Curvature line

The disturbance that we applied to our vehicle represents a turning radius of 200m. We have already defined the effect of curvature previously, so we increase our output bakstepping controller with L / R = 3/200.

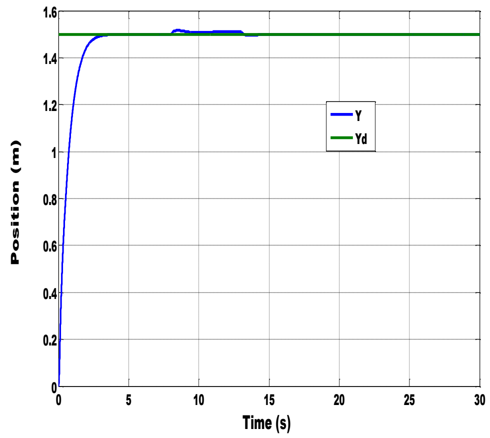

Figure 13. Variation of lateral distance

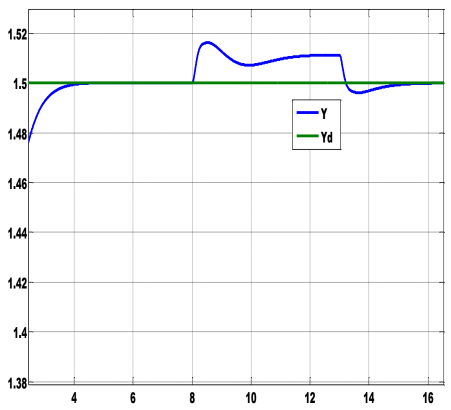

Figure 14. Variation of lateral distance (zoom)

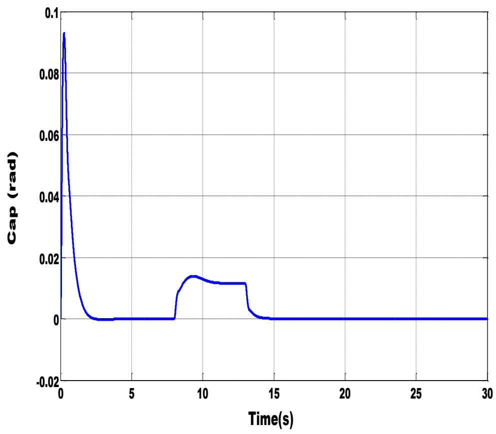

Figure 15. The cape angle

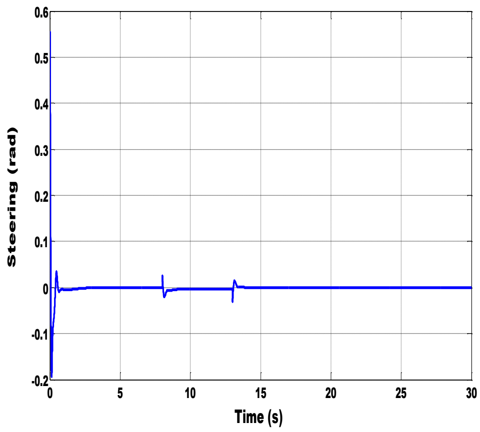

Figure 16. The steering angle

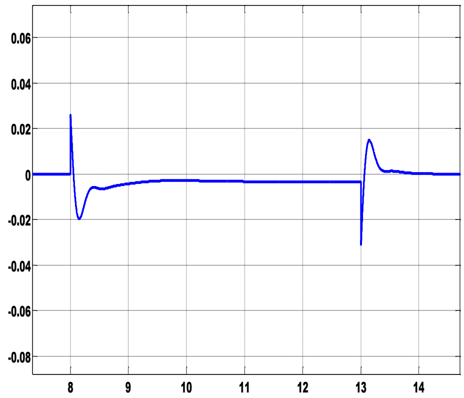

Figure 17. the steering angle (zoom)

We observe that the Figure 14 present the overshoot while the turn inferior to 0.02m, the same things in the Figure 15 the cap angle.we see in Figure 16 that the steering angle of the bend is negative which means that the steering is in the process of returning the vehicle to its desired position (Yd)

5. Conclusion

A backstepping controller was designed for lateral guidance of the electrical vehicle. In this design, the vehicle lateral displacement is affected by the relative yaw angle of the car with respect to the road centerline, and the relative yaw angle is controlled by the vehicle's front wheel steering angle. The closed loop performance of lateral and yaw dynamics can be speciffed and guaranteed simultaneously in this approach. The backstepping control has been applied to the electrical vehicle system with different situations: straight line, curvature line and lane change. Numerical simulation shows that the used controller is robust to external disturbances and it can make the system stable, accurate and fast.

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