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A New Nonlinear control strategy for a vehicle trajectory tracking in the presence of faults

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Abstract— **Starting from the thought that vehicle's dynamics and its control play an important role in an automated highway system for passenger cars, and continuing the work presented in [1] and [2], that deal with the dynamics of the vehicle, and the study of the influence of faults on the movement of the center of gravity and on the stability of the vehicle. we present in this work two important ideas, in the first one, the trajectory tracking control problem in the simple cases where there are no faults, and in the second one, the response of the proposed controller in the case of faults. In the control design, we will propose a nonlinear predictive controller, in which we will integrate by this approach both the active wheel steering control and the four wheel torques. For testing the validity of our proposed controller, a set of computer simulations, describing different trajectories are made; reasonable results will be demonstrated and shown.**

I. Introduction

FOLLOWING the increasing demands for safety and comfort, and starting from the fact that safe driving requires the driver to react extremely quickly in a dangerous situations, which is not possible as the driver who can be modeled as a high gain control system with a dead time overreacts, that result the instability of the system, consequently, the improvement of the vehicle dynamics by active chassis control is necessary for such catastrophic situations. The current strategy adopted by car manufactures in the development of thus control systems consists in replacing the driver for the simplest tasks and in assisting him as much as possible in the more complicated ones. In the plane (X, Y), the three main systems of chassis control are: lateral control, longitudinal control and the yaw control, which will be the objective of this paper.

Considerable attention has been given to the development of the control systems over the past few years, authors have investigated and developed different methods and different strategies for enhancing the stability and the handling of the vehicle such as, the design of the active automatic steering[4], the wheel ABS control[5], or the concept of a four-wheel steering (4WS) system which has been introduced to enhance vehicle handling [8] and [9],Some researchers have shown disadvantages on 4WS vehicles [6] and [7]. In this paper, we will employ the concept and the study of steering (4WS) and we will try to regulate the controller in a way that to stabilize as maximum as possible the comportment of the vehicle without the rear steering and that only for cost and real implementation issues in active car steering.

The objective is to have a vehicle follow a desired path by appropriately altering the steering angles and the couples, and to test the behaviour of this controller by tracing comportment of the center of gravity in the presence of faults in a closed loop system. By the fact that the relation between the real vehicle model and the input command is not bijective, we can find, and that will be demonstrated by simulations, that for following a certain trajectory In the plane (X, Y), more than one control input vector can be put.

This work continues the work [1] and [2], in which we have studied the comportment of the center of gravity of the vehicle in an open loop system, without the application of any control law, if a certain fault such as inflation pressure occurs at a one of its wheels,. In this paper, a non-linear controller is designed in order to test the comportment of the center of gravity, in a closed loop system, and to make this vehicle follows a predefined trajectory, the comportment of the center of gravity will be also tested in the presence of faults.

The remainder of this paper is organized as follows. We present the vehicle model for steering and wheel torque controls in Section 2. Section 3 deals with the design of the predictive controllers for vehicle behavoir. In Section 4, we present extensive simulations for the effectiveness of the control algorithms. Section 5 is devoted to some contributions and conclusions of the work.

Nomenclature

Ω_i is the angular velocity

r_i is the radius of the wheel i

C_{mi} is the motor couple applied at wheel i

C_{fi} is the braking couple applied at wheel i

F_{xi} is the longitudinal applied at wheel i

F_{yi} is the lateral force applied at each wheel i

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M is the total mass of the vehicle.

$F_{x\,res}$ is the forces of air resistance in X

Fy_{res} is the forces of air resistance in Y

$M_{z\,res}$ is the moment resulted from air resistance around the Z direction

ψ is the yaw angle

V_ψ is the yaw velocity [rad/s].

δ_f is the front steering angle

δ_r is the rear steering angle

V_x is the longitudinal velocity of the center of gravity

V_y is the lateral velocity of the center of gravity

V is the total velocity of the center of gravity in the (X, Y) plane.

L_1 is the distance between the center of gravity and the front axis center of gravity

L_2 is the distance between the center of gravity and the rear axis center of gravity

v_{xi} is the longitudinal velocity of the tire i

v_{yi} is the lateral velocity of the tire i

h is the height of the center of gravity

t_f is the front half gauge

t_r is the rear half gauge

p_i is the inflation pressure of the wheel i

α_i is the slip angle of the wheel i

F_{zi} is the vertical force

η_x is coefficient of adherence

λ_i is the slipping of the wheel i

X_p is the length of the contact patch

X_m is the length of the adhesion region patch

X_s is the state vector representing the model

T_r is the time constant relating to the rear steering

T_f is the time constant relating to the front steering

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II. VEHICLE MODEL

As described in [1] and [10], from the fundamental principles of dynamics the movement of the center of gravity in the plane (X, Y) is described as, figure(1).

$$\begin{aligned} M\dot{V}_x &= MV_\psi V_y + F_{x\,res} + \cos\delta_f[F_{x1} + F_{x2}] \\ &+ \cos\delta_r[F_{x3} + F_{x4}] - \sin\delta_f[F_{y1} + F_{y2}] \\ &- \sin\delta_r[F_{Y3} + F_{Y4}] \end{aligned} \tag{1}$$

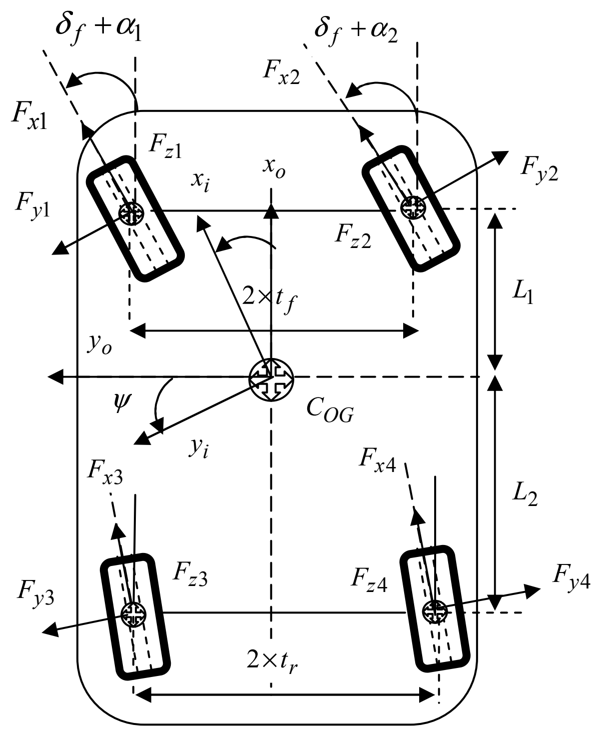

Fig.1:2 D-vehicle scheme

$$\begin{aligned} M\dot{V}_y &= -MV_\psi V_y + F_{yres} + \sin\delta_f[F_{x1} + F_{x2}] \\ &+ \sin\delta_r[F_{x3} + F_{x4}] + \cos\delta_f[F_{y1} + F_{y2}] \\ &+ \cos\delta_r[F_{y3} + F_{y4}]. \end{aligned} \tag{2}$$

$$\begin{aligned} I_z\dot{V}_\psi &= M_{z\,res} + l[\cos\delta_f(F_{x2} - F_{x1}) + \sin\delta_f(F_{y1} - F_{y2})] \\ &+ L_1[\sin\delta_f(F_{x1} + F_{x2}) + \cos\delta_f(F_{y1} + F_{y2})] \\ &- L_2[\sin\delta_r(F_{x3} + F_{x4}) - \cos\delta_r(F_{y3} + F_{y4})] \\ &+ l[\cos\delta_r(F_{x4} - F_{x3}) + \sin\delta_r(F_{y3} - F_{y4})]. \end{aligned} \tag{3}$$

$$I_r\dot{\Omega}_i = -r_i F_{xi} + C_{mi} - C_{fi}, \quad i = 1, \ldots, 4 \tag{4}$$

where F_{xi} and F_{yi} are non linear analytical functions depending on, p_i, η_x, δ_f, α_i, λ_i, as in [2],

As described in [3], we can write this model in the following manner,

$$\dot{X}_s = f(X_s) + g(X_s, U). \tag{5}$$

with $X_s = [V_x \quad V_y \quad V_\psi \quad \Omega_1 \quad \Omega_2 \quad \Omega_3 \quad \Omega_4]^T$.

We will increase the number of the state variables in the state vector in order to assemble both the rear and the front steering angles, and then we will decompose this state vector in two state vectors in order to have a cascaded system, and to facilitate its control.

$$X_{s1} = [V_x \quad V_y \quad V\psi]^T \tag{6}$$

$$X_{s2} = [\Omega_1 \quad \Omega_2 \quad \Omega_3 \quad \Omega_4 \quad \delta_f \quad \delta_r]^T \tag{7}$$

So, the system is divided as following:

$$\begin{cases} \dot{X}_{s1} = f_1(X_s) \\ \dot{X}_{s2} = f_2(X_s) + B\cdot U \end{cases} \tag{8}$$

with $B = \begin{bmatrix} B_{11} & 0 & 0 & . & . & 0 \\ 0 & B_{22} & 0 & . & . & 0 \\ 0 & 0 & B_{33} & . & . & 0 \\ . & . & . & B_{44} & . & 0 \\ . & . & . & . & B_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & B_{66} \end{bmatrix}$

where:

$$B_{11} = B_{22} = B_{33} = B_{44} = \frac{1}{Iz}, B_{55} = \frac{1}{T_f} \ and\ B_{66} = \frac{1}{T_r}$$

The non linear functions f_1 and f_2 can be obtained directly from equation (1, 2, 3 and 4).

$U = [C_{c1} \quad C_{c2} \quad C_{c3} \quad C_{c4} \quad C_{sf} \quad C_{sr}]^T$

Where C_{ci} defines the control couple applied at wheel i, C_{sf} and C_{sr} define the front and the rear steering control.

The front and the rear steering angles have steering actuator dynamics which are represented as a first order lag systems with unity gain, and which have the time constants T_f and T_r, and can be calculated from the following equations:

For the front steering

$$\dot{\delta}_r = -\frac{\delta_r}{T_r} + \frac{C_{sr}}{T_r} \qquad (9)$$

For the rear steering

$$\dot{\delta}_r = -\frac{\delta_r}{T_r} + \frac{C_{sr}}{T_r} \qquad (10)$$

III. Controller design

This section deals with the designing of a controller basing on the principle of the predictive control. In this work, we are interested by the position of the center of gravity in the (X,Y) plane, so for achieving that goal, and due to the direct relations between the state space X_{s1} and our goal, we suppose that the desired reference trajectory is defined by the longitudinal, lateral and the yaw velocities of the center of gravity, which is X_{s1} reference and we have to control our model by certain U in order to achieve this desired trajectory, a small schema is designed to explain the control figure.2.

From equation (8), X_{s1} is not directly commendable by U, but it is clear that, X_{s1} is a function of X_{s2}, in which we can see the relation with the command U.

From the principle of the predictive control, in which it we base on the state of the system at instant t, to predict the appropriate control at instant $(t+\tau)$, the step of prediction τ, in fact not constant, in this work we will suppose that it is constant and small enough, in order to have more precision, and the study of its variation will be considered as a perspective of this work.

To determine the control U we have to minimize the tracking error e_1 at $(t+\tau)$, so the first equation we will be re written as

$$J = \frac{1}{2} e_1(t+\tau)^T Q e_1(t+\tau) + \frac{1}{2} U^T R U \qquad (11)$$

Where:

$$e_1 = [X_{s1} - X_{s1ref}] \qquad (12)$$

(12)

The matrix R is positive definite and Q is positive semi definite, these two matrices are weighing matrices chosen in a way to have a minimum error and an appropriate control.

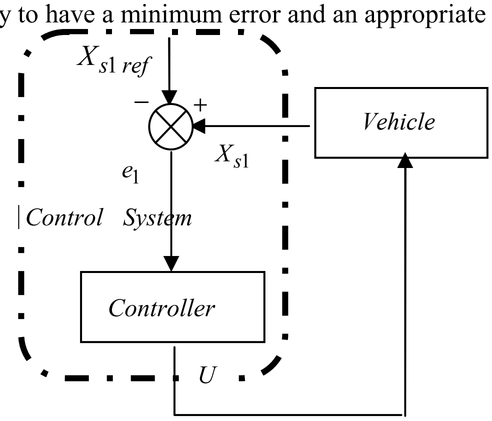

Fig.2: controller schema

The interest of minimizing the function J is that the differences between the reference values and the model values are as small as possible in order to follow the desired trajectory. A finite series expansion to order 2 can be used for the approximation of $e_1(t+\tau)$, as follows:

$$e_1(t+\tau) \cong e_1(t) + \tau \times \dot{e}_1(t) + \frac{\tau^2}{2} \times \ddot{e}_1(t) \qquad (13)$$

Where: $\dot{e}_1(t) = \dot{X}_{s1} - \dot{X}_{s1ref}$, and $\ddot{e}_1(t) = \ddot{X}_{s1} - \ddot{X}_{s1ref}$.

In order to evaluate this last equation, we have to make the derivative of the accelerations, and that means the derivative of each term in function of time, applying Chains role, we get

$$\ddot{X}_{s1}(t) = \frac{d}{dt}\dot{X}_{s1}(t) = \frac{\partial \dot{X}_{s1}}{\partial X_{s1}} \times \frac{dX_{s1}}{dt} + \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} \times \frac{dX_{s2}}{dt} \qquad (14)$$

Then

$$\ddot{e}(t) = \frac{\partial \dot{X}_{s1}}{\partial X_{s1}} \times \frac{dX_{s1}}{dt} + \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} \times \frac{dX_{s2}}{dt} - \ddot{X}_{s1ref} \qquad (15)$$

Combining the two equations (8) and (15), we have:.

$$\ddot{e}_1(t) = \frac{\partial \dot{X}_{s1}}{\partial X_{s1}} f_1(X_s) + \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} [f_2(X_s) + B \cdot U] - \ddot{X}_{s1ref} \qquad (16)$$

Then we have

$$e_1(t+\tau) = e_1(t) + \tau \times \dot{e}_1(t) + \frac{\tau^2}{2} \times \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} \times B \cdot U + \frac{\tau^2}{2}\left(\frac{\partial \dot{X}_{s1}}{\partial X_{s1}} f_1(X_s) + \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} f_2(X_s) - \ddot{X}_{s1ref} \right). \qquad (17)$$

To minimize J, the necessary condition is that

$\frac{\partial J}{\partial U} = 0$; which leads to:

$$U(t) = -\frac{\tau^2}{2} \times A^{-1}(t) \times \left(\frac{\partial \dot{X}_{s1}}{\partial X_{s2}} \times B \right)^T \times Q \times \left(e_1(t) + \tau \times \dot{e}_1(t) + \frac{\tau^2}{2} \times \left(\begin{array}{l} \frac{\partial \dot{X}_{s1}}{\partial X_{s1}} \times f_1(X_s) + \\ \frac{\partial \dot{X}_{s1}}{\partial X_{s2}} \times f_2(X_s) - \\ \ddot{X}_{s1ref} \end{array} \right) \right). \quad (18)$$

Where

$$A = \frac{\tau^4}{4} \times \left(\frac{\partial \dot{X}_{s1}}{\partial X_{s1}} \times B \right)^T \times Q \times \left(\frac{\partial \dot{X}_{s1}}{\partial X_{s1}} \times B \right) + R \quad (19)$$

IV. SIMULATIONS AND RESULTS

This section presents some simulation results in order to demonstrate three important ideas:

Firstly: there is no one to one relation between the system and the command, and in reality this is trivial because if we take for example two drivers, and we ask them to pass a certain road, we see that the two drivers may cut the same distance in the same duration of time, without the necessity of applying the same command at each instant. In this work we take firstly the component V_x, and we trace it with respect to the couples applied to the wheels (couple1,3) in figure(3,4) and with respect the front steering angle δ_f figure(5), we find that there are a set of combinations between the couples and the front steering angle that give the same value of the longitudinal velocity , this demonstration is made to insure that we can vary the weighing matrices if we want to give the importance to one element in the control input or to ignore one other, and that what happens in the simulations when we try to ignore δ_r.

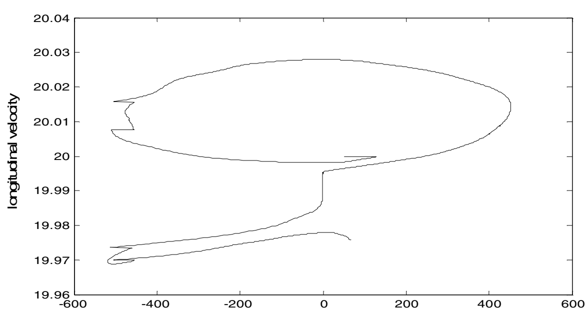

Fig.(3): the variation of the longitudinal velocity in function of the couple delivered to wheel 1

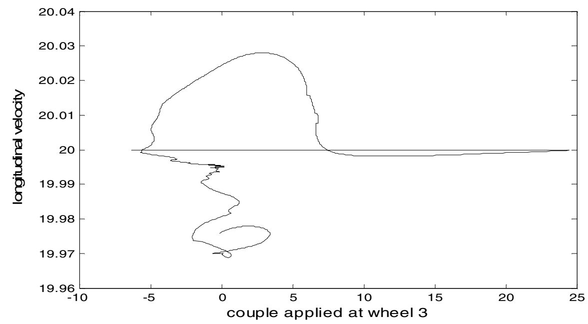

Fig.(4): the variation of the longitudinal velocity in function of the couple delivered to wheel 3

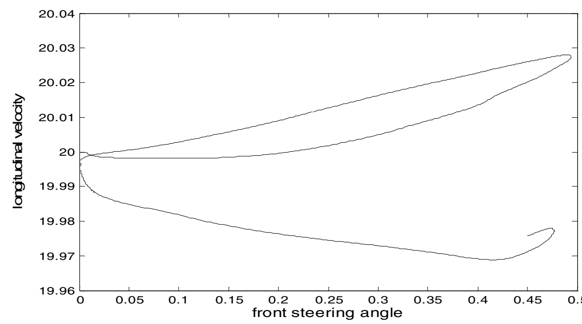

Fig.(5):the variation of the longitudinal velocity in function of the front steering angle

Secondly: that the controller is able to make the center of gravity of the vehicle follows a certain desired trajectory in the absence of the driver in the normal cases, meaning with no faults, in this subsection, we will aliment the system with a real inputs, real couple figure(6); and a filtered steering angle (the dashed curve of figure (7)), in figure (7) also we see a solid line curve which represents the steering angle given by the controller, figure (8)shows the 4 command couples applied at each wheel, here couple means (the motor couple (C_{mi}) -the braking couple (C_{fi})), figure (9) shows the angular velocities of the four wheels after the control, they are logical and reasonable because we see that the input couples are small and they are rounding around zero, that means the variation in the wheels angular velocity is rounding around the initial value. in reality, in the vehicle , there are sensors to measure the global velocity of the center of gravity and V_ψ and not for V_x and V_y , In this paper, we suppose that the three velocities of the center of gravity in the (X,Y) plane are given as a reference values, In fact, un measured velocities may be found using an observer, in which the use of sliding mode observer for the determination of velocities will be one of the perspectives of this work

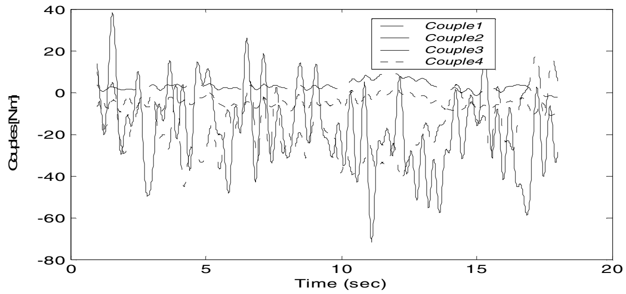

Fig.(6):the 4 couples applied at the wheels

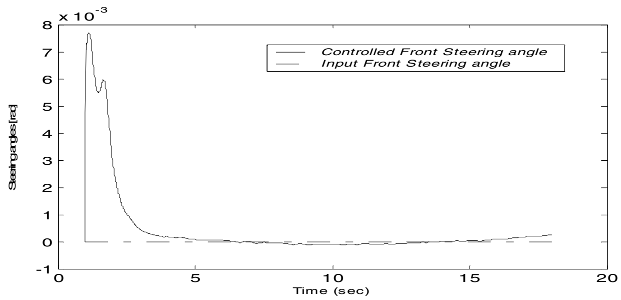

Fig. (7): the steering angle of reference and the command steering angle

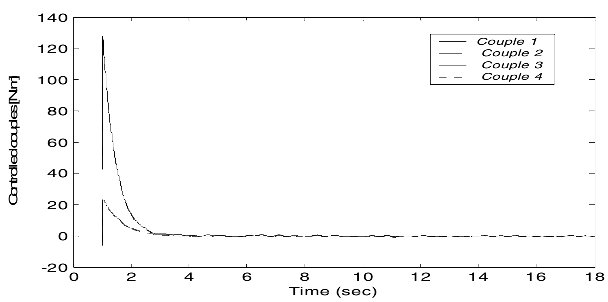

Fig.(8): the four wheel command couples

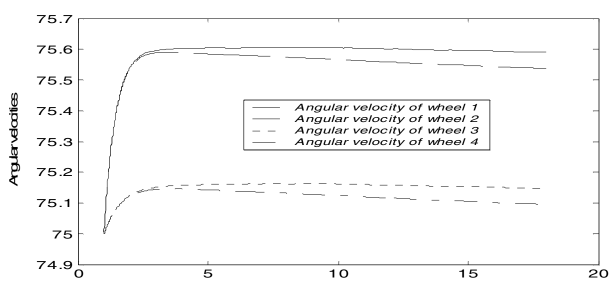

Fig. (9): the four angular rotations of the wheels after the second controller.

Thirdly : the behavior of this controller , when faults appear at different instances and on different wheels, for that purpose , in each figure we have two curves , one to represent the model in the absence of faults, and another one to represent it's comportment in the presence of faults,. So the same strategy will be shown as above but after 10 seconds an inflation pressure problem happens. .

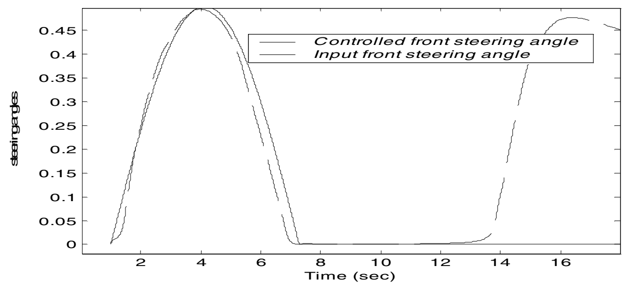

Fig. (10): the steering angle of reference (solid line) and the command steering angle (dashed line).

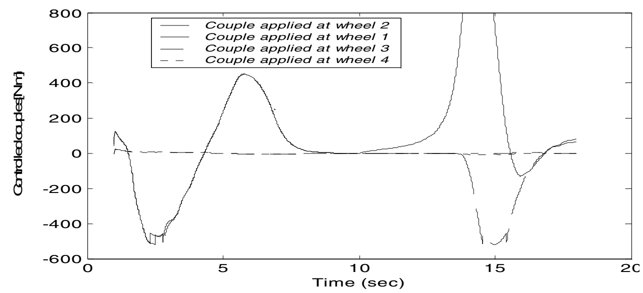

Fig.(11): the four wheel command couples

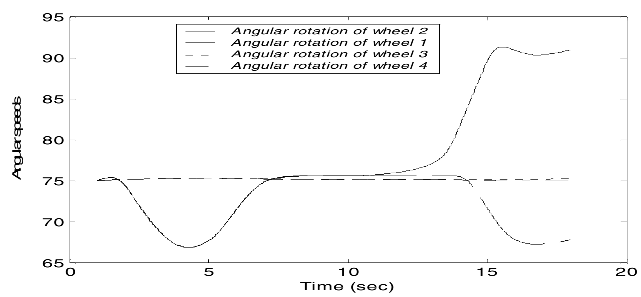

Fig. (12): the four angular rotations of the wheels after the second controller , the first four element in the vector

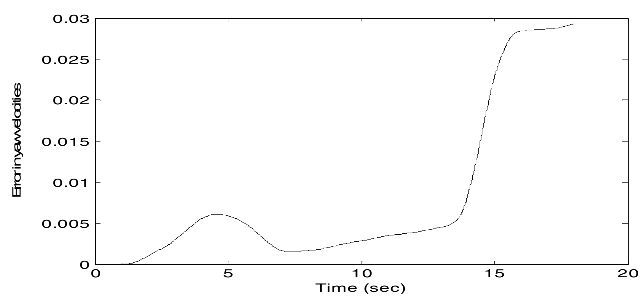

Fig. (13): the error in the Yaw

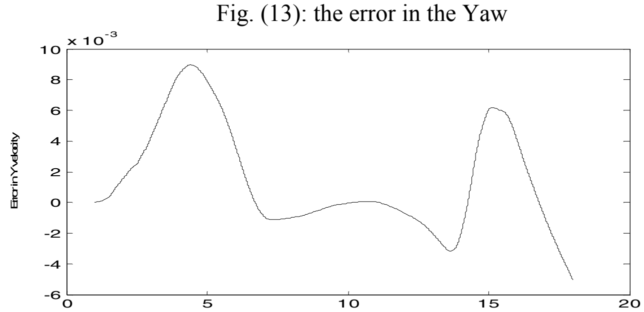

Fig. (14): the error in the V_y

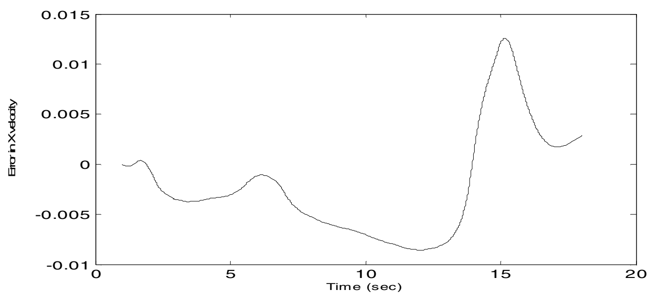

Fig. (15): the error in the V_x

V. CONCLUSION

In this paper, we have continued the work presented in [1] and [2], and that by presenting a nonlinear controller to solve the problems of trajectory tracking and to find a controller that may replaces the driver in the simple cases, and to help him in more difficult cases, computer simulations are made to show the influence of this controller at the center of gravity of the vehicle, reasonable results are demonstrated and shown.

VI. REFERENCES

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