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Behavior Modeling and Simulation of Double Wheeled Electric Vehicle Drive

***Abstract**. The principle of the electric vehicle is very simple: instead of an internal combustion engine with fuel stored in a tank and whose power is transmitted to the wheels through a gearbox, electric vehicle (EV) is powered by a traction system consisting of a central or two-wheel drive supplied by batteries, whose speed is controlled by an electronic control device. The propulsion system considered in this work is two driving wheels. It consists of two permanent magnet DC motors (PMDCM) directly coupled to the rear wheels of the vehicle forming a motor-wheel (driving wheel). Each motor-wheel is supplied by a static converter which is powered by batteries. The two sub systems (source-converter-motor) are coupled to an electronic differential in order to compensate the tendencies of direction of the vehicle and maintain a steady speed by adjusting the difference in speed of each motor-wheel according to the direction in the case of a turn.*

***Streszczenie.** W pojazdach elektrycznych stosuje się niekiedy system podwójnego napędu z elektronicznym sterowaniem. System składa się z dwóch silników prądu stałego bezpośrednio dołączonych do tylnych kół. (**Modelowanie właściwości i symulacja pojazdu elektrycznego z napędem na oba koła**)*

Keywords: Electric vehicle, Electronic differential, Permanent magnet DC motor, Yaw rate controller.
Słowa kluczowe: pojazd elektryczny, elektroniczny mechanizm różnicowy, podwójny napęd.

Introduction

The electric vehicle can be motorized in various ways according to the mechanical simplification degree of the chain of traction. In the oldest version, the latter can be made up of a single motor associated with a clutch and a gear box and in the most advanced version, of four driving wheels for the direct drive. In this paper we present an intermediate solution justified by technical and economic considerations. It is presented by a structure of speed control of two identical permanent magnet DC motors each one associated with a fixed ratio gearbox coupled to a rear wheel of the vehicle, thus forming a driving wheel. The two driving wheels are controlled separately by a static converter and their speed is managed by an electronic differential.

Structure with Two Independent Motors

The multi-machine multi-converter structure (MMS) proposed in this work is that of the control of two identical and independent PMDC motors. It is represented in the figure below where each motor is associated with a fixed ratio gearbox coupled to a rear wheel of the vehicle, thus forming a driving wheel. The unit is controlled by a static converter; it is a BUCK chopper type that allows varying the motor speed by adjusting the duty cycle. Both driving wheels are managed by an electronic differential to offset the change in speed in the case of a turn.

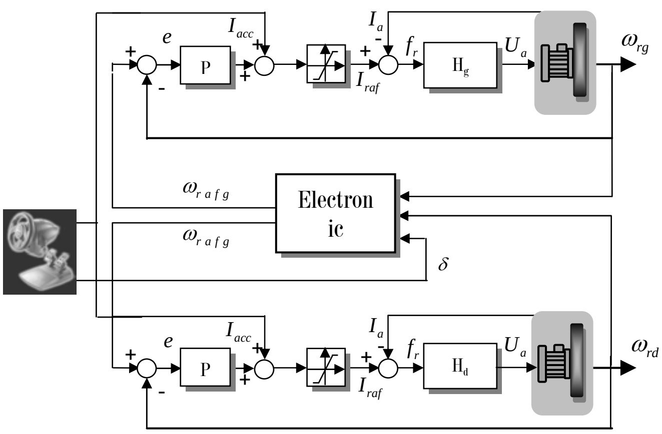


Fig.1 Two driving wheels traction chain structure

Electric Vehicle Modeling

The electric vehicle is a complex system composed of an assembly of several components of various types (mechanics, electric, thermal,…) in interaction. The analysis of the EV as a system then requires a modeling of the various intervening components in order to have a complete model.

Kinematic Model

Generally, any rigid body moving in space is animated by six degrees of freedom, three in translation and three in rotation. The vehicle is thus regarded as a body provided with the frame $R_v = (O, x_v, y_v, z_v)$ in which we associate the inertial reference point of positioning $R_i = (O, x_i, y_i, z_i)$.

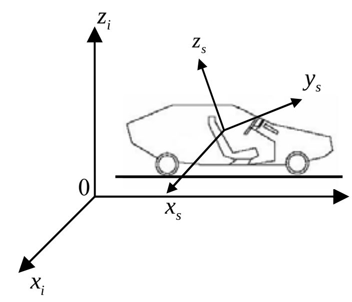


Fig.2 Presentation of the Vehicle in the Cartesian Frame

Orientation Equation

Using the approximation of the bicycle model, the steering angle δ and the radius of curvature R_{CG} of the vehicle path are connected to the distance L between the front axle and rear axle by [06]:

$$\tan\delta = \frac{L_v}{R_{CG}}$$

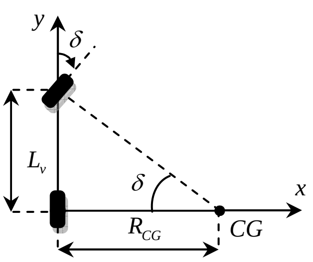


Fig.3 Bicycle model in the inertial frame

Position Equation

We assume that the vehicle performs small uniform rectilinear motions. To develop the simplified equations, we treat two types of motion (translation and rotation) between two moments t and t+Δt. The small angle approximation allows us to express the relationship between δ and the speed of rotation speed, (φ: vehicle orientation) as well as the relationship between the differential of the side coordinate of the center of the rear axle of the vehicle and the lateral deviation φ as follows [02]:

$$\begin{cases} \dot{x} = -V\varphi \\ \dot{\varphi} = \dfrac{V}{L}\delta \end{cases}$$

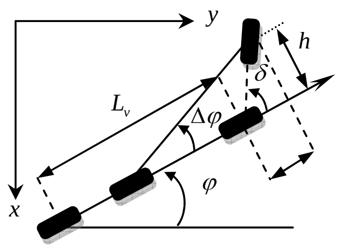


Fig.4 Kinematic model of the vehicle

Dynamic Model

The general formulation of the dynamic equations makes it possible to present the following various rotations:

Pitch (α) **:** q is the rotation speed around x axis.

Roll (β) **:** p is the rotation speed around y axis.

Yaw (φ) **:** r is a rotation around z axis.

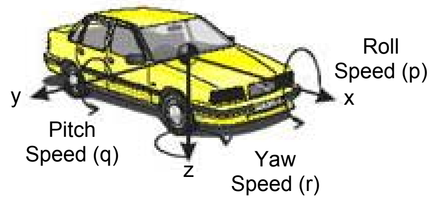


Fig.5 Variables of motion and speeds of the vehicle attached to the axis system

Model of the Efforts Applied to the EV

The vehicle presents a load which is characterized by a certain number of torques.

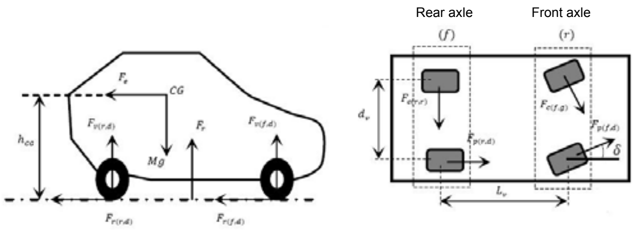
(1) $C_v = C_{rr} + C_{aero} + C_{slope} + C_{acc}$

The figure above illustrates the various forces highlighted for modeling. For this purpose, the aim of the modeling that we seek to work out is to approach as much as possible the real behavior of the electric vehicle. So, the finality of the work is not to release the most precise possible model by integrating the maximum of parameters, but to have a satisfactory model by its precision and its complexity. For this purpose, certain more or less exact models are necessary. These concern the kinematics and the dynamics of the vehicle [06].

- $F_{r(i,j)}$: Bearing resistance forces,
- $F_{v(i,j)}$: Vertical reaction forces,
- $F_{c(i,j)}$: Centripetal forces,
- $F_{p(i,j)}$: Propulsion Forces,

with: $(i = r, f \text{ and } j = g, d)$

- F_e : External resultant forces following y,
- F_r : External resultant forces following z,



(a) (b)

Fig.6 Lateral view (a) from below (b) different forces applied to vehicle

Bearing resistance torque: Corresponds to the resistance against the deformations of the tire due to the contact wheel-ground, the vehicle displacement speed, tire pressure, the nature of the ground and the condition of the bandage.

(2) $F_{rr} = \left[C_{rr_{av}} M_{av} + C_{rr_{ar}} M_{ar} \right] \approx C_{rr} Mg$

Where $M_{av}, M_{ar}, C_{rr_{av}}$ and $C_{rr_{ar}}$ are the masses and the torques carried respectively by the front and rear wheels. Thus the bearing torque is:

(3) $C_{rr} = Mg\, F_{adh}\, R_r$

Aerodynamic torque: Is the torque corresponding to a set of forces exerted on the vehicle body and which resists to the penetration into the air. These forces are:

- F_x : The trail force. It opposes the propulsion of the vehicle.
- F_y : The lift force. Tends to raise the vehicle.
- F_z : The drift force. Tends to tilt the car.

(4) $F_{aero} = \frac{1}{2}\rho_{air}\, SC_x\, (V_{veh} - V_{wind})^2$

(5) $V = (V_{veh} - V_{wind})^2$

Thus the aerodynamic torque exerted on the vehicle is given by:

(6) $C_{aero} = \frac{1}{2}\rho_{air}\, SC_x\, R_r^3 \omega_r^2$

Torque related to the slope: When the vehicle moves on an inclined level, its weight breaks up into two components, normal and tangential. Its torque is given by:

(7) $C_{slope} = Mg\, R_r \sin\alpha_p$

Force related to acceleration: F_{acc} represents the dynamic term of acceleration or deceleration of the vehicle.

(8) $F_{acc} = M\dfrac{dV_v}{dt} = M\,\gamma$

Finally, the total effort of resistance to the advance of the vehicle is:

(9) $F_v = M\left[g(C_{rr} + \sin\alpha_p) + \gamma \right] + \frac{1}{2}\rho_{air}\, SC_x V^2$

Modeling of the Chain of Traction

The design of an electric drive system is a rather demanding task. It represents a set of elements designed to achieve a technological process, consisting of an electric motor powered by a voltage source through a static converter, a movement mechanical converter and a system of control and regulation. Each motor is equipped with a fixed ratio speed reducer and attached to the wheel constituting a driving wheel [04].

The mechanical equation of the unit is given by the following expression [03]:

(10) $$J\frac{d\omega_m}{dt}+f_r\,\omega_m=C_m-C_r$$

with:

(11) $$J_e=J_m+\frac{J_{veh}}{\eta N_{red}^2}$$ and $$C_r=\frac{1}{\eta N_{red}}C_{veh}$$

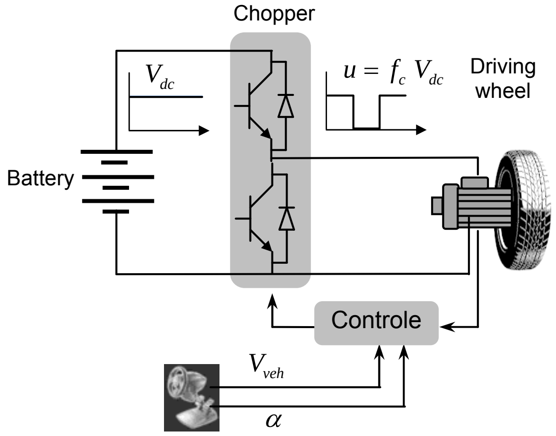


Fig.7 Control structure of the traction chain of a driving wheel with buck chopper

Modeling of the Static converter

The static converter is an element of connection between the motor and the source. The choice of its structure rises from the used motor. It is selected such as the maximum power requested by the motor is covered by 150% of the rated output power of the converter and according to standard ICS-3-301 NEMA (National Electrical Manufactures Association), it must support 150% of the rated current for one minute during (accelerations-decelerations) [05]. The converter used is a reversible current BUCK step-down chopper controlled using the comparison of a sawtooth reference signal having period equal to the period of chopping and given amplitude V_{max} with a control voltage V_c. The point of intersection of the two signals imposes the duration of the conduction of the chopper.

(12) $$V_{ds}=\hat{V}_{ds}\frac{t}{T}$$

(13) $$f_c=\frac{T_c}{T}=\frac{V_{ref}}{\hat{V}_{ds}}$$

(14) $$U_{av}=f_c\,V_{dc}=\frac{V_{ref}}{\hat{V}_{ds}}V_{dc}$$

(15) $$H_o=\frac{V_{dc}}{\hat{V}_{ds}}$$

- V_{ds} : Sawtooth signal
- T_c : Duration of conduction
- U_{av} : Average load voltage

Actually it is noted that there is a delay $0\prec\tau\prec T_c$ between the control moment and its effect on the converter. By introducing the average delay $\tau_0=T_c/2$ the transfer function of the chopper is given by the following expression [03]:

(16) $$H_h(s)=H_o e^{-\tau_o t}$$

After approximation one obtains the following relation:

(17) $$H_h(s)\approx\frac{H_o}{1+\tau_o\,s}$$

Modeling of the PMDCM

The electric motor is the fundamental structural element of any electric drive system. It is chosen on the basis of calculation of the load power P_{ch} required by the driven mechanism so that the motor rated power P_n = 1.1 to 1.3 P_{ch}. Concerning heating effect, a going beyond the limiting temperature by 10°C involves a reduction of 50% of the motor lifespan. In many cases, the checking of the criterion of heating of the motor can be done by the torque:

(18) $$C_n>C_{eq}$$

with

- C_n : Rated torque of the motor
- C_{eq} : Equivalent torque required by the load

(19) $$C_{eq}=\sqrt{\frac{1}{t_{cy}}\sum_{i=1}^{n}C_i^2}$$

The overload criterion ensures that the stall torque developed by the motor exceeds the torque required by the load.

(20) $$C_{dec}>C_{\max}$$

According to the motion equation, the required torque is obtained by the following expression:

(21) $$C(t)=C_r(t)+J_{\sum j}\frac{d\omega_m}{dt}$$

with:

- $J_{\sum j}$: The total rotational inertia

Thus the dynamic model of the PMDCM is defined by:

(22) $$U_a=R_aI_a+L\frac{di_a}{dt}+k_\varphi\omega$$

(23) $$C=k_\varphi i_a=J_m\frac{d\omega}{dt}+f\,\omega+C_r$$

(24) $$\omega_m=\frac{1}{1+s\tau_{em}+s^2\tau_e\tau_{em}}\left[\frac{U_a}{k_\varphi}-\frac{\tau_{em}(1+s\tau_e)C_r}{J_m}\right]$$

with:

- $\tau_e=\frac{L_a}{R_a}$: Electric time-constant
- $\tau_{em}=\frac{R_aJ}{R_a+k_\varphi^2}$: Electromagnetic time-constant

From the electrical and mechanical equations of the permanent magnet DC motor the state equation model is presented as follows [1]:

(25) $$\dot{x}=Ax+Bu$$

(26) $$\begin{pmatrix}\frac{di_a}{dt}\\ \frac{d\omega}{dt}\end{pmatrix}=\begin{pmatrix}-\frac{R_a}{L_a} & -\frac{k_\varphi}{L_a}\\ -\frac{k_\varphi}{L_aJ} & 0\end{pmatrix}\begin{pmatrix}i_a\\ \omega\end{pmatrix}+\begin{pmatrix}\frac{1}{L_a} & 0\\ 0 & -\frac{1}{J}\end{pmatrix}\cdot\begin{pmatrix}U_a\\ C_r\end{pmatrix}$$

The motor speed control is done by a PI controller. The chopper is controlled by a logical function that determines its duty cycle $f_x=I_{ref}-I_a$

(27) $$I_{ref}=\left(k_p+\frac{k_I}{p}\right)\cdot(\omega_{ref}-\omega_m)+I_{acc}$$

(28) $$\dot{f}_x=k_p\cdot(\dot{\omega}_{ref}-\dot{\omega}_{rm})+k_I\cdot(\omega_{ref}-\omega_{rm})-\frac{dI}{dt}$$

For $\dot{f}_x = 0$ and by replacing in the model (30) one obtains the following total state equation:

$$(29)\quad \begin{pmatrix} \frac{di_a}{dt} \\ \frac{d\omega_m}{dt} \end{pmatrix} = \begin{pmatrix} -\frac{k_p k_\varphi \cdot \varphi}{J} & -k_i \\ \frac{k_\varphi \cdot \varphi}{J} & 0 \end{pmatrix} \cdot \begin{pmatrix} i_a \\ \omega_m \end{pmatrix} + \begin{pmatrix} k_p & k_I & \frac{k_p}{J} \\ 0 & 0 & -\frac{1}{J} \end{pmatrix} \cdot \begin{pmatrix} \frac{d\omega_{ref}}{dt} \\ \omega_{ref} \\ C_r \end{pmatrix}$$

The Electronic Differential

Behavior of the vehicle when turning

We will consider that the vehicle is in a perfectly circular turn allowing it to change its direction of rolling by 90°.

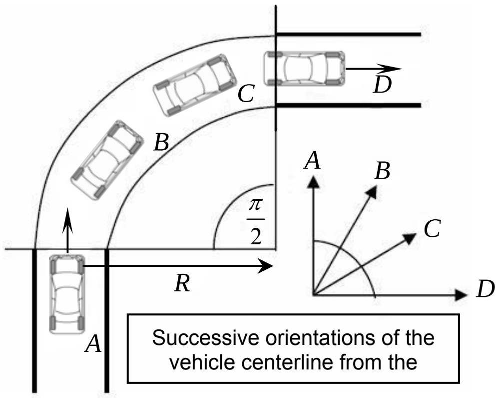


Fig.8 Yaw rate of the vehicle when taking a turn

As the yaw angle is changing in the same way as the angle swept in the turn by the vehicle we have the yaw rate. This speed is zero in both states A and D because the vehicle is in a straight line situation; in addition it is constant during the two states B and C since the vehicle is there in situation of turn. To move from state A to state B, the vehicle will have to undergo an acceleration of yaw in entrance of turn and conversely it will need a deceleration of yaw at exit of turn.

At the entrance of the turn the driver operates the steering wheel. The front wheels change direction while the vehicle is still on a straight line. This puts the front tires in situation of drift and they produce guide efforts directed towards the interior of the turn. The rear wheels are zero drift since the vehicle has not started its yaw motion. The rear guide efforts are null. During the transitional zone the vehicle is rotated which is gradually drifting the rear wheels.

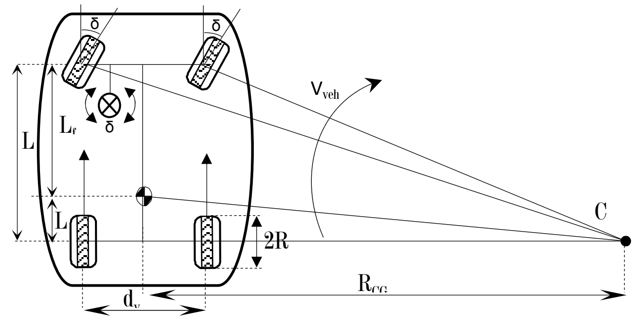


Fig.9 Vehicle structure in a turn

The model of the electronic differential

In the case of a straight line, each driving wheel is controlled by a static converter with the same speed reference. But in a turn this speed is different because the two wheels will not have the same speed. This difference is managed by an electronic differential that allows us to calculate the difference in speed in order to be able to apply it in the control. As the yaw angle is changing in the same way as the angle

$$(30)\quad \begin{cases} V_{rd} = V_{veh} + \Delta V_{veh} \\ V_{rg} = V_{veh} - \Delta V_{veh} \end{cases}$$

- V_{rd}, V_{rg} : are respectively the linear velocity of the right and left wheel.
- V_{veh} : linear velocity of the vehicle.
- ΔV_{veh} : the difference in speed between two driving wheels

Thus:

$$(31)\quad \begin{cases} V_{rd} = \omega_{veh}(R_{CG} + \frac{d_v}{2}) \\ V_{rg} = \omega_{veh}(R_{CG} - \frac{d_v}{2}) \end{cases}$$

$$(32)\quad R_{CG} = \frac{L_v}{\tan\delta}$$

Then the angular velocity of each driving wheel is given by the following expression:

$$31)\quad \begin{cases} \omega_{rd} = \dfrac{L_v - \frac{d_v}{2}\tan\delta}{L_v}\omega_{veh} \\ \omega_{rg} = \dfrac{L_v + \frac{d_v}{2}\tan\delta}{L_v}\omega_{veh} \end{cases}$$

$$(32)\quad \Delta\omega = \omega_{rg} - \omega_{rd} = \frac{d_v \tan\delta}{L_v}\omega_{veh}$$

with
- $\delta > 0$: for right turn
- $\delta = 0$: for straight line
- $\delta < 0$: for left turn

Table 1. Geometrical parameters of the vehicle

M_v	Vehicle mass	500kg
L_v	Wheelbase	2,2 m
d_v	Distance between the driving wheels	1,3 m
R_r	Radius of the wheel	0,23 m

Simulation Results

The simulation model of the vehicle is presented for two cases of different applied loads, the first case determines the behavior of vehicle dynamics under the influence of the variation in speed in a straight line (acceleration and deceleration), while the second case is used to visualize the behavior of the vehicle during a double turn on land while enhancing the operation of the electronic differential.

Case 1: The vehicle dynamics in straight line

On flat ground in a straight line, the system is subjected to a speed step of 20km/h due to the action on the accelerator by the driver in order to defeat the forces of inertia (figure.12). At t = 20sec the driver doubles its speed by pressing more on the accelerator (figure.15). In both situations, the two driving wheels rotate in the same angular velocity so that the speed variation developed in the electronic differential is zero. To perform a shutdown at t = 40 sec, the two driving wheels run as a generator under the influence of the inertia of the vehicle so that energy is removed by dissipation.

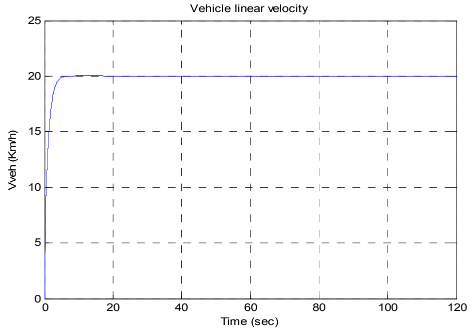


Fig.10 Linear velocity of the vehicle in straight line 20km/h

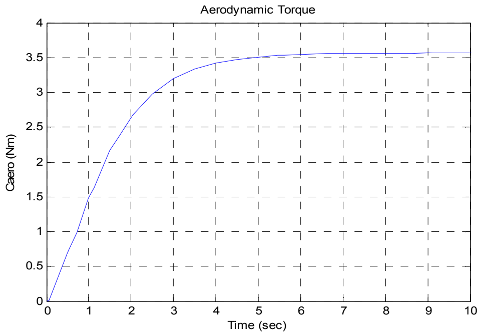


Fig.11 Aerodynamic torque V_{veh}=20km/h

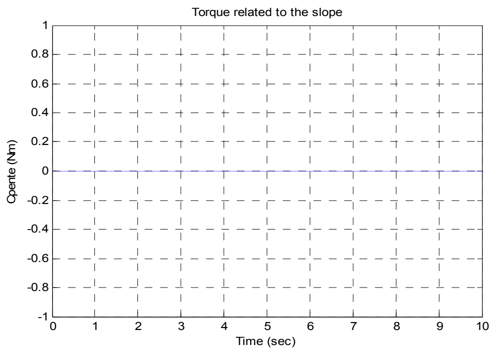


Fig.12
Torque related to the slope

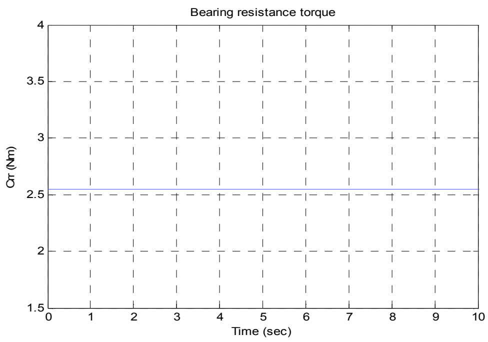


Fig.13 bearing resistance torque

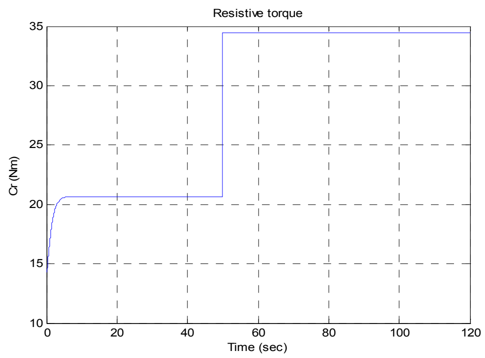


Fig.14 Resistive torque to overcome for traction

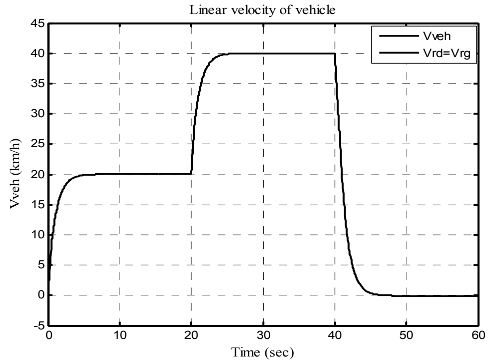


Fig.15
Llinear velocity of the vehicle with change of consign 20km/h and 40km/h

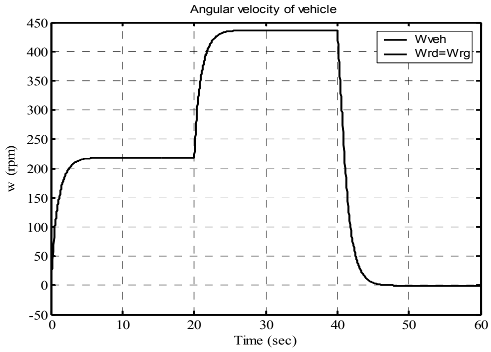


Fig.16 Angular velocity of the vehicle with change of consign

Case 2: Vehicle dynamics in a turn

The vehicle engages in a double turn (on the right then on the left) with a speed of 20km/h (figure.17).

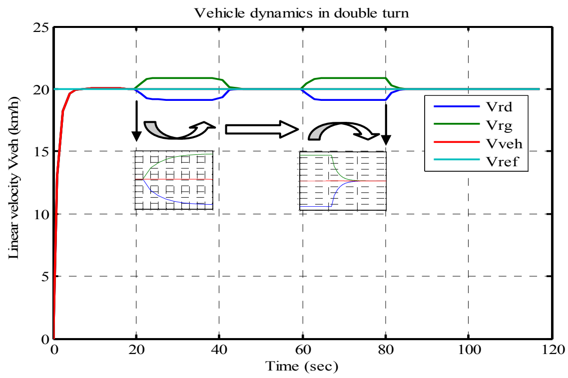


Fig.17 Vehicle dynamics in double turn

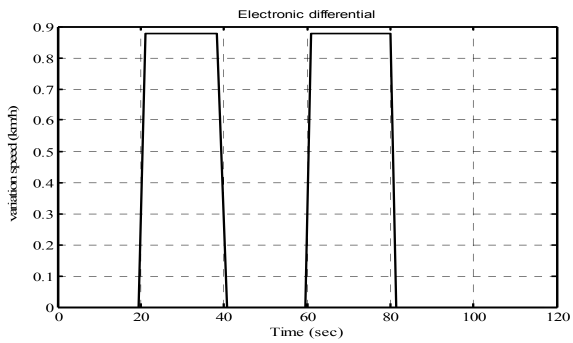


Fig.18 Variation speed on the level of the differential

In this case, the speed of the driving wheel which is closer to the center of turn is lower than that of the most distant. This means that their yaw rate is constant (condition B and C – figure.8) and the electronic differential has compensated the variation of speed (figure.18).

Conclusion

The interest of this work is to present the structure of EV in the Cartesian coordinate system and give the model of the various parts which constitutes it (kinematic and dynamic). As it allows studying a propulsion topology based on an intermediate motorization constituted by of a chain of traction to two driving wheels defined by a multi machine multi converter system. The motor control is provided by two buck choppers and a PI controller. The vehicle behavior is represented by two cases, in straight line and turn, where we toured the dynamics of the yaw rate of the vehicle when turning. The simulation results show that this structure allows producing the electronic differential which controls the rotational speed of the driving wheels with high precision in order to pass the turn as efficiently as possible. Considered essential, it is part of the system equipment thus ensuring stability and good stability of the vehicle.

As perspective, this work may be subject to practical implementation for specific applications such as expanded spaces, enclosed or closed (airport, air terminal, military area, golf course ... etc) for a linear speed not exceeding the 50km / h and carrying two people.

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