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Research Article

Longitudinal and Lateral Tire Road Forces Estimation for Electric Vehicle with Four in-wheel Motors

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Abstract: This study introduces an estimation process for longitudinal and lateral tire road forces, for that an electric vehicle with four in-wheel motors has been implemented. This structure permits to control independently each one wheel so that deals to torque's control with fast response which represent the best advantage in this kind of vehicles with PMSM which take a dominant position in several systems with variable speed drive because of their, low inertia, high efficiency and no maintenance. The performance of this concept is tested and demonstrated using Matlab/Simulink. The simulation results show that the proposed approach is a promising technique to provide accurate estimations of vehicle dynamic states.

Keywords: Electric vehicle technology, in-wheel motor, longitudinal control, lateral and longitudinal motion, tire forces, vehicle dynamic

INTRODUCTION

Nowadays, the environmental problems such as global warming, exhaustion of fossil fuels and air pollution are getting serious. Therefore, Electric Vehicles (EVs) have attracted a great deal of interest as zero-emission vehicle. In addition, EVs have following three advantages:

- Development of in-wheel motors enables individual control of each wheel
- Generated torque can be measured precisely from motor current.
- Torque response is quick.

These advantages are effective for vehicle motion control. However, the motion is governed by the forces generated between the tires and the road, For example, lateral tire force (also known as side or cornering force) is the necessary force to hold a vehicle through a turn. It is generated by the lateral tire deformation in the contact patch (Doumiati *et al*., 2009). In fact, what happens is that when the front wheels of a vehicle are steered, a slip angle is created, which gives rise to a lateral force, this force is usually represented in function of its sideslip angle. The lateral tire road force is a nonlinear function of the tire slip angle. Under normal driving situations (low slip angle), a vehicle responds predictably to the driver's inputs. As the vehicle approaches the handling limits, for example, during an evasive emergency maneuver, or when a vehicle undergoes high accelerations, high slip angle occurs and the vehicle's dynamic becomes highly nonlinear and its response becomes less predictable and potentially very dangerous where providing safety is very important and needs knowledge and accuracy of vehicle's dynamic.

The accurate and reliable information about the state of the vehicle and these forces is very important (Linhui *et al*., 2008), it leads to a better evaluation of the road friction and the vehicle's possible trajectories and to a better vehicle control. Moreover, it makes possible the development of a diagnostic tool for evaluating the potential risks of accidents related to poor adherence or dangerous maneuvers (Huang and Wang, 2013). The safety control systems like ESP and

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TCS function well only when they are known (Rabhi *et al*., 2007).

Because of these forces are unmeasured for economical reasons, the use of model-based estimation techniques is therefore necessary to provide their estimation. Recently, many analytical and experimental studies on the estimation of the longitudinal/lateral tire force and the friction coefficient between the tire and the road have been performed.

This study presents an estimation strategy to estimate the longitudinal/lateral tire forces. The longitudinal/lateral tire-force estimator consists of three steps: longitudinal and lateral velocities estimation, four longitudinal slips and sideslip angle estimation and longitudinal and lateral tire forces estimation. The performance of the proposed estimators has been evaluated via Matlab/Simulink.

STRUCTURE OF THE PROPOSED MODEL

The model's vehicle is a four in wheel motors vehicle where each motor is fed by a two level inverter, the motors chosen are PMSM permanent magnet synchronous machines which take a dominant position in several systems with variable speed drive because of their, low inertia, high efficiency and no maintenance (Yantour *et al*., 2006). The four motors are controlled by a DTC control (Hartani *et al*., 2014). The direct torque and flux control has been introduced by I. TAKAHASHI in 1985 from the flux-oriented method and the principle of the DC motor. A selection table is used to determinate the control sequence that should be applied to the voltage inverter switches, such as the torque and flux errors are kept within the specified bands.

DYNAMIC MODEL

The dynamic motion of the vehicle is modeled by three equations that represent respectively the longitudinal and lateral translational motion and the yaw rotational movement:

$$\begin{cases} M_v \ddot{X} = F_X \\ M_v \ddot{Y} = F_Y \\ J_z \ddot{\psi} = M_z \end{cases} \tag{1}$$

F_X F_X, F_Y and M_Z are respectively the total forces acting in X and Y directions and the total yawing moment. They are clearly linked with the velocities in reference frame by the obvious relationship (Hartani *et al*., 2010):

$$\begin{pmatrix} \dot{X} \\ \dot{Y} \\ \dot{\psi} \end{pmatrix} = \begin{pmatrix} \cos(\psi) & -\sin(\psi) & 0 \\ \sin(\psi) & \cos(\psi) & 0 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} v_x \\ v_y \\ r \end{pmatrix} \tag{2}$$

Although there are many ways to obtain the mathematical model, a procedure based on Lagrange equations will be followed here; neglecting the relational kinetic energy of the wheels the kinetic energy of the vehicle is as follows:

$$T = 0.5.M_v \left(\dot{X}^2 + \dot{Y}^2 \right) + 0.5.J_v \dot{\psi}^2 \tag{3}$$

And the equations of motion can be obtained by the relation:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} = Q_i \tag{4}$$

where, the coordinates q_i are X ,Y and ψ and Q_i are the corresponding forces F_X , F_Y and M_Z . Performing the relevant derivatives with respect to time and introducing forces F_X and F_Y written with reference to vehicle frame, we obtain:

$$\begin{aligned} \frac{F_X}{M_v} &= (\dot{v}_x - r v_y) \\ \frac{F_Y}{M_v} &= (\dot{v}_y + r v_x) \\ \frac{M_Z}{J_v} &= \dot{r} \end{aligned} \tag{5}$$

Figure 1 shows a schematic diagram of a vehicle model, which has three Degrees-of-Freedom (DOF) for longitudinal motion, lateral motion and yaw motion of the vehicle. A body-fixed coordinate system with the origin at the vehicle Center of Gravity (COG) is used to set up the model. According to Fig. 1, the kinematic relationships among the vehicle velocity, yaw rate and acceleration are as follows:

$$\begin{aligned} M_v(\dot{v}_x - r v_y) &= \left(F_{x1} + F_{x2}\right).\cos(\delta) \\ &- \left(F_{y1} + F_{y2}\right).\sin(\delta) + F_{x3} + F_{x4} - F_{res} \end{aligned} \tag{6}$$

$$\begin{aligned} M_v(\dot{v}_y + r v_x) &= \left(F_{x1} + F_{x2}\right).\sin(\delta) - \left(F_{y1} + F_{y2}\right). \\ &\cos(\delta) + F_{y3} + F_{y4} \end{aligned} \tag{7}$$

$$\begin{aligned} J_v \dot{r} &= l_F\left(F_{x1} + F_{x2}\right).\sin(\delta) + l_F\left(F_{y1} + F_{y2}\right).\cos(\delta) \\ &- l_R\left(F_{y3} + F_{y4}\right) - \frac{b_F}{2}\left(F_{x1} - F_{x2}\right) \end{aligned} \tag{8}$$

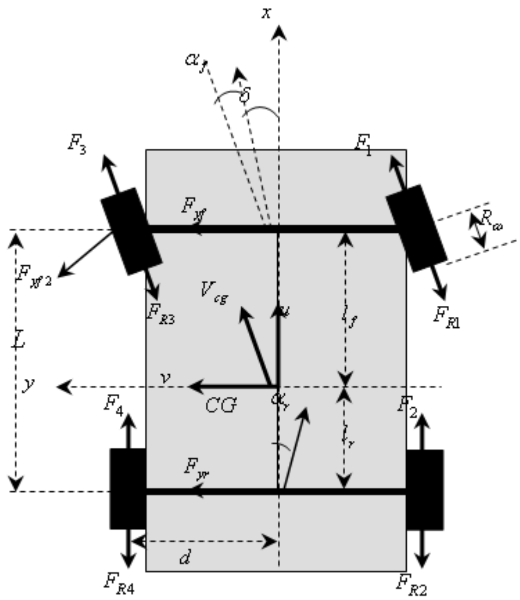


Fig. 1: Schematic diagram of three DOF vehicle model

F_{res} is the resistant forces which including the aerodynamic drag force F_{aero}, climbing force F_c and rolling force F_{rr}. They have the following expressions,

$$F_{aero} = 0.5\rho S_{px} C_f v_x^2$$
$$F_c = M_v g \sin(\alpha_p)$$
$$F_{rr} = C_{rr} M_v g \tag{9}$$

TIRE MODEL

The tires, which generate longitudinal, lateral forces and moments, have a significant effect on the dynamic characteristics of vehicles. These tire forces are explained by a complex relation among tire–road friction, normal force on the tire, variable slip angles and elastic tire properties. To model tire force generation, several tire models have been developed. A widely used empirical tire model (Magic formula tire model) which was used in Hsiao *et al*. (2011) and Matuško *et al*. (2008) is dominantly based on empirical formulations deriving from tire test data and a large number of tire parameters (Nam *et al*., 2012).

The nonlinearities of vehicle tires will become a critical factor during emergency maneuvers in which the linear tire model is not sufficiently accurate anymore. To account for the nonlinearities, Dugoff's tire model is used to model individual tire forces along the longitudinal and lateral axes. Towards controller derivation and analysis, Dugoff's model may be analytically derived at controller's development stage and the longitudinal friction force and lateral friction force of the wheel (Zhao *et al*., 2011).

The longitudinal driving force and the lateral force can be determined by the following formula as (Acarman, 2008):

$$F_{xi} = \frac{C_x \lambda_i}{1+\lambda_i} f(\sigma_i) \tag{10}$$

$$F_{yi} = \frac{C_y \tan\alpha_i}{1+\lambda_i} f(\sigma_i) \tag{11}$$

$$\sigma_i = \frac{\mu F_{zi}(1+\lambda_i)}{2.\sqrt{(C_x\lambda_i)^2 + (C_y \tan\alpha_i)^2}}$$
$$f(\sigma_i) = \begin{cases} 1 & \sigma_i \ge 1 \\ (2-\sigma_i)\sigma_i & \sigma_i < 1 \end{cases} \tag{12}$$

It is necessary to consider load transfer in virtue of longitudinal and lateral acceleration of vehicle for analyzing the vehicle behavior of braking and driving situation. The lateral wind and road gradient are neglected, so the vertical load of each wheel must be calculated (Feiqiang *et al*., 2008). The normal force for each wheel can be calculated as:

$$F_{z1} = \frac{M_v(L_R g - h a_x)}{2(L_F + L_R)} - \frac{M_v h a_y (L_R g - h a_x)}{(L_F + L_R) b_F g}$$
$$F_{z2} = \frac{M_v(L_R g - h a_x)}{2(L_F + L_R)} + \frac{M_v h a_y (L_R g - h a_x)}{(L_F + L_R) b_F g}$$
$$F_{z3} = \frac{M_v(L_F g + h a_x)}{2(L_F + L_R)} - \frac{M_v h a_y (L_F g - h a_x)}{(L_F + L_R) b_R g}$$
$$F_{z4} = \frac{M_v(L_F g - h a_x)}{2(L_F + L_R)} + \frac{M_v h a_y (L_F g - h a_x)}{(L_F + L_R) b_r g} \tag{13}$$

The sideslip angles of the wheels are calculated based on geometric derivation using wheel velocity vectors and the steering angle δ (Nam *et al*., 2013):

$$\alpha_1 = \delta - \tan^{-1}\left(\frac{v_y + L_F r}{v_x - \frac{b_F}{2} r}\right)$$
$$\alpha_2 = \delta - \tan^{-1}\left(\frac{v_y + L_F r}{v_x + \frac{b_F}{2} r}\right)$$
$$\alpha_3 = -\tan^{-1}\left(\frac{v_y - L_R r}{v_x - \frac{b_R}{2} r}\right)$$
$$\alpha_4 = -\tan^{-1}\left(\frac{v_y - L_R r}{v_x + \frac{b_R}{2} r}\right) \tag{14}$$

By applying a driving force to a tire which has a certain slip (λ), the longitudinal coefficient (μ) is influenced by the wheel sideslip angle. Moreover, the longitudinal slip can be defined for the four wheels as:

$$\lambda_i = \frac{R\omega_i - ut_i}{\max(R\omega_i, ut_i)} \quad i = [1....4] \tag{15}$$

where, R is the wheel radius, ω_i is the angular velocity of the in-wheel motor and ut_i is the linear speed at which the contact zone moves on the ground.

TIRE FORCES ESTIMATION

Estimation of tire friction forces becomes an active research topic recently. A variety of estimation techniques have been proposed in the literature. Most of them are based on either tire models or vehicle models. The simplest longitudinal tire model assumes that the longitudinal tire force is linearly proportional to the tire slip ratio, which is the relative difference between the translational velocity and the angular velocity of a wheel.

Recently, many analytical and experimental studies on the estimation of the longitudinal/lateral tire force between the tire and the road have been performed.

An Extended Kalman Filter (EKF) has been implemented to estimate the state and the longitudinal and lateral tire-force histories of a 9-DOF vehicle, a nonlinear Kalman Filter in Lee *et al*. (2012). This estimation process shows good robustness properties, even in the face of abrupt changes in road conditions (Cho *et al*., 2010). The Unscented Kalman Filter (UKF), (Cheng *et al*., 2011), is an alternative to the EKF for its flexibility. In Matuško *et al*. (2008), a new neural network based estimation scheme was proposed, which makes friction force estimation insensitive to modelling inaccuracies. Lateral tire forces, the sideslip angle and tire–road friction have been estimated by an adaptive observer that uses a combination of a vehicle model and a tire-force model (Baffet *et al*., 2007). In (Rabhi *et al*., 2007; Vasiljevic *et al*., 2012), the longitudinal tire force of each wheel was estimated using the moment balance equation of each wheel, where in Linhui *et al*. (2008) a sliding mode observer was implemented for their estimation. Baffet *et al*. (2007) and Huang and Wang (2013) proposed a sliding mode observer to estimate the longitudinal and lateral forces simultaneously based on a simplified vehicle model; however the rear tires were assumed in pure rolling and hence their longitudinal tire forces were neglected. Besides, only the sums of the front and rear lateral tire forces can be obtained. In this study, the longitudinal and lateral forces estimation is proposed on an observer for dynamic states estimation.

Longitudinal velocity estimation: When estimating longitudinal velocity, we use a model including only:

$$\dot{v}_x = a_x \tag{16}$$

The wheel speed measurements ω_i are transformed to measurements of v_x, assuming zero slips. Using $v_{x,i} = R_{dyn}\omega_i \cos\delta_i$ and $v_{i,x} = v_x \pm b_i r$, we get the following expression for the transformed measurement from wheel i :

$$v_{x,i} = R_{dyn}\omega_i \cos\delta_i \pm b_i r \tag{17}$$

Using these, we propose the following observer:

$$\hat{v}_x = a_x + \sum_{i=1}^{4} K_i(a_x, \omega)\left(v_{x,i} - \hat{v}_x\right) \tag{18}$$

The observer gains K_i depend on the longitudinal acceleration and the wheel speed measurements (and possibly other information), to reflect when ω_i are expected to provide good estimates of v_x, that is, when the tire slips are low (Imsland *et al*., 2006; Grip *et al*., 2009).

Lateral velocity estimation: For the lateral velocity, we start by introducing $\hat{a}_y\left(t, \hat{v}_x, \hat{v}_y\right)$, which denotes an estimate of the lateral acceleration a_y. The estimate $\hat{a}_y\left(t, \hat{v}_x, \hat{v}_y\right)$ is formed by using the nonlinear friction model for each wheel, where the measurements of the steering wheel angle, yaw rate and wheel speeds, as well as the estimated velocities $\hat{v}_x$ and $\hat{v}_y$ are used as inputs. The friction forces modeled for each wheel are added up in the lateral direction of the vehicle and divided by the mass, resulting in the lateral-acceleration estimate $\hat{a}_y\left(t, \hat{v}_x, \hat{v}_y\right)$. The t in $\hat{a}_y\left(t, \hat{v}_x, \hat{v}_y\right)$ denotes the dependence of $\hat{a}_y$ on time-varying signals such as the steering wheel angle (Grip *et al*., 2009). We also write:

$$\tilde{a}_y(t, \tilde{v}_x, \tilde{v}_y) := a_y - \hat{a}_y(t, \hat{v}_x, \hat{v}_y) \tag{19}$$

The lateral-velocity estimate is given by:

$$\dot{\hat{v}}_y = a_y - v_x r + K_{vy}\left(a_y - \hat{a}_y\left(t, \hat{v}_x, \hat{v}_y\right)\right) \tag{20}$$

ESTIMATION PROCESS

We proposed an estimation method based on dynamic states estimation v_x and v_y as were mentioned above, since longitudinal and lateral forces are expressed according to longitudinal and lateral slips, as much as these variables are the main factor for forces generation, we have estimated $\hat{\lambda}_i = 1..4$ and $\hat{\alpha}_i$ and from Eq. (14) and (15) estimated tires slips and side slips will be given respectively by:

$$\hat{\lambda}_i = \frac{R\omega_i - \hat{ut_i}}{\max\left(R\omega_i, \hat{ut_i}\right)} \quad i = [1...4] \tag{21}$$

$$\hat{\alpha}_1 = \delta - \tan^{-1}\left(\frac{\hat{v}_y + L_F r}{\hat{v}_x - \frac{b_F}{2} r}\right)$$

$$\hat{\alpha}_2 = \delta - \tan^{-1}\left(\frac{\hat{v}_y + L_F r}{\hat{v}_x + \frac{b_F}{2} r}\right)$$

$$\hat{\alpha}_3 = -\tan^{-1}\left(\frac{\hat{v}_y - L_R r}{\hat{v}_x - \frac{b_R}{2} r}\right)$$

$$\hat{\alpha}_4 = -\tan^{-1}\left(\frac{\hat{v}_y - L_R r}{\hat{v}_x + \frac{b_R}{2} r}\right) \tag{22}$$

From Eq. (10) and (11) estimated forces are given by:

$$\hat{F}_{xi} = \frac{C_x \hat{\lambda}_i}{1 + \hat{\lambda}_i} \hat{f}(\sigma_i) \tag{23}$$

$$\hat{F}_{yi} = \frac{C_y \tan \hat{\alpha}_i}{1 + \hat{\lambda}_i} \hat{f}(\sigma_i) \tag{24}$$

SIMULATION AND DISCUSSION

The four in wheel electric vehicle model on Fig. 2 is constructed in Matlab/simulink. In order to test the performance of the proposed approach the vehicle's front wheels are steered at left then at right with a steering angle given in Fig. 3. The presented approach is performing to estimate the states and the individually generated tire force. Simulation results are given according to specification of vehicle, Table 1 with specifications PMSM, (Table 2). Accouting aerodynamic drag force F_{aero} and rolling force F_{rr},

Table 1: The specifications of the vehicle used in simulation

Constants	Designation	Values
M_v	Vehicle mass	1562 kg
J_v	Vehicle inertia	2630 kg.m²
J_ω	Wheel inertia	1, 284kg.m²
L_F	Distance from the CG to front axle	1, 104 m
L_R	Distance from the CG to rear axle	1, 421 m
h	Heigh of the vehicle ctroid (CG)	0, 5 m
S_f	Frontal area of vehicle	2, 04 m²
ρ	Air density	1, 2 kg.m^{-3}
C_{px}	Drag coefficient	0, 25
C_{rr}	Rolling resistance coefficient	0, 01
C_x	Longitudinal stiffness of each tire lateral	37407N/rad
C_y	Lateral stiffness of each tire lateral	51918N/rad
R_ω	Wheel radius	0, 294 m
b	A look ahead distance	5 m

Table 2: The specifications of motors

Constants	Designation	Values
R_s	Resistance	0,03 Ω
L_s	Inductance	0,2 mH
Φ_f	Permanent magnet flux	0,08Wb
p	Pole pairs	4

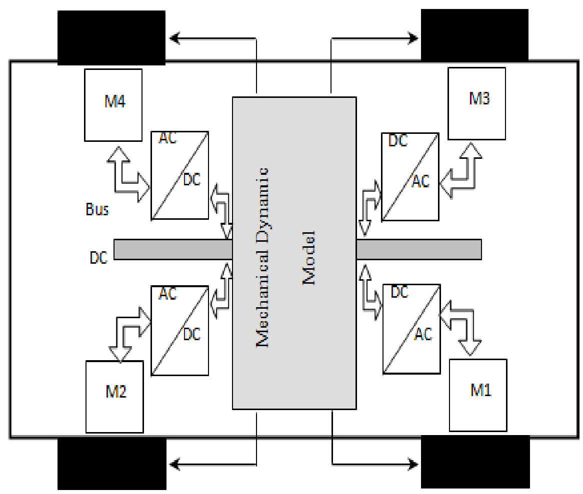


Fig. 2: Studied structure of electric vehicle

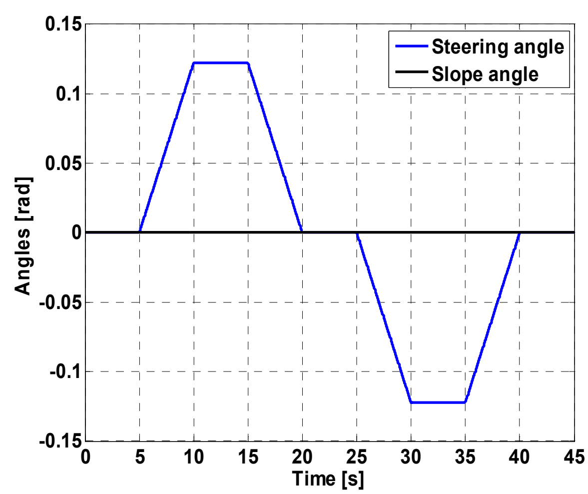


Fig. 3: Steering angle and slope angle

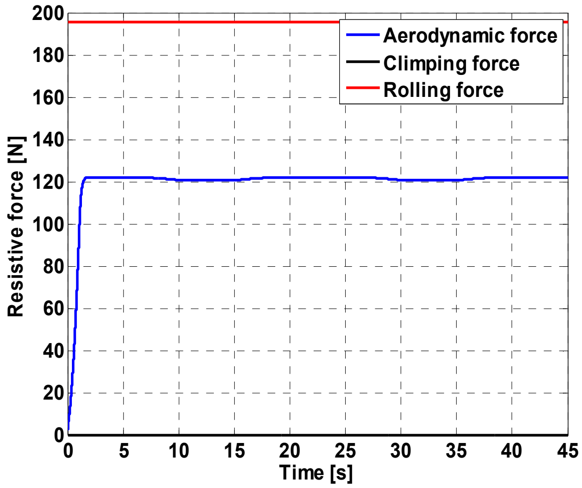
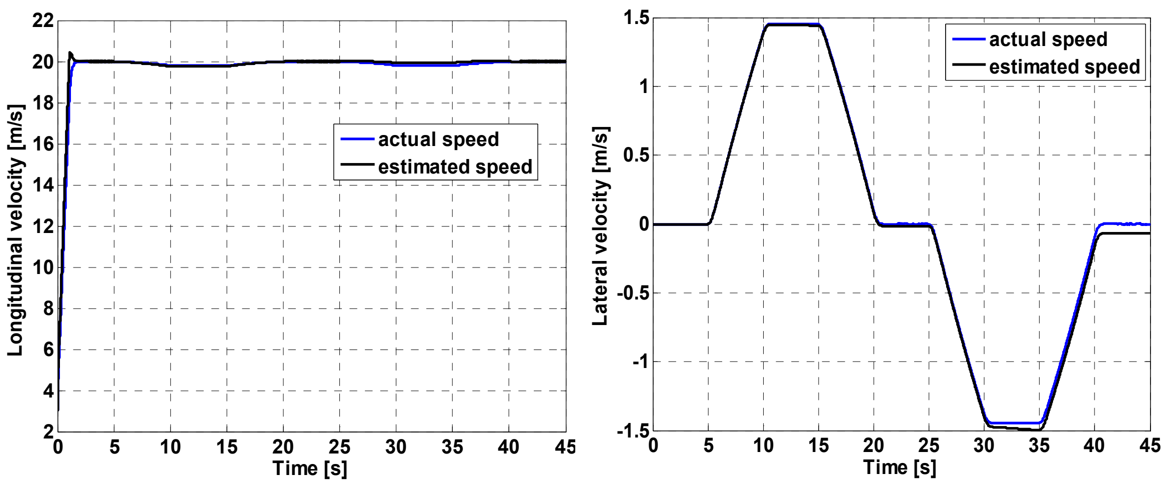


Fig. 4: Resistive forces

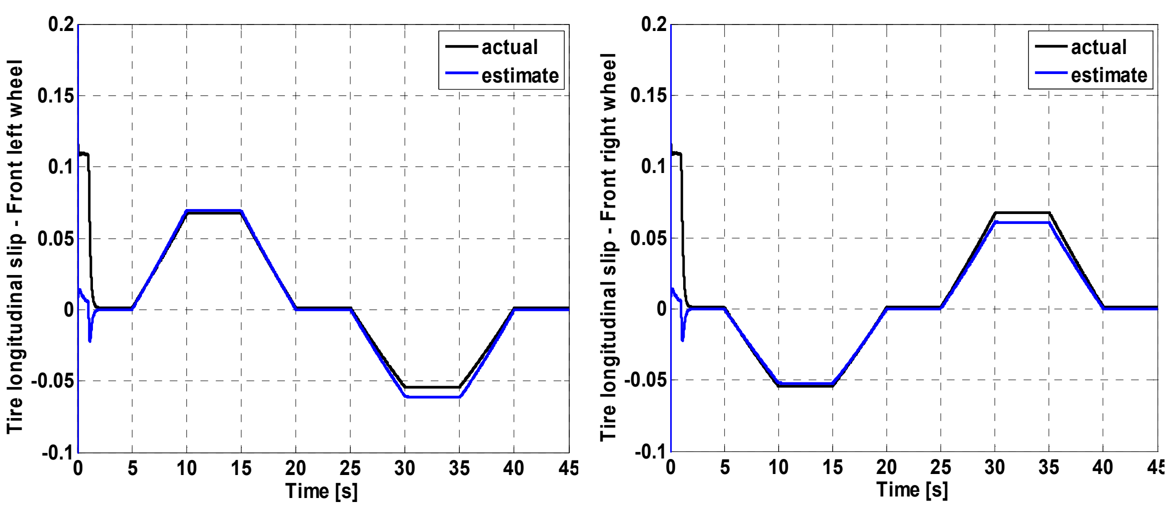
Fig. 4 and assumpting that the vehicle rolls on a plane road, (no slopangle) (Fig. 3). Based on simulation results presented on the following figures, through those reponses, it can be show that the derived model could represent a well vehicle dynamics.

Firstly, we estimated longitudinal and lateral velocities where in Fig. 5a the estimated longitudinal velocity state may track the velocity of the observer model and Fig. 5b give a good traking of the lateral estimated velocity to simulated one. secondly, longitudinal slips of the four wheels follow the vehicle model outputs, Fig. 6a to 6d in a hand, in another hand and according to Fig. 7a to 7d tire sideslip angles were appraise in accordance of the later estimations. Finally, it can be seen from Fig. 8a to 8d that follow steps in longitudinal tire road forces estimation lead to a satisfactory results. The time responses of the individual tire model output force in the lateral



(a) (b)

Fig. 5: Longitudinal and lateral velocities and their estimations



(a) (b)

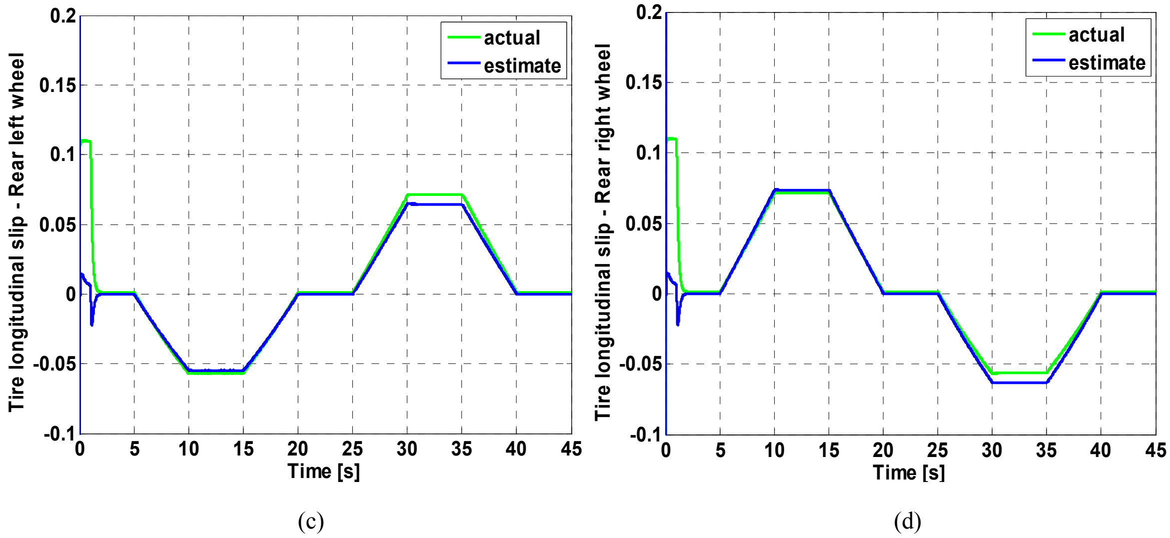


Fig. 6: Tire longitudinal slip and their estimations

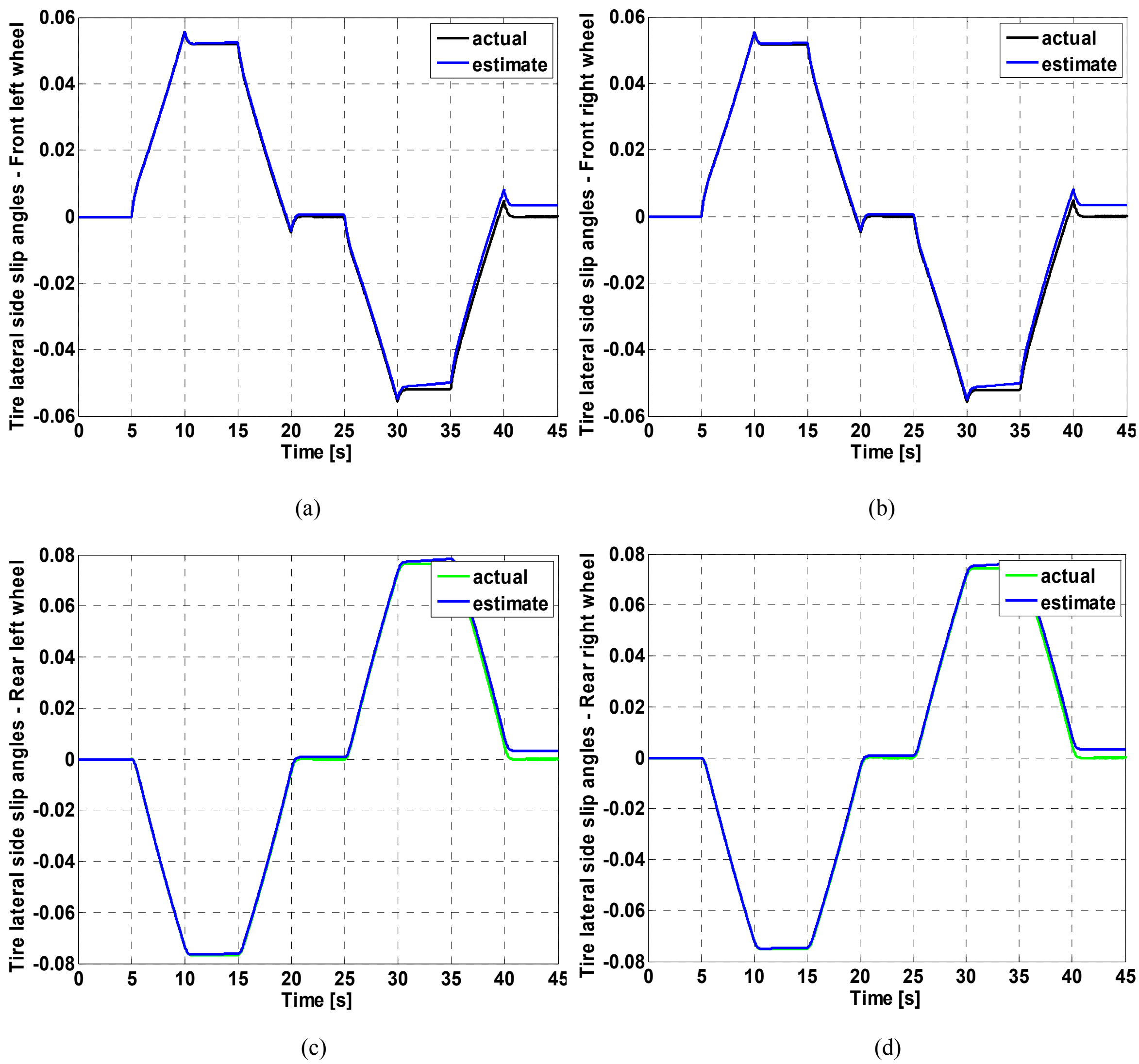
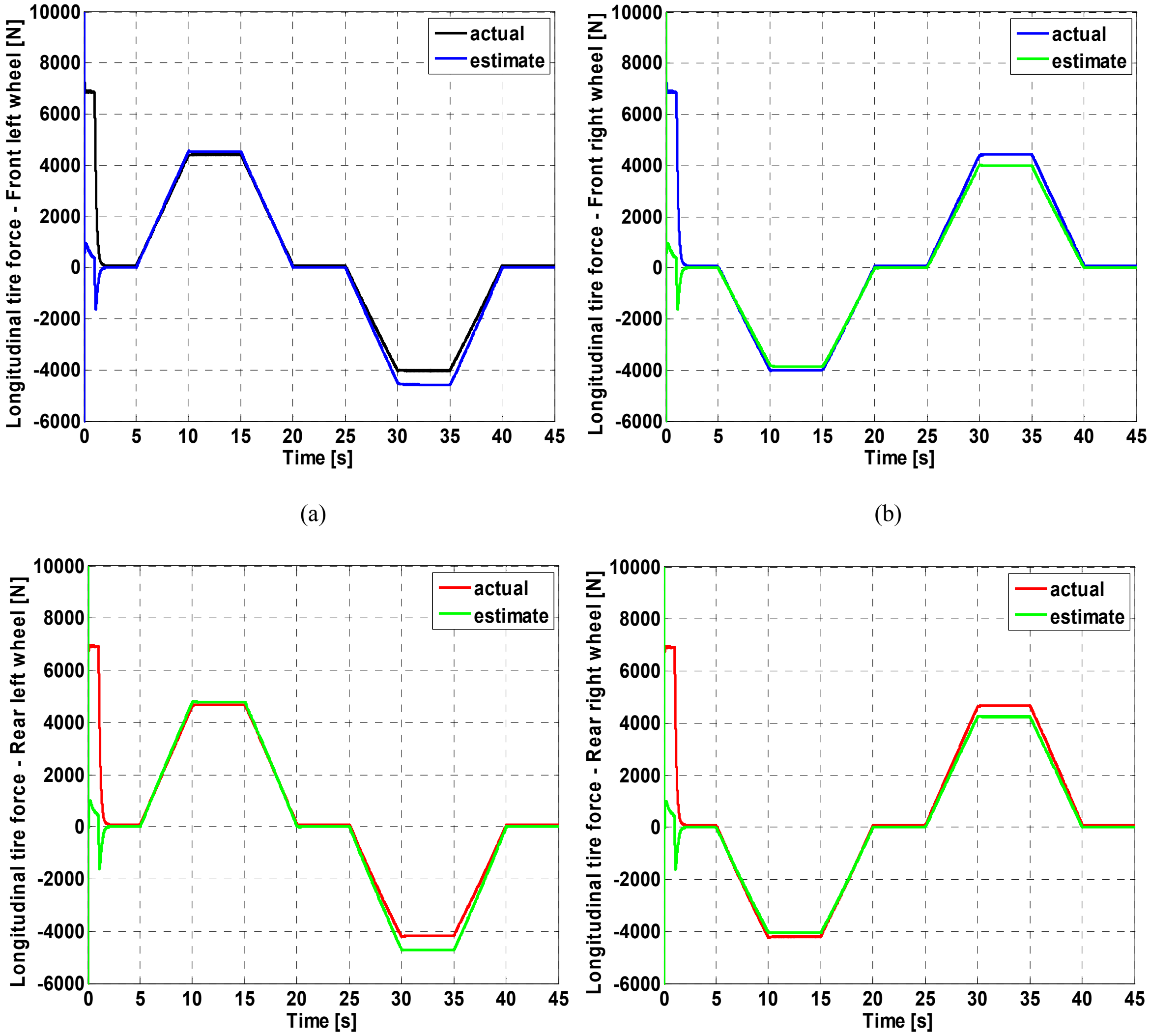


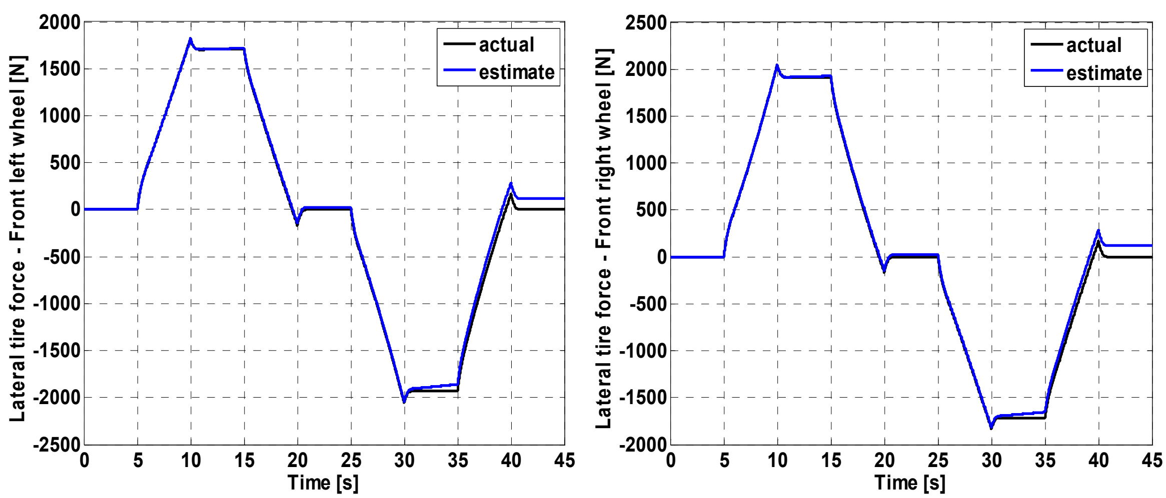
Fig. 7: Tire lateral side slip and their estimations



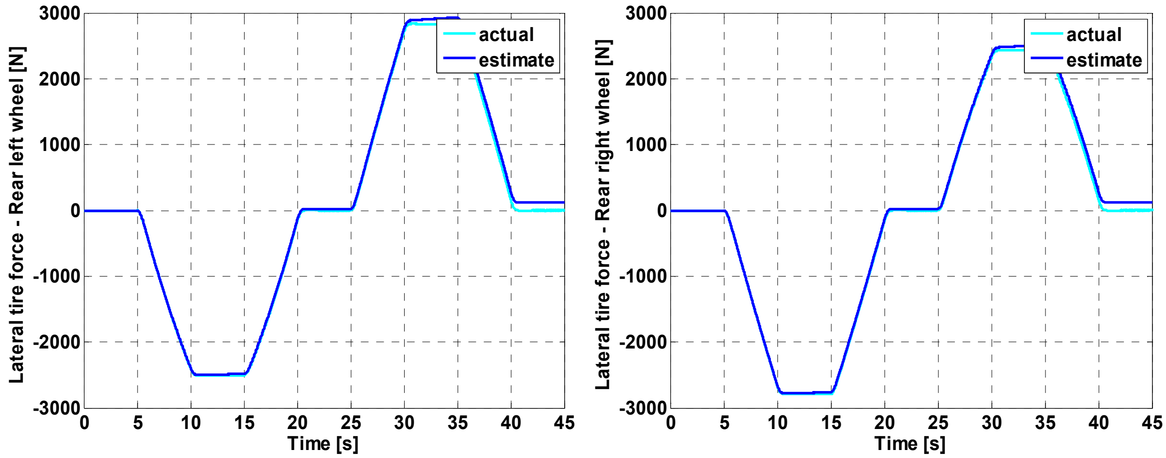
(a) (b)

(c) (d)

Fig. 8: Longitudinal tire forces and their estimations



(a) (b)



(c) (d)

Fig. 9: Lateral tire forces and their estimations

direction and their estimated values are presented in Fig. 9a to 9d.

CONCLUSION

In this study, we have discussed an estimation approach for longitudinal and lateral tire road forces, for these and first a four in wheel electric vehicle is carried out. Next, we proposed velocities estimations, in addition and based on these estimations longitudinal tire slips and side slip angles are presented. Then, the simulation results show the performance of this method and verify its effectiveness and accuracy on offered reponses especialy in steady state.

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