Study of an Optimization Criterion of Mounting Blocks for Drivability Evaluation of an Electric Vehicle

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***Abstract*— In order to lead properly an electric vehicle project, it is very important to assess and optimize the driver perception during manoeuvres such as tip-in and tip-out. This aspect of vehicle customer perception is called "drivability". Precedents papers have shown that the motor block (i.e. electric motor and speed reducer) rolling motion onto its mounting blocks has an important effect on the drivability rating. Moreover, this drivability aspect is crucial for an electric vehicle (because of the high torque gradient at low speeds or between regenerative and motor modes). A usual criterion used to study the impact of the motor block rolling on the drivabilty rating is the energy decoupling between roll and surge modes. This criterion, called "EcTx criterion" warrants to limit the oscillations of the longitudinal force transmitted from the motor block to the car body. As this criterion is independent of the torque command, it is used for any manoeuvre. Its application to a Key-On/Key-Off manoeuvre on conventional vehicle is well-known and enables to predict the rating of this manoeuvre. Nevertheless, questions are remaining about the appliance of this criterion to a tip-in manoeuvre. Simulations have shown that the EcTx criterion can also be used for a tip-in manoeuvre and gives a good prediction of the oscillations amplitude for the longitudinal force transmitted from the motor block to the car body. However, the driveline mode is also involved in the vehicle dynamics during a tip-in manoeuvre. Moreover, this paper shows that the oscillations of the motor block onto its mounting blocks can also damp the driveline mode oscillations thanks to a specific phase shift between the two modes. Therefore, EcTx criterion can be improved by taking into account the driveline equivalent stiffness and inertia into its calculation.**

Keywords: electric vehicle; drivability; vehicle dynamics; mountings blocks; coupled modes; kinetic energy.

I. Introduction

Within the context of the new European emission standard, automotive engineers have to focus their research on fuel-efficient vehicles. Therefore, many of car manufacturers are oriented on zero emission vehicles as a solution, hence the electric vehicle.

In order to lead properly this vehicle project from the very early design steps, it is very important to assess and optimize the customer perception during vehicle use. We also need to assess the difference between the vehicle response to driver request and that expected from the driver himself. This aspect of vehicle customer perception is called "drivability" and is evaluated by numerous manoeuvres such as tip-in and back-out of the accelerator pedal ("tip-out") or acceleration from a rest position ("take-off"). During tip-in, the torque gradient causes oscillations of the driveline and a jerk of the vehicle. Moreover, precedent works have shown that a torque excitation causes the roll motion of the motor block (i.e. the electric motor and the speed-reducer) onto its mounting blocks and therefore shocks on the vehicle acceleration [1].

The acceleration shocks amplitude during a tip-in is commonly used as a requirement in order to rate the drivability of the vehicle [2]. To meet these requirements, physical criteria on hardware components are very useful. Thus, one of the criterion used for mounting blocks pre-sizing is the "EcTx criterion", which gives the percentage of kinetic energy decoupling for the motor block surge dynamic. Actually, our aim is that the roll motion of the motor block will not be energetically transmitted to the longitudinal motion of the car body. To ensure that, we set a specification that the EcTx must be the highest possible. It means that the highest possible part of the kinetic energy of the motor block surge dynamic mode should come from the motor block longitudinal translation degree of freedom (others criterions can also be used to reach the same modal decoupling objective, see [3] and [4]). Thus, the higher the EcTx is, the lower are the oscillations of the longitudinal forces transmitted from the motor block to the car body. This criterion allows an accurate prevision of the drivality rating for Key-On/Key-Off (K02) or idle manoeuvres on conventional vehicle. Actually, the motor torque during a Key-On/Key-Off is null, so that the forces received by the car body only comes from the motor block. Nevertheless, as the EcTx citeria is independent from the motor torque excitation [5], it is also used for any torque excitation, especially the tip-in manoeuvre. Then, the mounting block stiffnesses used for this manoeuvre are obtained for a 10Hz excitation, or can also

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be set by applying a multiplication coefficient to the static KO2 stiffness.

Nevertheless, many previous works have shown that for a tip-in manoeuvre, the acceleration oscillations are also impacted by the driveline mode (see [6] for example). Therefore, our aim is to demonstrate that the oscillations of the motor block onto its mounting blocks can also damp the driveline mode oscillations because of the phase shift between the two modes.

Therefore, we will first present the hypothesis and the main calculations steps for the EcTx determination. Secondly, in order to correlate this criterion with the drivability rating, we will use our drivability simulation platform [7]. This simulation platform will enable to compare the shocks of the vehicle acceleration for a tip-in manoeuvre and for several mounting blocks stiffnesses (i.e. for several EcTx values).

II. EcTx Calculation

A. Hypotheses and parameters

The motor block is considered as a perfectly stiff mass (which can move onto six degrees of freedom) connected to the car body by three mounting blocks, each considered as a point composed of three uncoupled translation stiffnesses.

In order to describe the vehicle dynamics, the used frames must be presented (see Fig.1.). The Galilean frame $R_0=(U_0,V_0,W_0)$ is used to fix a reference for the vehicle in order to locate the car body during its motion. Its location has to be defined by the user: so this frame's center is set at the center of front wheel axles. The second frame is $R_{grid} = O_{xyz} = (O, \vec{u}_x, \vec{u}_y, \vec{u}_z)$ "car body grid frame" and has the same center as R_0. This frame is used to locate connected bodies to the car body solid (for example suspensions or junction with engine mounts). All the matrixes and vectors written in this part are expressed in R_{grid}. We also define the motor block mass/inertia matrix M_{mb} at its center of gravity G:

$$M_{mb} = \begin{bmatrix} m_{mb} & 0 & 0 & 0 & 0 & 0 \\ 0 & m_{mb} & 0 & 0 & 0 & 0 \\ 0 & 0 & m_{mb} & 0 & 0 & 0 \\ 0 & 0 & 0 & I_{xx} & -I_{xy} & -I_{xz} \\ 0 & 0 & 0 & -I_{xy} & I_{yy} & -I_{yz} \\ 0 & 0 & 0 & -I_{xz} & -I_{yz} & I_{zz} \end{bmatrix}_{Rgrid} , \qquad (1)$$

where m_{mb} is the motor block mass, I_{xx} is its moment of inertia around u_x axis and I_{xy} is its product of inertia around u_x and u_y axes.

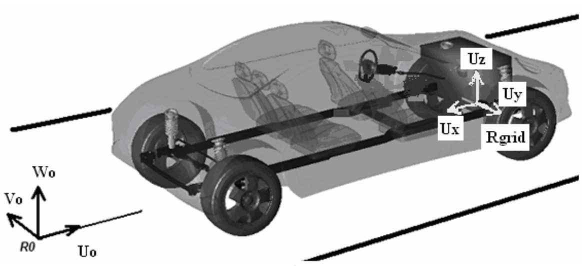

Fig. 1. Frames used for the calculations

For each mounting block i (i=1..3), the stiffness matrix K_i is

$$K_i = \begin{pmatrix} K_{ix} & 0 & 0 \\ 0 & K_{iy} & 0 \\ 0 & 0 & K_{iz} \end{pmatrix}_{Rgrid} , \qquad (2)$$

where K_{ij} is the stiffness of mounting block i onto j direction.

The displacement $\vec{\delta I}$ of the mouting block i fixation point I is function of G displacement (in both translation and rotation motions, respectively $\vec{D}_G$ and $\vec{\theta}$), as described by

$$\vec{\delta I} = (\vec{D}_G + \vec{IG} \wedge \vec{\theta}) ; \qquad (3)$$

with the following decompositions of the previous vectors:

$$\begin{cases} \vec{D}_G = X_G * \vec{u}_x + Y_G * \vec{u}_y + Z_G * \vec{u}_z \\ \vec{\theta} = \theta_x * \vec{u}_x + \theta_y * \vec{u}_y + \theta_z * \vec{u}_z \\ \vec{IG} = BL_{ix} * \vec{u}_x + BL_{iy} * \vec{u}_y + BL_{iz} * \vec{u}_z \end{cases} . \qquad (4)$$

In (4) BL_{ij} is the lever arm in j direction between mounting block i and G.

Thus, the expression of $\vec{\delta I}$ is given by

$$\vec{\delta I} = \begin{vmatrix} X_G + BL_{iy} * \theta_z - BL_{iz} * \theta_y \\ Y_G + BL_{iz} * \theta_x - BL_{ix} * \theta_z \\ Z_G + BL_{ix} * \theta_y - BL_{iy} * \theta_x \end{vmatrix} . \qquad (5)$$

For useful notation, we define the following vector:

$$q_I = [q_1 q_2 q_3 q_4 q_5 q_6] = \left[X_G Y_G Z_G \theta_x \theta_y \theta_z \right] . \qquad (6)$$

The potential energy E_{pi} of the mounting block i at its fixation point I is given by

$$E_{pi} = \frac{1}{2} * \vec{\delta I}^t * K_i * \vec{\delta I} , \qquad (7)$$

As E_{pi} is now known, we can give a simple expression for each coefficient of the {motor block + mounting blocks} system 6*6 stiffness matrix K_{SMO}:

$$K_{SMO}(a,b) = \sum_{i=1}^{3} \frac{\partial^2 E_{pi}}{\partial q_a \partial q_b}, \qquad (8)$$

with a and b natural numbers between 1 and 6.

As both mass and stiffness matrix of our system are known, we can now assess its kinetic energy.

B. Kinetic energy of the system

The application of the Lagrange equation to our system S={motor block+mounting blocks} allows us to write (9), which gives the equilibrium of S in the case of little displacements during free vibrations:

$$M_{mb} * \ddot{q} + K_{SMO} * q = 0, \qquad (9)$$

where $q=\{x\ y\ z\ \theta_x\ \theta_y\ \theta_z\}$ is the system's generalized coordinates vector.

We search a synchronous solution for (9), i.e we define Y and $\Phi \neq 0$ so that $q(t)=Y*\Phi(t)$, where t is time and Y is the eigen vectors matrix. As these eigen vectors define a basis for the vector space of the 6*1 vectors, Y is invertible. Moreover, as M_{mb} is a positive defined matrix, M_{mb} is also invertible, so that we can write

$$\frac{\ddot{\Phi}}{\Phi} = -M_{mb}^{-1} * K_{SMO}. \qquad (10)$$

As M_{mb} and K_{SMO} are respectively positive and semi-positive defined, $\exists \Omega$ so that

$$\begin{cases} \Omega^2 * M_{mb} * Y = K_{SMO} * Y \\ \ddot{\Phi} + \Omega^2 * \Phi = 0 \end{cases}. \qquad (11)$$

We only focus on the spatial resolution of (11). Therefore, we search the solutions of the following generalized eigenvalues problem:

$$\left(K_{SMO} - \Omega^2 * M_{mb}\right) * Y = 0. \qquad (12)$$

We make the assumption that (12) admits at least a non-null solution Y. Therefore, we have

$$Y \in Ker\left(K_{SMO} - \Omega^2 * M_{mb}\right),\ Y \neq 0. \qquad (13)$$

As $Ker\left(K_{SMO} - \Omega^2 * M_{mb}\right) \neq \{0\}$, we have

$$\det\left(K_{SMO} - \Omega^2 * M_{mb}\right) = 0. \qquad (14)$$

Resolution of (14) allows finding all the eigen frequencies ω_m. We found the associated eigen vectors Y_m by solving the following homogenous system

$$\left(K_{SMO} - \omega_m^2 * M_{mb}\right) * Y_m = 0, \qquad (15)$$

with m natural number between 1 and 6.

The kinetic energy of S is given by

$$T = \frac{1}{2} * \dot{q}^t * M_{mb} * \dot{q}. \qquad (16)$$

Then, kinetic energy of S for the mode m is given by

$$T_m = \frac{1}{2} * \left(m_{mb} * \sum_{j=1}^{3} Y(j,m)^2 + Y(4:6,m)^t * I_g * Y(4:6,m) \right) * \dot{\Phi}(t), \qquad (17)$$

where $Y(4:6,m)=Y_m(4:6)$ and I_g is the inertia matrix of the motor block in R_{grid}.

We also define $Y(:,m)=Y_m=\{x_m\ y_m\ z_m\ \theta_{xm}\ \theta_{ym}\ \theta_{zm}\}$, so that the kinetic energy of S for the mode m is the following:

$$\begin{aligned} T_m = & \frac{1}{2} m_{mb} * \left(x_m^2 + y_m^2 + z_m^2 \right) * \dot{\Phi}(t) ... \\ & + \frac{1}{2} (I_{xx}\theta_{xm}^2 + I_{yy}\theta_{ym}^2 + I_{zz}\theta_{zm}^2) * \dot{\Phi}(t) ... \\ & - (I_{xy}\theta_{xm}\theta_{ym} + I_{xz}\theta_{xm}\theta_{zm} + I_{yz}\theta_{ym}\theta_{zm}) * \dot{\Phi}(t) \end{aligned}. \qquad (18)$$

In addition to the six classical dimensions, we can notice that three additional rotation-coupling dimensions appear.

Finally, the energetic contribution ratio of the surge's degree of freedom in the total kinectic energy of mode m is

$$E_c T_{mx} = \frac{1}{2} \frac{m_{mb} x_m^2 \dot{\Phi}(t)}{T_m}. \qquad (19)$$

Moreover, we can recognize the motor block surge mode among all m modes: it is the mode, for which the contribution of the surge degree of freedom is the greater. Therefore, the EcTx value is given by the greater value of $E_c T_{mx}$.

Then, the EcTx criterion warrants decoupling of the motor block surge and rolling motions by ensuring that more than 95% of the kinetic energy of the surge mode must come from the contribution of the surge degree of freedom.

III. Impact Of The EcTx Criterion On Drivability Rating

Our aim is to assess the impact of a variation of the EcTx value (by changing mounting block stiffness) on the drivability rating for a tip-in manoeuvre. In order to reach this objective, we will use our electric vehicle simulation platform, which is composed of a physical model and a control part.

The first part models physical characteristics of the electric vehicle with LMS-AMESim®. The second part sets the control of the vehicle with Matlab/Simulink®. The physical model of the vehicle is representative of the vehicle longitudinal and vertical accelerations, as well as its pitching. The control part developed on Matlab/Simulink® computes the torque command of the electric motor in all cases of study from both acceleration and braking pedals. This torque setpoint must guarantee a good quality of drivability.

In this study, we will only focus on the physical part of our model, which will be the only part used for the correlation between EcTx and drivability rating. Therefore, we will first give a description of this physical model.

A. Description of the electric vehicle simulation platform

We have modelled an electric vehicle based on light duty car on AMESim® (see Fig.2 for the sketch of the model). The dynamic field of the motor block drivability concerns the frequencies between 0 and 20 Hz. Thus, the study of the vehicle drivability needs to take into account all the mass-spring systems, which could have a low resonance frequency. These mass-spring systems are those who have a high mass and a low stiffness: car body and motor mass and inertia, unsprung masses, stiffness of vehicle suspensions and motor mountings and both tires and transmissions stiffness and inertias.

The motor block submodel moves onto all possible degrees of freedom. The motor block is considered perfectly stiff and hanged on the car body through the mounting blocks. The motor rotor is turning inside the motor block. The three mounting blocks are considered as damped spring with stiffness depending on the displacement in compression or traction for all directions of translation. The positions of the mounting blocks for both car body and motor block sides are documented in a junction submodel.

The motor block dynamic is given in the local inertia frame R_i, which origin is the center of gravity of the motor block. This frame is used here in order to give easily the inertia matrix $[I_{mb}]$ of motor block. Therefore, we express the rotation dynamic of the motor block in this frame R_i by

$$[I_{mb}]*\dot{\vec{\Omega}}_{Ri/R0} = \sum \vec{T}_{ext+\text{int}\to mb} - \left[\vec{\Omega}_{Ri/R0} \wedge \left([I_{mb}]*\vec{\Omega}_{Ri/R0}\right)\right]. \quad (20)$$

In (20), $\vec{\Omega}_{Ri/Ro}$ is the rotation speed of the motor block expressed in R_0 (and the rotation speed between frames R_i and R_0), and $\Sigma\, \vec{T}_{ext+\text{int}\to mb}$ is the sum of both externals and internals torques applied to the motor block center of gravity. The externals torques consist of the reaction torque from the differential and the torque transmitted from the car body to the motor block via the mounting blocks. The internal torque is the reaction torque of the motor rotor inertia. This inertia is submitted to the motor command torque, the primary stiffness reaction torque and a friction torque.

The translation dynamic of the motor block is expressed in the frame R_0 and at G by the following equation:

$$m_{mb} * \vec{\gamma}_{Ri/Ro} = \sum \vec{F}_{ext\to mb} = \vec{F}_{mount\to mb} + \vec{F}_{gravity\to mb}, \quad (21)$$

where m_{mb} is the motor block mass, $\vec{\gamma}$ is the motor block acceleration and $\vec{F}_{ext\to mb}$ is the sum of all externals forces applied to the motor block. These forces are those transmitted from the car body to the motor block through the mounting blocks $\vec{F}_{mount\to mb}$ and also the weight of the motor block $\vec{F}_{gravity\to mb}$.

The car body submodel is set on its suspensions and is responding to the external actions of the wind and the electric motor. A symmetric vehicle is modeled, which can only be animated in a straight road motion. It means that the steering motion of the vehicle is not taken into account.

The vehicle suspension submodel represents stiffness and damping in both vertical and longitudinal directions (in vehicle frame). The vertical stiffness depends on compression displacement in order to represent the bumper stops. As we make the approximation of a symmetric car body, both suspensions of an axle will be represented by one equivalent suspension.

The wheel submodel is able to characterize the forces at the ground/tire contact, the rolling resistance (as a function of the wheel speed and the vertical load) and both road slope and slip. We use a simplified Pacejka model to compute the longitudinal force of the wheel; see [8] for more details.

The maximum power of the synchronous electric motor is about 40kW, for a maximum torque of 190N.m. The torque delivered by the electric motor is transmitted to the wheels through a simple speed-reducer of ratio r =11.3.

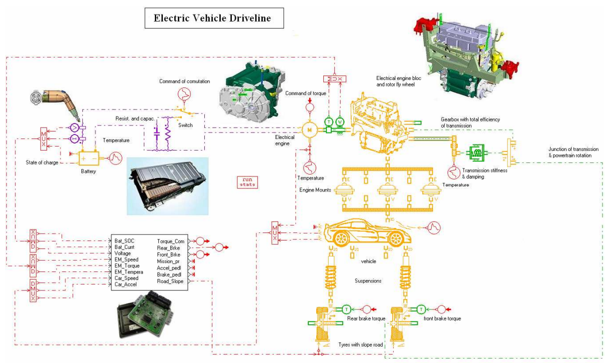

Fig. 2. Sketch of the electric vehicle physical model on LMS-AMESim®

By delivering a negative torque, the motor turns to generator and is able to recharge the battery during the deceleration phases. The electric motor losses depend on torque and rotary velocity. The reducer and the differential losses are merged in the reducer submodel and set as a constant efficiency. More precisions about this platform, including the main equations employed can be found in [7].

B. Results of simulation and discussion

The manoeuvre simulated is a full tip-in after stabilization of the speed at 30 kph. The torque command is build by two main blocks. First, an interpretation of the driver request enables to compute a raw torque command. Then, the torque setpoint filter (which principle is described in [9]) allows a damping of the drivetrain oscillations. Simulations are done for different mounting blocks stiffness values resulting in three EcTx values (49%, 67 % and 80%).

Firstly, Fig.3. shows that oscillations of the longitudinal force transmitted by the motor block to the car body are very high for the lowest EcTx value. The damping of these oscillations are better for EcTx=67% and EcTx=80%, which both give very close oscillations. The drivability for this manoeuvre is rated by the vehicle longitudinal acceleration oscillations amplitude (especially the first oscillation amplitude, also called “kick”), which is given by Fig.4. The acceleration oscillations are better damped for a high EcTx value, for which the manoeuvre rating will be therefore better.

Nevertheless, as we can see on Fig.3., the force applied by the motor block on the car body has a greater kick for EcTx=80% than for EcTx=67%, so that we have to compare the other longitudinal forces (i.e. forces coming from both front and rear suspensions) in order to justify the acceleration curves. Thus, Fig.5. gives all longitudinal forces applied to the car body for EcTx=67% and EcTx=80%. The forces coming from both front and rear suspensions are almost the same for both EcTx values, especially for the kick. In other words, only the variation of the “motor block force” kick explains the lower acceleration kick for EcTx=80%.

For EcTx=80%, an optimal dynamic behaviour between motor block forces and suspensions forces allows a better acceleration damping. Actually, as shown on Fig.5., the time response of the motor block is greater for EcTx=80% and warrants a kick which is opposite to the kick of the resulting suspensions force. Moreover, oscillations of the suspensions forces are mainly caused by the driveline mode resonance (see [7] for more details about frequency and origin of this mode). Thereafter, the kick of the total longitudinal force is damped thanks to this greater time response, and the following oscillations are damped thanks to the optimal phase shift between driveline and motor block modes. Therefore, an improvement of the EcTx criterion in order to apply it to a tip-in manoeuvre would be to include the driveline stiffness and inertia into the EcTx calculations presented in part II.A, so that the dynamic coupling of driveline and motor block modes can be taken into account.

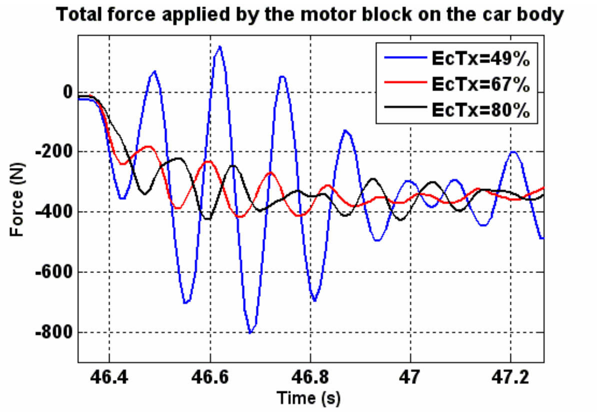

Fig. 3. Longitudinal force applied by the motor block on the car body

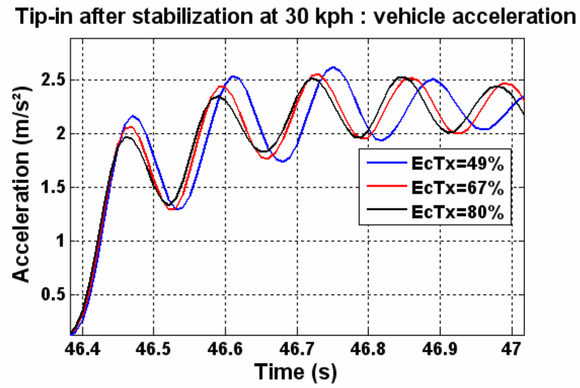

Fig. 4. Vehicle Acceleration for different EcTx values

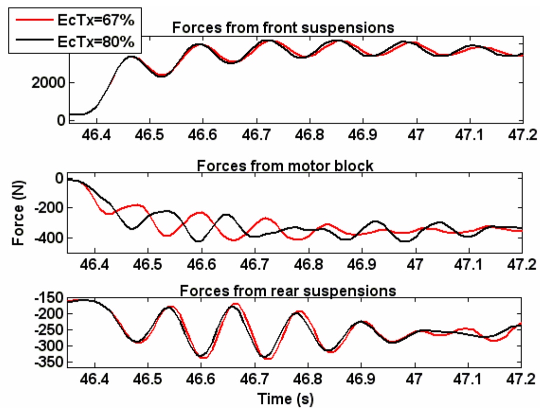

Fig. 5. Longitudinal forces applied on the car body

REFERENCES

[1] O. Hayat, M. Lebrun, E.Domingues, “Powertrain driveability evaluation : analysis and simplification of dynamic models”, SAE paper n°011326, 2003. 2003, vol. 112, n°7, pp. 569-578

[2] R.E. Horey, C.B. Holmes, “Vehicle driveability – its characterisation and measurement”, SAE paper n°010949, 1999.

[3] P.E.Geck, R.D.Patton, “Front wheel drive engine optimization”, SAE paper n°840736, 1984.

[4] T. Arai, T.Kubozuka, S.D. Gray, “Development of an engine mount optimization method using modal parameters”, SAE paper n°932898, 1993.

[5] E. Courteille, F.Mortier, L.Leotoing, E.Ragneau, “Optimisation d’un système de suspension moteur pour une amélioration du confort vibro-acoustique”, Proceedings of the 16e Congrès Français de Mécanique, Nice, september 2003.

[6] J. Baumann, D.D. Torkzadeh, A. Ramstein, U. Kiencke, T. Schegl. “Model-based predictive anti-jerk control”, In Control Engineering Practice 14, 259-266, 2006.

[7] B. Eller, J.F-Hetet, S.André, G.Hennequet, “Electric vehicle platform for drivability analysis”, Proceedings of the IEEE ICCA 2010 conference, Xiamen, 2010.

[8] E. Bakker, L.Y. Nyborg, H.B. Pacejka. “Tyre modeling for use in vehicle dynamics studies”, SAE paper n°870421, 1987.

[9] M.Gati, “Modélisation, observation et commande de systèmes dynamiques hybrides : application à l’automobile”, PhD dissertation, Université de Paris-Sud, 2006.