Practical Model for Real Structure and Performance of Heavy Vehicles

Younès Abbassi¹, Youcef Aït-Amirat² and Rachid Outbib³

1. Department of Energy, University of Technology of Belfort-Montbéliard, Belfort 90010, France

2. Department of Energy, University of Franche-Comté, Belfort 90010, France

3. LSIS (Laboratoire des Sciences de l'Information et des Systèmes), University of Aix-Marseille, Marseille 13397, France

Received: November 14, 2014 / Accepted: December 01, 2014 / Published: February 25, 2015.

Abstract: In this paper, a new practical model for real heavy vehicle structure is developed to investigate dynamic responses under steering/acceleration or braking maneuvers. The generalized six DoFs (degrees-of-freedom) nonlinear vehicle model M_1 including longitudinal, lateral, yaw, vertical, roll and pitch dynamics is validated using the measured data reported in different studies. This model takes the CG (center of gravity) of sprung mass, unsprung mass and total vehicle mass into account. Based on this model, the effects of the inertia parameters on the vehicle dynamic responses are investigated for more comprehensive assessments of the model structure. Another nonlinear vehicle model M_2 derived from M_1 which assumes that the vehicle has a single CG as reported in literature is also developed. The dynamic responses of the vehicle model M_1 are compared with those of the model M_2 to demonstrate the performance potential of the proposed nonlinear model. The results of dynamic responses with the nonlinear vehicle model M_1 suggest that the model could offer considerable potential in realizing enhanced ride and handling performance, as well as improved roll and pitch properties in a flexible manner.

Key words: Modeling, nonlinear vehicle models, inertial parameters, heavy vehicle dynamics, ride, handling.

1. Introduction

Steering, braking or acceleration maneuvers of road vehicles induce generally comprehensive magnitudes of vehicle yaw, roll and pitch motion and thus the knowledge of the position of various center of gravity. During motion in particular in turn, the CG (center of gravity) of the vehicle, the CG of the unsprung mass and that of the sprung mass are different arising from dynamic load transfers due to the cornering and braking forces developed by the tires [1, 2]. Taking into account both CG of the sprung mass and unsprung mass can give more flexibility to the vehicle and inhibit the longitudinal and lateral load transfers. To study the vehicle dynamics, a range of linear and nonlinear models of road vehicles have been reported in the

literature for analysis of ride, handling and directional control characteristics. These models vary from simple linear models [3, 4] to complex nonlinear models [4, 5]. Actually, the vehicle dynamics is coupled into a single dynamic model in which all elements are interconnected and act differently. This means that the vehicle is composed with different rigid bodies and elastic elements and not with a single rigid body which has only center of gravity. The different models generally consider that the vehicle has only one center of gravity G . Accordingly, the vehicle is assimilated to one solid placed firmly on the wheels. The majority of the reported studies concentrate on solving for the design conflict between the ride comforts and handling performance requirements. While the majority of studies focus more on the ride dynamics aspect ignoring the effects of inertial parameters. As a consequence of this consideration to the vehicle dynamics, the flexibility in maneuvering is altered and

Corresponding author: Younès Abbassi, Dr. Eng., research fields: vehicles system dynamics and nonlinear dynamics and control. E-mail: younes.abbassi@ac-besancon.fr.

the inertial parameters such moments of inertia can be directly reduced. Therefore, yaw, roll and pitch increase that will have a direct impact on the vehicle handling [6, 7], braking and traction performance [8]. These effects are well reported in the literature and the above references are by no means exhaustive. In the absence of real dynamic model which takes into account both the inertial parameters of sprung mass and unsprung mass and total vehicle mass, the discrepancies between the real vehicle design and its model dynamic become large. These discrepancies affect such active control system which often used fixed values for the inertial parameters that lie somewhere between the maximum and minimum expected values.

In this works, the main goal is to show how the vehicle motion is affected when we consider only one CG for the total vehicle mass. It also shows the stability and flexibility properties results when we consider simultaneously the CG of sprung mass G_s and the CG of unsprung mass G_u . Moreover, the roll and pitch characteristics of the vehicle suspension are inhibited as well as the lateral and longitudinal load transfers. As a result, the suspension could provide adequate attenuation of the road-induced vibration.

This paper is organized as follows: Section 2 develops equations of motion for the present works and derives two nonlinear vehicle models M_1 and M_2 suitable for the present study; Section 3 discusses the simulation results and the validation of the proposed vehicle model M_1 ; finally, Section 4 gives conclusions.

2. Nonlinear Vehicle Models

A range of linear and nonlinear models of road vehicles are reported in the literature for vehicles analysis, handling and control [9, 10]. These models consider that the vehicle has only one CG as shown in Fig. 3. This means that the distances c_p and c_u and the height h_u are zero, i.e., $c_p = c_u = h_u = 0$ and $h_p = h$. Therefore, the roll, pitch and yaw moments of inertia of the total vehicle mass: I_x , I_y and I_z are equal to those of the only sprung mass: I_{xs} , I_{ys} and I_{zs} , i.e., $I_x = I_{xs}$, $I_y = I_{ys}$

and $I_z = I_{zs}$. It results that many geometrical quantities are not taken into account and can thus influence the dynamic behavior of the vehicle. These geometric quantities appear when we consider that the vehicle is composed of two bodies: sprung mass and unsprung mass and each has its own center of mass.

2.1 Nonlinear Vehicle Model: First Model M_1

In this study, a generalized six DoFs (degrees-of-freedom) nonlinear model of a two axis vehicle, incorporating sprung and unsprung mass configurations, is developed to investigate dynamic responses to steering and braking/acceleration inputs as well as excitations arising from road roughness or crosswinds. Moreover, this nonlinear vehicle model offers the essential vehicle flexibility and may thus be applicable for the studies of rollover prevention strategies.

The proposed model shown in Fig. 1 includes six DOFs, three frame Cartesian displacements and three Euler angles: roll angle θ , pitch angle ϕ and yaw angle ψ . The rates of the Euler angles (roll rate V_θ , pitch rate V_ϕ and yaw rate V_ψ) are derived from Ref. [11] and can be written as:

$$\begin{bmatrix} \dot{\theta} \\ \dot{\phi} \\ \dot{\psi} \end{bmatrix} = \begin{bmatrix} V_\theta + \phi V_\psi \\ V_\phi - \theta V_\psi \\ V_\psi + \theta V_\phi \end{bmatrix} \quad (1)$$

where, $\dot{\theta}$, $\dot{\phi}$ and $\dot{\psi}$ are the roll, pitch and yaw angular velocities. V_θ , V_ϕ and V_ψ are roll, pitch and yaw velocities, respectively.

The total vehicle mass M has a center of mass G and is composed of sprung mass M_s and unsprung mass M_u . The sprung mass could rotate about its roll axis, while the unsprung mass M_u is supposed to be the mass of a one solid which contains the total mass of the wheels and others solids attached to the bottom of the suspension.

The coordinates of the CG of total vehicle mass, sprung mass and unsprung mass in the vehicle reference frame (x_0, y_0, z_0) are $(0, 0, h)$, $(-c_u, 0, h_u)$

Fig. 1 Heavy vehicle structure for model M_1 .

Fig. 2 Nonlinear braking/acceleration and cornering forces.

and $(c_u, 0, h_u)$, respectively. The exact position of the CG of unsprung mass is generally determined by the vehicle constructor. The roll angle and the moments of inertia of the unsprung mass are usually small and thus could be neglected. The kinetic and potential energy of the unsprung mass depend generally on the total mass of wheels for simplicity, the unsprung mass M_u can be assumed to be the sum of the total mass of the wheels M_{wi} ($i = 1, \dots, 4$) and the front and rear spindle mass (M_{spf} , M_{spr}).

$$M_u = M_{spf} + M_{spr} + \sum_i M_{wi} \quad (2)$$

The front-wheel steering input δ_f is assumed to be identical for both the front wheels, while the rear-wheel steering input δ_r could also be conveniently included in the vehicle model for analysis of vehicles with 4WS (four-wheel steering).

The nonlinear braking, acceleration F_{xi} and cornering forces F_{yi} shown in Fig. 2 developed by the tires [12], are evaluated as functions of the normal

forces, slip angle and slip ratio using the Magic Formula tire model which is widely used as a leading tire model [13, 14]. The magic formula tire model is used to derive braking/acceleration, cornering forces and aligning moment developed by a tire, as a function of the slip angle α_i , ($i = 1, \dots, 4$), and the normal tire forces F_{zi} . By assuming small steering angles, the slip angle used in the tire model M_1 is can be expressed as

$$\begin{aligned} \alpha_1 &= \delta_f - \frac{V_y - \theta V_z + (a + c_u - c_u)V_\psi}{V_x + \phi V_z + l V_\psi} \\ \alpha_2 &= \delta_f - \frac{V_y - \theta V_z + (a + c_u - c_u)V_\psi}{V_x + \phi V_z - l V_\psi} \\ \alpha_3 &= \delta_r - \frac{V_y - \theta V_z - (b + c_p - c_u)V_\psi}{V_x + \phi V_z + l V_\psi} \\ \alpha_4 &= \delta_r - \frac{V_y - \theta V_z - (b + c_p - c_u)V_\psi}{V_x + \phi V_z - l V_\psi} \end{aligned} \quad (3)$$

where, δ_f and δ_r are the front and rear steering angles, respectively. V_x , V_y and V_z are longitudinal, lateral and vertical velocities of the sprung mass, respectively. a , b represent the distance from G to front and rear wheel axes. The distance l is the half gauge of the vehicle. c_p is the distance from G to G_s in x direction. c_u is the distance from G to G_u in x direction. V_ψ is the yaw velocity.

The excitations arising from crosswinds are incorporated in the equations of motion by an equivalent longitudinal force F_{ax} and equivalent lateral force F_{ay} applied to the vehicle CG [15].

$$\begin{aligned} F_{ax} &= -\frac{1}{2} S_x \rho c_{ax} V_x^2 = -k_{ax} V_x^2 \\ F_{ay} &= -\frac{1}{2} S_y \rho c_{ay} V_y^2 = -k_{ay} V_y^2 \end{aligned}$$

where, S is the aerodynamic area (shadow area); c_a is the aerodynamic coefficient and ρ is the air density ($\text{kg}\cdot\text{m}^{-3}$), default = 1.25.

The equations of motion for the sprung M_s and unsprung masse M_u can be obtained by using Lagrangian dynamics, and the nonlinear vehicle model M_1 can be written as:

$$\begin{cases}
M\dot{V}_x = MV_\psi V_y - M_s h V_\psi V_\theta - A_h \dot{V}_\phi \\
\quad + F_{xf} \cos \delta_f + F_{xr} \cos \delta_r - F_{yf} \sin \delta_f \\
\quad - F_{yr} \sin \delta_r - k_{ax} V_x^2 \\
M\dot{V}_y = -MV_\psi V_x + M_s h \dot{V}_\theta - A_c \dot{V}_\psi \\
\quad + F_{xf} \sin \delta_f + F_{xr} \sin \delta_r + F_{yf} \cos \delta_f \\
\quad + F_{yr} \cos \delta_r - k_{ay} V_y^2 \\
M_s \dot{V}_z = M_s g - (K_f + K_r)z - (B_f + B_r)V_z \\
\quad + (aK_f - bK_r)\phi + (aB_f - bB_r)V_\phi \\
\quad - M_s \phi \dot{V}_x \\
I_x \dot{V}_\theta = M_s h_p (\dot{V}_y + V_\psi V_x) + A_{yz} V_\psi V_\phi + A_{yz} V_\psi V_\phi \\
\quad + I_{xzs} \dot{V}_\psi - (l^2 K_f + l^2 K_r + B_{af} + B_{ar})\theta \\
\quad - (l^2 B_f + l^2 B_r)V_\theta - M_s gh\theta \\
\quad - (a^2 K_f + b^2 K_r - M_s gh)\phi \\
\quad - (a^2 B_f + b^2 B_r)V_\phi \\
I_y \dot{V}_\phi = M_s c_p \dot{V}_z - A_h (\dot{V}_x - V_\psi V_y) + A_{zx} V_\psi V_\theta \\
\quad - (a^2 K_f + b^2 K_r - M_s gh)\phi \\
\quad - (a^2 B_f + b^2 B_r)V_\phi + (aB_f - bB_r)V_z \\
\quad + (aK_f - bK_r)z \\
I_z \dot{V}_\psi = A_{xy} V_\theta V_\phi + a(F_{xf} \sin \delta_f + F_{yf} \cos \delta_f) \\
\quad - b(F_{xr} \sin \delta_r + F_{yr} \cos \delta_r) \\
\quad + l(\Delta F_{xf} \cos \delta_f - \Delta F_{yf} \sin \delta_f) \\
\quad + l(\Delta F_{xr} \cos \delta_r - \Delta F_{yr} \sin \delta_r) + I_{xzs} \dot{V}_\theta
\end{cases} \quad (4)$$

where,

$$\begin{cases}
A_h = M_s h_p + M_u h_p \\
A_c = M_s c_p + M_u c_u \\
A_{yz} = I_{ys} - I_{zs} \\
A_{zx} = I_{zs} - I_{xs}
\end{cases} \quad \text{and} \quad \begin{cases}
I_x = I_{xs} + M_s h_p^2 \\
I_y = I_{ys} + M_s (h_p^2 + c_p^2) \\
\quad + M_u (h_u^2 + c_u^2) \\
I_z = I_{zs} + M_s c_p^2 + M_u c_u^2
\end{cases} \quad (5)$$

$$\begin{cases}
F_{xf} = F_{x1} + F_{x2} \\
F_{xr} = F_{x3} + F_{x4} \\
\Delta F_{xf} = F_{x2} - F_{x1} \\
\Delta F_{xr} = F_{x4} - F_{x3}
\end{cases} \quad \text{and} \quad \begin{cases}
F_{yf} = F_{y1} + F_{y2} \\
F_{yr} = F_{y3} + F_{y4} \\
\Delta F_{yf} = F_{y2} - F_{y1} \\
\Delta F_{yr} = F_{y4} - F_{y3}
\end{cases} \quad (6)$$

In the developed model M_1 , the roll and pitch inertias of unsprung mass are not included for simplicity reason. The notations V_x , V_y and V_z are longitudinal, lateral and vertical velocities of the vehicle, respectively. V_θ , V_ϕ and V_ψ are the roll, pitch and yaw rate. g is acceleration due to gravity and a , b represent the distance from G to front and rear wheel axes. The distance l is the half gauge of the vehicle. B_{af} and B_{ar} are stiffness constants of the front and rear anti-roll bars, respectively. K_f and K_r are stiffness constants of the front and rear suspensions, respectively. B_f and B_r are stiffness constants of the front and rear damping, respectively. I_{xz} and I_{xzs} are pitch-plane cross moment of inertia of total vehicle mass and sprung mass, respectively.

2.2 Nonlinear Vehicle Model: Second Model M_2

To investigate the effects of the inertial parameters on the vehicle stability and maneuvering capabilities, we consider a second model M_2 usually used in literature as proposed in Refs. [16, 17]. This model is deduced from model M_1 when the CG of sprung and unsprung mass are merged with the CG of the total vehicle mass as shown in Fig. 3. This means in model M_2 , we have:

$$G = G_s = G_u \Rightarrow \begin{cases} h = h_p = h_u \\ M = M_s + M_u \\ c_p = c_u = 0 \end{cases} \quad (7)$$

Then, the parameters given in Eq. (5) become in M_2 :

$$\begin{cases} A_h = Mh, A_c = 0 \\ A_{yz} = I_y - I_z \\ A_{zx} = I_z - I_x \end{cases} \quad \text{and} \quad \begin{cases} I_x = I_{xs} + Mh^2 \\ I_y = I_{ys} + Mh^2 \\ I_z = I_{zs} \end{cases} \quad (8)$$

So, we obtain:

Fig. 3 Heavy vehicle structure for model M_2 .

$$\begin{cases}
M\dot{V}_x = M(V_\psi V_y - hV_\psi V_\theta - h\dot{V}_\phi) \\
\quad + F_{xf} \cos \delta_f + F_{xr} \cos \delta_r - F_{yf} \sin \delta_f \\
\quad - F_{yr} \sin \delta_r - k_{ax} V_x^2 \\
M\dot{V}_y = -MV_\psi V_x + M_s h \dot{V}_\theta \\
\quad + F_{xf} \sin \delta_f + F_{xr} \sin \delta_r + F_{yf} \cos \delta_f \\
\quad + F_{yr} \cos \delta_r - k_{ay} V_y^2 \\
M_s \dot{V}_z = M_s g - (K_f + K_r)z - (B_f + B_r)V_z \\
\quad + (aK_f - bK_r)\phi + (aB_f - bB_r)V_\phi \\
\quad - M_s \phi \dot{V}_x \\
I_x \dot{V}_\theta = M_s h_p (\dot{V}_y + V_\psi V_x) + A_{yz} V_\psi V_\phi + A_{yz} V_\psi V_\phi \\
\quad + I_{xz} \dot{V}_\psi - (l^2 K_f + l^2 K_r + B_{af} + B_{ar})\theta \\
\quad - (l^2 B_f + l^2 B_r)V_\theta - M_s g h \theta \\
\quad - (a^2 K_f + b^2 K_r - M_s g h)\phi \\
\quad - (a^2 B_f + b^2 B_r)V_\phi \\
I_y \dot{V}_\phi = -A_h (\dot{V}_x - V_\psi V_y) + A_{zx} V_\psi V_\theta \\
\quad - (a^2 K_f + b^2 K_r - M_s g h)\phi \\
\quad - (a^2 B_f + b^2 B_r)V_f + (aB_f - bB_r)V_z \\
\quad + (aK_f - bK_r)z \\
I_z \dot{V}_\psi = A_{xy} V_\theta V_\phi + a(F_{xf} \sin \delta_f + F_{yf} \cos \delta_f) \\
\quad - b(F_{xr} \sin \delta_r + F_{yr} \cos \delta_r) \\
\quad + l(\Delta F_{xf} \cos \delta_f - \Delta F_{yf} \sin \delta_f) \\
\quad + l(\Delta F_{xr} \cos \delta_r - \Delta F_{yr} \sin \delta_r) + I_{xz} \dot{V}_\theta
\end{cases} \quad (9)$$

3. Results and Discussion

3.1 Vehicle Model M_1 Validation

The validity of the nonlinear vehicle models is examined using the available measured data reported in Ref. [18] under slalom maneuvers and with a constant speed. The relative performance of vehicle models M_1 and M_2 is further evaluated in terms of responses to acceleration/braking and steering inputs and with the vehicle speeds: 30, 60 and 90 km·h⁻¹. The number of cones used for the test is four cones and the distance between cones is $d = 40$ m. The maximum lateral

displacement is $y_{\max} = 3.5$ m. The model M_1 simulations are performed with a front steering angle $\delta_f = 8^\circ$. Figs. 4a-4c present comparisons of the model M_1 responses to a slalom maneuvers in terms of longitudinal acceleration, lateral acceleration and yaw rate with the reported measured data. However, some differences between the simulation and measured data can also be observed. This may be attributed partially to the parameters and variables neglected in the modeling (six-DOFs), differences in the tires properties considered in the simulation model, inertias of sprung and unsprung mass and steering inputs used in the simulation model. The comparisons show good agreements between the simulation model responses and the measured data. So, the vehicle model M_1 obtained is good under considered slalom maneuvers.

3.2 Comparisons of Two Models M_1 and M_2

To compare the performance of the proposed two nonlinear vehicle models M_1 and M_2 , different simulations are performed under slalom maneuvers and with vehicle speed 60 km·h⁻¹.

Fig. 5 compares respectively the longitudinal velocity V_x and the position of the vehicle in both models M_1 and M_2 . The results show that the responses of longitudinal velocity and the vehicle position in the earth fixed frame (0, x, y) have lower magnitudes in model M_1 than in model M_2 .

Fig. 6 presents comparisons of the two models responses to slalom maneuver in term of roll and pitch angles and with vehicle speed 60 km·h⁻¹.

The results clearly show that the roll response of the model M_1 is considerably lower compared with that of M_2 . This is partially attributed to the higher roll moments of inertia and the added parameters in vehicle model M_1 as shown in Eq. (5). The peak pitch angle response of the first model M_1 is considerably lower compared with that of the model M_2 .

Fig. 7 illustrates comparisons of the roll and pitch angles values of the vehicle models M_1 and M_2 with three different speeds (30, 60 and

Fig. 4 Comparison of vehicle model M_1 during slalom maneuvering with the measured data reported in Ref. [18]: (a) longitudinal acceleration; (b) lateral acceleration and (c) yaw rate.

Fig. 5 (a) Longitudinal velocity in models M_1 and M_2 and (b) vehicle position in the earth fixed frame.

Fig. 6 (a) Roll angle and (b) pitch angle.

Fig. 7 Comparisons of the roll and pitch angles in models M_1 and M_2 under different speeds.

90 km·h⁻¹). The results show that the roll and pitch values of the model M_1 are considerably lower compared with those of the model M_2 . The difference between the two models in terms of roll and pitch angles is relatively large when the vehicle speed increases. This may be partially attributed to the dynamic lateral and longitudinal load transfers which depend on the vehicle speed [13]. This means that the added parameters in the vehicle model M_1

tend to decrease the impacts of the load transfers and thus the magnitudes of vehicle roll and/or pitch motions.

4. Conclusions

This study investigated the potential benefits of such practical model M_1 on vehicle ride, handling, roll and pitch dynamic responses under various speeds and steering maneuvers. The dynamic responses of the proposed model M_1 are evaluated using generalized six-DOF nonlinear models. The roll, pitch, longitudinal and lateral dynamic performance are assessed under critical handling maneuvers. The simulation results show that the proposed model M_1 could yield considerably enhanced roll, pitch and yaw dynamics, handling performance as well as improved directional and stability driving of vehicle behavior. The practical benefits of the proposed model which takes CG of sprung mass, unsprung mass and total vehicle mass into account were further shown. The comparisons results of the proposed nonlinear vehicle models M_1 with the nonlinear model M_2 clearly demonstrate the potential advantages of model M_1 compared to M_2 in enhancing overall vehicle in terms of roll, pitch and directional responses.

References

- [1] Abbassi, Y., Ait-Amirat, Y., and Outbib, R. 2014. "Nonlinear Feedback Control and Trajectory Tracking of Vehicle." *International Journal of Systems Science* doi:10.1080/00207721.2014.880195.
- [2] Cao, D. P., Rakheja, S., and Su, C. Y. 2010. "Roll- and Pitch-Plane-Coupled Hydro-pneumatic Suspension. Part 2: Dynamic Response Analyses." *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 48 (4): 507-28.
- [3] Ackermann, J. 1994. "Robust Decoupling, Ideal Steering Dynamics and Yaw Stabilization of 4WS Car." *Automatica* 30 (11): 1761-8.
- [4] Abbassi, Y., Amirat, Y. A., and Outbib, R. 2011. "Nonlinear Feedback Control of Vehicle Speed." In *Proceedings of the 18th IFAC (International Federation of Accountants) World Congress*, 6279-84.
- [5] Sayers, M. W. 1990. "Symbolic Computer Methods to Automatically Formulate Vehicle." Ph.D. thesis, University of Michigan.
- [6] Heydinger, G., Bixel, R., Durisek, N., Yu, E., and Guenther, D. 1998. "Effects of Loading on Vehicle Handling." Presented at the SAE (Society of Automotive Engineers) International Congress and Exposition, Michigan, USA.
- [7] Whitehead, R., Travis, W., Bevely, D., and Flowers, G. 2004. "A Study of the Effects of Various Vehicle Properties on Rollover Propensity." Presented at the SAE Automotive Dynamics, Stability & Controls Conference and Exhibition, Michigan, USA.
- [8] Hac, A. 2005. "Effects of Brake Actuator Error on Vehicle Dynamics and Stability." Presented at the SAE 2005 World Congress & Exhibition, Michigan, USA.
- [9] Hegazy, S., Rahnejat, H., and Hussain, K. 2000. "Multi-body Dynamics in Full Vehicle Handling Analysis under Transient Maneuver." *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 34 (1): 1-24.
- [10] Cao, D. P., Rakheja, S., and Su, C. Y. 2010. "Roll- and Pitch-Plane Coupled Hydro-pneumatic Suspension." *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 48 (3): 361-86.
- [11] Shim, T., and Ghike, C. 2007. "Understanding the Limitations of Different Vehicle Models for Roll Dynamics Studies." *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility* 45 (3): 191-216.
- [12] Abbassi, Y., Ait-Amirat, Y., and Outbib, R. 2007. "A Global Model for Vehicle Dynamics." In *Proceeding of AVCS (Advanced Vehicle Control Systems)*, 285-90.
- [13] Pacejka, H. B. 2005. *Tire and Vehicle Dynamics*. 2nd ed.. United Kingdom: Butterworth-Heinemann.
- [14] Wong, J. Y. 2006. *Theory of Ground Vehicles*. 3rd ed.. New York: Wiley.
- [15] Klasson, J. 2002. "A Generalized Crosswind Model for Vehicle Simulation Purposes." *Vehicle System Dynamics Supplement* 37: 350-9.
- [16] Novel, B., and Chou, H. 2005. "Global Vehicle Control Using Differential Braking Torques and Active Suspension Forces." *Vehicle System Dynamics* 43 (4): 261-84.
- [17] Li, L., Wang, F. Y., and Zhou, Q. Z. 2006. "Integrated Longitudinal and Lateral Tire/Road Friction Modeling and Monitoring for Vehicle Motion Control." *IEEE Transactions on Intelligent Transportation Systems* 7 (1): 1-19.
- [18] Bernard, J. E., Winkler, C. B., and Fancher, P. S. 1973. *A Computer Based Mathematical Method for Predicting the Directional Response of Trucks and Tractor-Trailers*. UMTRI technical report UM-HSRI-PF-73-1, The University of Michigan, USA.