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# An Electric Simulator of a Vehicle Transmission Chain Coupled to a Vehicle Dynamic Model

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## 1. Introduction

During the two past decades, important vehicle simulators have been developed. The evolution of these simulation tools has attracted the attention of several industrials. The aim of the concept is to seek about effective methods and accurate models which allow to reach this objective and to minimize the cost and the time devoted to the development phases of vehicle systems (Kiencke & Nielsen, 2005), (Pill-Soo, 2003), (Deuszkiewicz & Radkowski, 2003).
With the vehicle simulators, the users can simulate the driving vehicle or new vehicle safety component. It offers several benefits for a designing and comprehension of the vehicle behaviours in order to improve the passenger's safety. Also the interaction of the driver, vehicle and road (environment) is studied with the help of vehicle simulators (Donghoon & Kyongsu, 2006), (Larouci et al, 2006), (Larouci et al, 2007).
The aim of this chapter is to present a method to carry out a vehicle transmission simulator coupled to a vehicle dynamic model. The transmission simulator uses electric actuators with dedicated control laws to reproduce the mechanical characteristic of the real vehicle transmission chain. The vehicle dynamic model takes into account the longitudinal, the vertical and the pitch motions. The coupled electric simulator validates the theoretical studies (automatic gearbox, test of heat engine, dynamic behaviour, passenger comfort, automated driving...) by measurements without need to the real transmission system and the real environment of the vehicle. Such a method allows to reduce significantly the cost and the time of the development phases of vehicles.
The present work is organized as follows. In the first part, a vehicle transmission simulator and vehicle dynamic model are developed. The second part focuses on the decoupled transmission simulator and the vehicle dynamic. Then a coupling approach of the previous models will be shown. The control performances of the electric simulator part depend on the electric actuator parameters which can be change under the vehicle environmental constraints (temperature, vibration…). In order to overcome these drawbacks and to improve the control law robustness of the electric simulator a sliding mode control will be proposed.
Finally, a comparison between the vehicle dynamic performances obtained using the coupled and decoupled models will be presented and discussed

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## 2. The vehicle transmission system simulator

The electric simulator of the vehicle transmission chain simulates the mechanical characteristic of the transmission system. This simulator uses two electric actuators controlled with dedicated control laws. The first one reproduces the dynamic driving torque developed by the heat engine and available at the output of the bridge, while the second one simulates the resisting torque imposed by the vehicle load (the whole resisting efforts to the vehicle advance plus inertias).

### 2.1 Modeling of the real vehicle transmission system

Figure 1 illustrates the various forces applied to a vehicle during its motion on a road with a slope of angle $\alpha$. These forces include the driving force and the mean resisting forces.

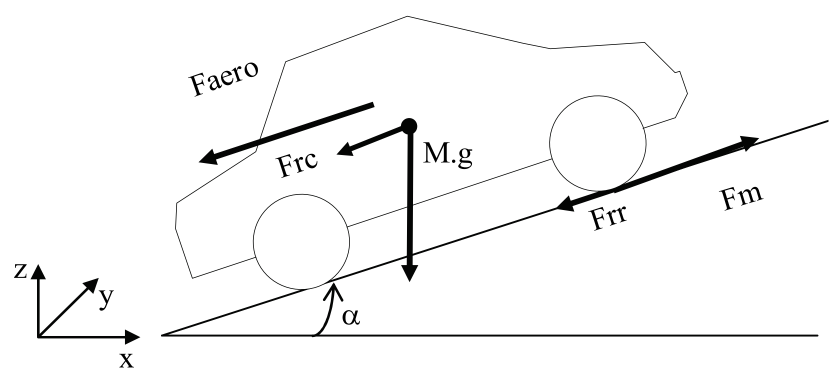


Fig. 1. Forces applied to a vehicle in a slope

Fm, Faero, Frr and Frc are, respectively, the driving force, the aerodynamics force, the rolling friction force and the resisting force in a slope, (Bauer, 2005), (Minakawa et al., 1999). To model the real vehicle transmission system, we suppose that the transmission losses are neglected (the efficiency of clutch and gear box reaches 1) and only longitudinal forces are considered (Liang et al., 2003), (Nakamura et al., 2003), (Sawas et al., 1999), (Krick, 1976). Using these assumptions, the following equations can be written:

#### 2.1.1 According to the heat engine

$$\left(J_{th} + J_{eb}\right) \cdot \frac{d\Omega_{th}}{dt} = C_{m\_th} - C_{r\_eb} \tag{1}$$

$J_{th}$ and $J_{eb}$ are, respectively, the inertias of the heat engine and the input shaft of the gearbox. $\Omega_{th}$ is the angular speed of the heat engine. $C_{m\_th}$ and $C_{r\_eb}$ are, respectively, the heat engine torque and the resisting torque (the resisting torque at the input of the gearbox seen by the heat engine) (see figure 2).

#### 2.1.2 According to the bridge

$$\left(J_{sp} + J_{roues}\right) \cdot \frac{d\Omega_{sp}}{dt} = C_{m\_sp} - C_{r\_roues} \tag{2}$$

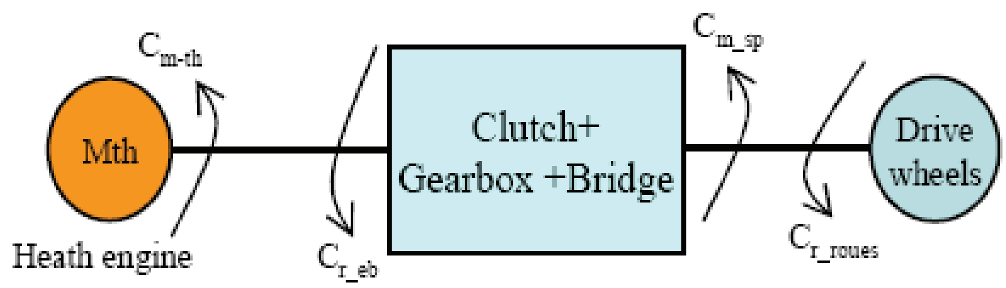


Fig. 2. Simplified transmission system

$J_{sp}$ and $J_{roues}$ are, respectively, the inertia at the output of the bridge and the inertia of the wheels. $\Omega_{sp}$ is the angular speed at the output of the bridge. $C_{m\_sp}$ and $C_{r\_roues}$ are, respectively, the torque at the output of the bridge and the resisting torque at the wheels.

### 2.1.3 According to the centre of gravity of the vehicle

$$M \cdot \frac{dV}{dt} = F_m - F_{aero} - F_{rr} - F_{rc} \tag{3}$$

$$\begin{cases} F_{aero} = \frac{1}{2} \cdot \rho \cdot C_x \cdot S_f \cdot V^2 \\ F_{rr} = f_{rr} \cdot M \cdot g \cdot \cos(\alpha) \\ F_{rc} = M \cdot g \cdot \sin(\alpha) \\ \Omega_{th} = \Omega_{sp} \cdot R_t \\ V = \Omega_{sp} \cdot R_{sc} \end{cases}$$

| | | |
|---|---|---|
| M | the total vehicle mass | kg |
| V | the vehicle longitudinal speed | m/s |
| ρ | density of the air | kg/m³ |
| $C_x$ | the drag coefficient | |
| $S_f$ | the frontal (transverse) section of the vehicle | m² |
| $f_{rr}$ | the coefficient of rolling friction | |
| g | the acceleration of gravity | 9.81 m/s² |
| α | the slope angle | rad |
| $R_{sc}$ | loaded radius (ray of the driving wheel) | m |
| $R_t=R_b.R_p$ | total reduction ratio | |
| $R_b$ | the gearbox ratio | |
| $R_p$ | the bridge ratio | |

Table 1. Nomenclature

The transmission is supposed without losses. So:

$$C_{r\_roues} = F_m \cdot R_{sc}$$

$$C_{m\_sp} = C_{r\_eb} \cdot R_t$$

From the equation 3, we deduce that:

$$\left(J_{sp} + J_{roues} + M \cdot R_{sc}^2\right) \cdot \frac{d\Omega_{sp}}{dt} = C_{m\_sp} - C_{sr\_sp} \tag{4}$$

Where:

$C_{sr\text{-}sp}$ is the total resisting torque in the steady state at the output of the bridge (5):

$$C_{sr\_sp} = \frac{1}{2} \cdot \rho \cdot C_x \cdot S_f \cdot R_{sc}^3 \cdot \Omega_{sp}^2 + M \cdot g \cdot R_{sc} \cdot \left[\sin(\alpha) + f_{rr} \cdot \cos(\alpha)\right] \tag{5}$$

### 2.2 Modeling of the equivalent system

In order to reproduce the behavior of the real vehicle transmission chain, an equivalent model using two electric actuators is considered (figure 3). In this model, the electric actuator M1 simulates the heat engine, while the second actuator (M2) simulates the resisting forces.

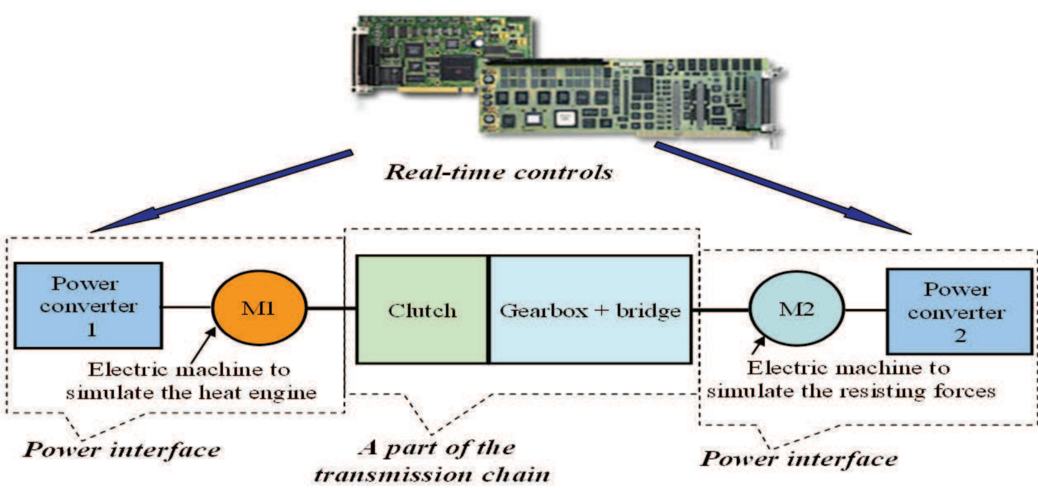


Fig. 3. A first equivalent model

In order to work in a reduced torque scale and to validate the coupling of a transmission model to a vehicle dynamic one, the previous configuration (figure 3) is reduced to the configuration presented in figure 4 where the actuator M2 simulates the whole resisting torque due to aerodynamic frictions, rolling frictions, resisting torque in a slope and inertia with a torque reduction factor (fc2). However, the electric actuator M1 simulates both the heat engine and the gearbox with a torque reduction factor (fc1).

This model can be used to test control strategies of automatic gearbox and to study the influence of these strategies on the vehicle dynamic behavior in order to improve the passenger comfort for example.

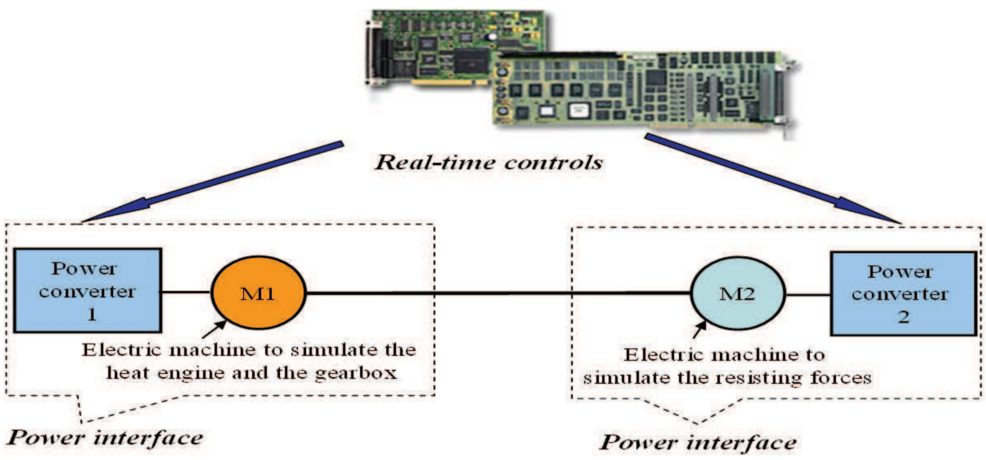


Fig. 4. A second equivalent model

Considering fc2 = fc1= fc yields:

$$\Omega_1 = \Omega_2 = \Omega_{sp} = \frac{\Omega_{th}}{R_t} \text{ and } C_{m\_1} = \frac{C_{m-sp}}{fc}$$

$\Omega_1$ and $\Omega_2$ are the angular velocities of the electric actuators M1 and M2. Therefore, the equation 2 can be written as follows:

$$(J_{sp} + J_{roues}) \cdot \frac{1}{fc} \cdot \frac{d\Omega_1}{dt} = C_{m\_1} - \frac{1}{fc} \cdot C_{r\_roues} \tag{6}$$

The mechanical equation on the common tree of the two electric actuators is:

$$(J_1 + J_2) \cdot \frac{d\Omega_1}{dt} = C_{m\_1} - C_{r\_2} \tag{7}$$

Where:

$J_1$ and $J_2$ are the moment of inertia of the actuators M1 and M2. $C_{m\_1}$ and $C_{r\_2}$ are the torques of the actuators M1 and M2 respectively.

### 2.3 Torque control laws of the electric actuators

The resisting torque which must be developed by the actuator M2 ($C_{r\_2}$) is deduced from equations (6) and (7). So:

$$C_{r\_2} = \frac{1}{fc} \cdot C_{r\_roues} + \left[ \left( J_{sp} + J_{roues} \right) \cdot \frac{1}{fc} - (J_1 + J_2) \right] \cdot \frac{d\Omega_1}{dt} \tag{8}$$

Where:

$$C_{r\_roues} = M \cdot R_{sc}^2 \cdot \frac{d\Omega_{sp}}{dt} + C_{sr\_sp} = M \cdot R_{sc}^2 \cdot \frac{d\Omega_1}{dt} + C_{sr\_sp}$$

$C_{sr\text{-}sp}$ is the total resisting torque in the steady state given by equation 5. So:

$$C_{r\_2} = \frac{1}{fc}\cdot\left[\frac{1}{2}\cdot\rho\cdot C_x\cdot S_f\cdot R_{sc}^3\cdot\Omega_1^2 + M\cdot g\cdot R_{sc}\cdot\left(\sin(\alpha)+\cos(\alpha)\right)\right] + \\ +\left[\left(J_{sp}+J_{roues}+M\cdot R_{sc}^2\right)\cdot\frac{1}{fc} - \left(J_1+J_2\right)\right]\cdot\frac{d\Omega_1}{dt} \tag{9}$$

In the same way, the torque which must be developed by the actuator M1 ($C_{m\_1}$) is deduced from equations (1) and (7). So:

$$C_{m\_1} = \frac{R_t}{fc}\cdot\left[C_{m\_th} - R_t\cdot\left(J_{th}+J_{eb}\right)\cdot\frac{d\Omega_1}{dt}\right] \tag{10}$$

As a result the torque control laws of the actuators M1 and M2 ($C_{m\_1\_ref}$ and $C_{r\_2\_ref}$ ) are expressed as follows:

$$C_{m\_1\_ref} = \frac{R_t}{fc}\cdot\left[C_{m\_th} - R_t\cdot\left(J_{th}+J_{eb}\right)\cdot\frac{d\Omega_1}{dt}\right] + f_{v1}\cdot\Omega_1 + C_{s1} \tag{11}$$

$$C_{r\_2\_ref} = \frac{1}{fc}\cdot\left[\frac{1}{2}\cdot\rho\cdot C_x\cdot S_f\cdot R_{sc}^3\cdot\Omega_1^2 + M\cdot g\cdot R_{sc}\cdot\left(\sin(\alpha)+\cos(\alpha)\right)\right] \\ +\left[\left(J_{sp}+J_{roues}+M\cdot R_{sc}^2\right)\cdot\frac{1}{fc} - \left(J_1+J_2\right)\right]\cdot\frac{d\Omega_1}{dt} - f_{v2}\cdot\Omega_1 - C_{s2} \tag{12}$$

$f_{v1}$ and $f_{v2}$ are the viscous friction coefficients of the actuators M1 and M2.
$Cs_1$ and $Cs_2$ are the torques induced by the dry frictions of the two electric actuators.
These control laws compensate the losses induced by viscous and dry frictions of the two electric actuators.

### 2.4 Simulation results

A speed regulator is included in the simulation model. It determines the position of the accelerator pedal which allows to track a desired speed.
A vehicle starting test is carried out to validate the modeling of the equivalent transmission chain. It consists to evaluate the time necessary to reach a vehicle desired speed of 90 km/h (figure 5).

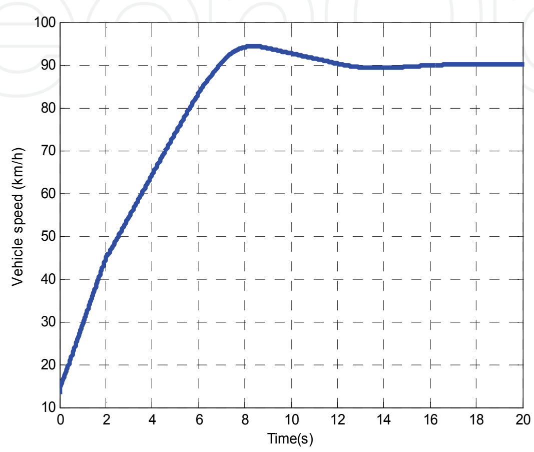


Fig. 5. A vehicle starting test

The regulator parameters are adjusted to obtain a vehicle starting time (12s in figure 5) close to the starting time given by the manufacturer (11.6s), (Grunn & Pham, 2007).
Figure 6 presents the desired torque and the real one developed by the electric actuator M1 in a case of a road profile characterized by different slopes (figure 7). This torque is the image of the torque available at the output of the bridge (with a reduction coefficient fc = 100). As a result, the real torque is very close to the desired one. Moreover, this torque is more important at the vehicle starting and in front of slopes to overcome the vehicle inertia and the resisting torque caused by these slopes.

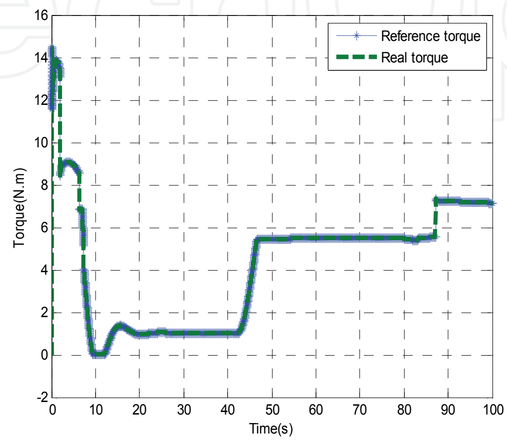


Fig. 6. Real and reference (desired) torques of the machine M1

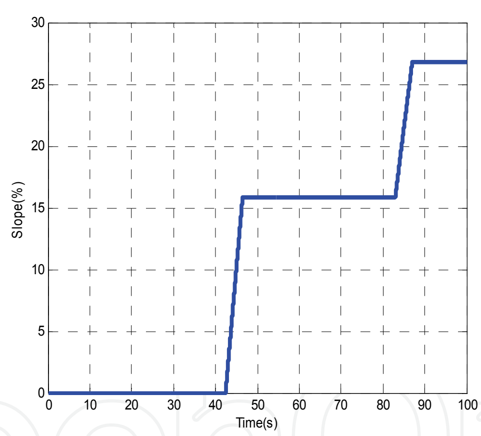


Fig. 7. Slopes of the considered road profile

## 3. The vehicle dynamic model

In this part a three degree-of-freedom vehicle dynamic model is presented. It takes into account the longitudinal, the vertical and the pitch motions of a vehicle. In this model, the yaw, the roll and the transversal motions are ignored. Only translations according to the longitudinal (x) and vertical (z) directions and the pitch rotation are considered.
Under these assumptions, the overall motion of the vehicle can be described by three equations (13). The first one characterizes the longitudinal dynamic. The second one represents the dynamics of the vertical motion and the latest describes the pitch motion, (Grunn & Pham, 2007).

$$\begin{cases} \dot{V} = \dfrac{F_x - m \cdot h \cdot \ddot{\varphi}}{M} \\ \ddot{Z} = \dfrac{F_z}{m} \\ \ddot{\phi} = \dfrac{M \cdot M_y - m \cdot h \cdot F_x}{M \cdot (I_y + m \cdot h^2) - (m \cdot h)^2} \end{cases} \tag{13}$$

where:
M: the total vehicle mass
m: the sprung mass
h: the vertical distance between the vehicle centre gravity and the pitch centre
$I_y$: the moment of inertia according to the y-axis
In this model, the resulting forces $F_x$ controls the longitudinal dynamics, (Pacejka, 2005). The vertical motion is controlled by a resulting force $F_z$ and the pitch motion is controlled by the pitch moment $M_y$. In this vehicle dynamic model (decoupled model), the transmission system is modeled by a gain and the driving torque is supposed proportional to the position of the accelerator pedal.

## 4. Coupling of the transmission of the simulator to the vehicle dynamic model

The coupled model (figure8) associates the transmission simulator and the vehicle dynamic model by replacing the transmission part of the vehicle dynamic model by the transmission simulator presented in second part. In this case, the resisting torque which must be developed by the actuator M2 takes into account the pitch effect. This torque is the same one calculated in the vehicle dynamic model plus the viscous and dry frictions of the actuator M2.

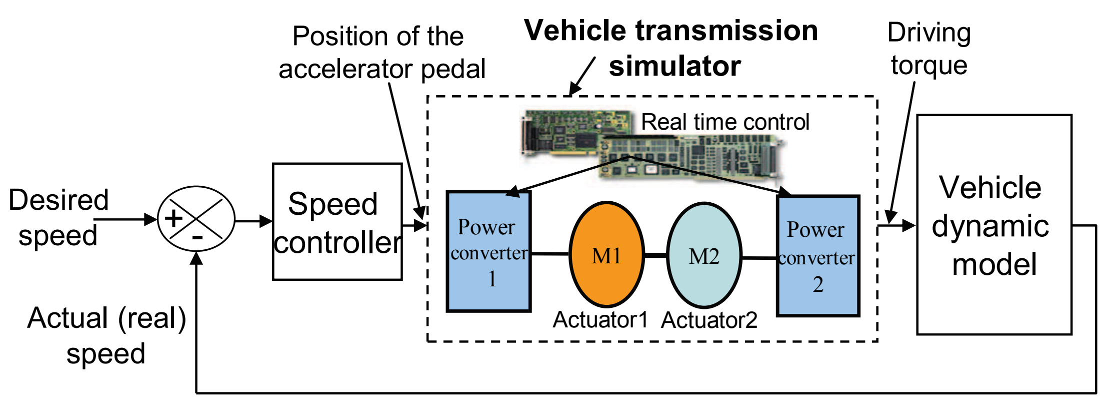


Fig. 8. Block diagram of the coupled model

The control performances of the electric simulator depend on the electric actuator parameters which can be change under the vehicle environmental constraints (temperature, vibration…). In order to eliminate this problem and to improve the control law robustness of the electric simulator a sliding mode control is used.

### 4.1 Sliding mode control law

In this part we develop a first sliding mode control law. The DC machine is described by the following equation:

$$U(t) = L\frac{dI(t)}{dt} + RI(t) + K_m\Omega(t) \tag{14}$$

U is the supply voltage. L,R,$K_m$ and $\Omega$ are respectively the inductance, the resistance the torque coefficient and the velocity of the DC machine.
First, the sliding surface S is chosen as follows:

$$S = \xi_1 + c_1\int_0^t \xi_1 d\tau \tag{15}$$

$c_1$ is a control parameter.
$\xi_1$ is the error between the real courant (I) and the desired one ($I_{des}$):

$$\xi_1 = I - I_{des} \tag{16}$$

The equivalent control input is first computed from $\dot{S} = 0$ .

$$\dot{S} = -\frac{R}{L}I - \frac{E}{L} + \frac{1}{L}U + c_1\xi_1 - \dot{I}_{des} = G + BU \tag{17}$$

Where:

$$\begin{cases} G = -\dfrac{R}{L}I - \dfrac{E}{L} + c_1\xi - \dot{I}_{des} \\ B = \dfrac{1}{L} \end{cases}$$

The equivalent control input is thus ($U_{eq}$):

$$U_{eq} = -\frac{G}{B}$$

By choosing a constant and proportional approach, we finally obtains:

$$U = U_{eq} - K_1 sign(S) - K_2(S) \tag{18}$$

$$sign(S) = \begin{cases} 1 & if\ S > 0 \\ -1 & if\ S < 0 \\ 0 & if\ S = 0 \end{cases}$$

$K_1$ and $K_2$ are control parameters. When the system is far from the sliding manifold, the behaviour is dominated by $K_2$ term, however $K_1$ term becomes dominant when approaching the manifold. A good choice of $K_1$ and $K_2$ will allows to reduce both the convergence time and the well-known chattering phenomena near the sliding manifold (Chaibet et al, 2004). In our case $K_1$=0.05, $K_2$=1.

### 4.2 Simulation results

Different simulations are presented to compare the dynamic performances of the coupled model and the decoupled one for a vehicle desired speed vdes = 60km/h on a straight road. The figures 9 and 10 show the vehicle speed and the longitudinal acceleration for the both models (coupled and decoupled models).

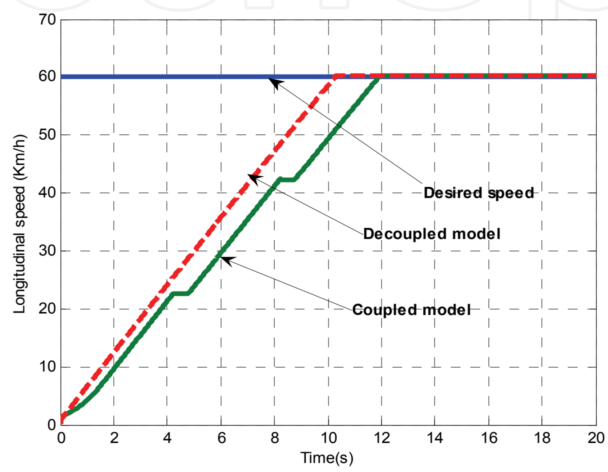


Fig. 9. Longitudinal vehicle speed

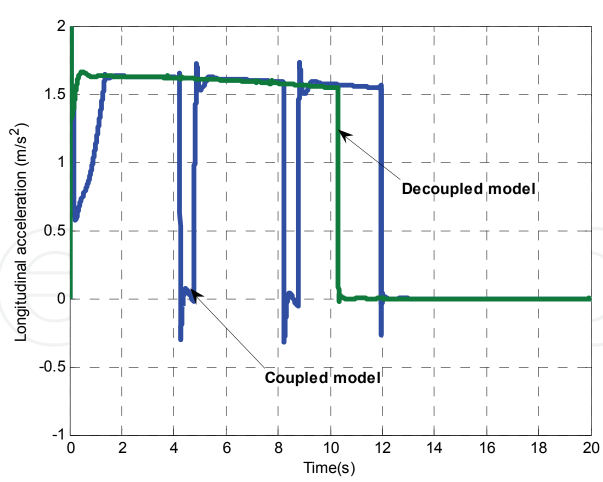


Fig. 10. Longitudinal acceleration

As results, the desired speed is reached more quickly in the case of the decoupled model. This difference is due to the time delay caused by the change of the commuted speeds. In addition and for the same reason, important variations on the longitudinal acceleration of the coupled model are detected.

Note that these variations (-0.3m/s² to 2m/s²) respect the passenger comfort limits (Chaibet et al, 2005), (Nouveliere & Mammar, 2003), (Huang & Renal, 1999).
The figures 11, 12 and 13 show respectively, the vertical acceleration of the sprung mass, its vertical movement and the pitch motion.
We note that the important variations obtained in the case of the coupled model are induced by the change of the commuted speeds.
Through these results, we can deduce that the integrating of the transmission chain simulator in the vehicle dynamic model allows to detect more dynamic variations and to reflect the vehicle dynamic behavior with high accuracy.

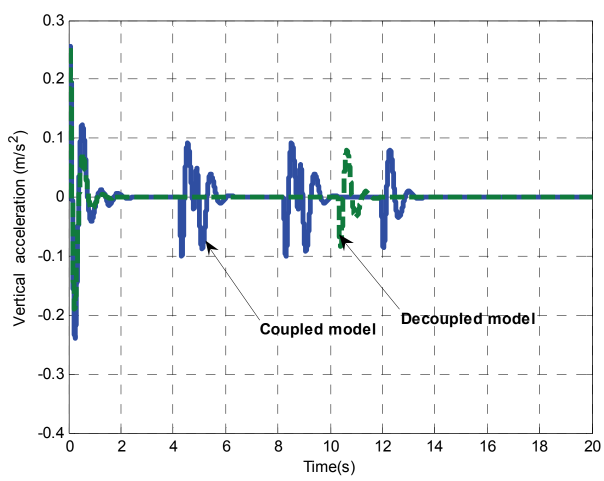


Fig. 11. Vertical acceleration of the sprung mass

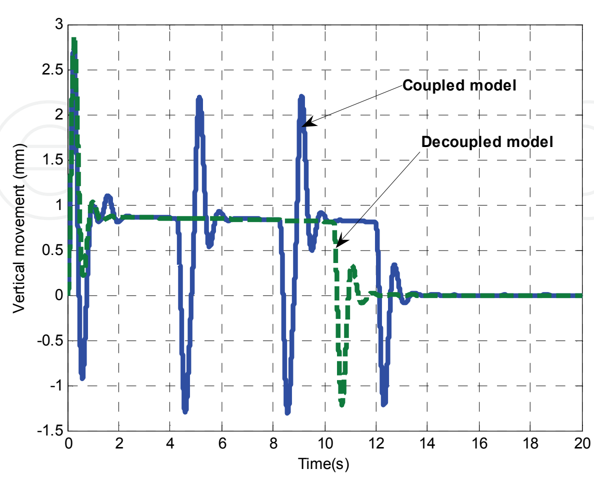


Fig. 12. Vertical movement of the sprung mass

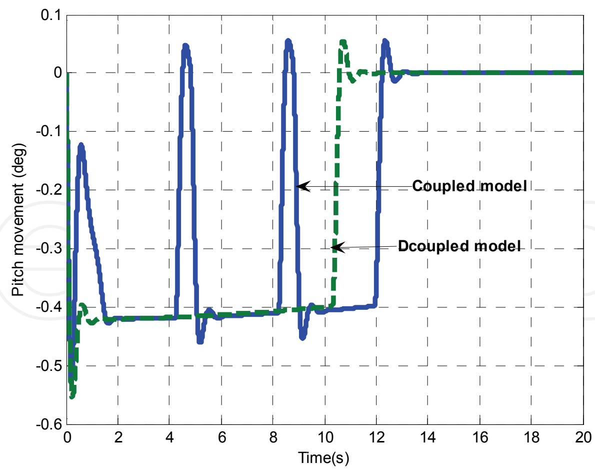


Fig. 13. Pitch movement

## 5. Conclusion

An electric simulator of the vehicle transmission chain coupled with a vehicle dynamic model is presented in this chapter. The transmission simulator uses two electric actuators with speed and torque control. The first actuator simulates both the heat engine and the gearbox. The second one simulates the forces resisting to the vehicle advance as well as the inertias.

The dynamic vehicle model includes longitudinal, vertical and pitch motions. The coupled model represents the vehicle dynamic behavior with high accuracy. This model is an interesting solution to carry out studies on transmission and vehicle dynamic aspects (development of control strategies of automatic gearbox by taking into account the dynamic behavior, improvement of safety and passenger comfort, test of intelligent vehicle…) without need to the real transmission system and the real environment of the vehicle. Therefore, it allows to reduce significantly the time and the cost of the development phases of the transmission and dynamic behavior systems.

## 6. References

Kiencke, U.; Nielsen, L. (2005). *Automotive Control Systems For Engine, Driveline and Vehicle*, Springer, ISBN 3-540-23139-0, Berlin

Pill-Soo, K. (2003). Cost modeling of battery electric vehicle and hybrid electric vehicle based on major parts cost, *Power Electronics and Drive Systems, PEDS 2003,* pp. 1295- 1300, ISBN 0-7803-7885-7, Singapore, 17-20 Nov. 2003

Deuszkiewicz, P.; Radkowski, S. (2003). On-line condition monitoring of a power transmission unit of a rail vehicle, *Mechanical Systems and Signal Processing journal*, Vol., 17, No., 6, (November 2003), page numbers (1321-1334)

Donghoon, H.; Kyongsu, Y. (2006). Evaluation of Adaptive Cruise Control Algorithms on a Virtual Test Track, *Proceedings of American control conference,* pp. 5849-5854, ISBN 1-4244-0209-3, Minneapolis, Minnesota, USA, June 14-16, 2006

Larouci, C.; Feld, G & Didier, Jp. (2006). Modeling and control of the vehicle transmission chain using electric actuators, *IEEE Industrial Electronics, IECON 2006,*pp. 4066-4071, ISBN 1-4244-0390-1, Paris, France, November 7-10-2006

Larouci, C.; Dehondt, E.; Harakat, A & Feld, G. (2007). Modeling and Control of the Vehicle Transmission System Using Electric Actuators; Integration of a Clutch, *IEEE International Symposium on Industrial Electronics ISIE,* pp. 2202-2207, ISBN 978-1-4244-0755-2, Vigo, Spain, June 04-07, 2007

Bauer, S. (2005). *Mémento de technologie automobile*, Bosch Edition, page numbers (1-961), ISBN 3934584195, 9783934584198, Germany

Minakawa, M.; Nakahara, J.; Ninomiya, J & Orimoto, Y. (1999). Method for measuring force transmitted from road surface to tires and its applications, *JSAE Review*, Vol., 20, No., 4, (October 1999), page numbers (479-485)

Liang, H.; To Chong, K.; Soo No, T & Yi, S.Y. (2003).Vehicle longitudinal brake control using variable parameter sliding control, *Control Engineering Practice*, Vol.,11, No., 4, (April 2003), page numbers (403-411)

Nakamura, K.; Kosaka, H.; Kadota, K.; & Shimizu, K. (2003). Development of a motor-assisted 4WD system for small front-wheel-drive vehicles, *JSAE Review*, Vol., 24, No., 4, (October 2003), page numbers (417-424)

Sawase, K.; Sano, Y. (1999). Application of active yaw control to vehicle dynamics by utilizing driving/breaking force, *JSAE Review*, Vol.,20, No.,2, (April 1999), page numbers (289-295)

Krick,G .(1973). Behaviour of tyres driven in soft ground with side slip, *Journal of Terramechanics,* Vol., 9, No., 4, (1973), page numbers (9-30)

Grunn,E.; Pham,A .(2007). A 0 D Modelling for Automotive Dynamic, *Journal Européen des Systèmes Automatisés JESA,* Vol., 41, No., 1, (2007), page numbers (30 – 70), France

Pacejka, H.B. (2005). *Tyre and vehicle dynamics*, Publisher Butterworth-Heinemann, page numbers (1-672), ISBN-13/EAN: 9780750669184

Chaibet,A.; Nouveliere,L. ; Netoo .M & Mammar,S. (2004). Sliding mode Control for vehicle following at Low Speed, *IEEE International French-Speaking Conference on Automatics (CIFA),* Douz, Tunisia, November 2004

Chaibet, A.; Nouveliere, L.; Mammar, S.; Netto, M & Labayrade, R. (2005). Backstepping control synthesis for both longitudinal and lateral automated vehicle, *IEEE Intelligent Vehicle Symposium,* pp:42-47, ISBN 0-7803-8961-1, Las Vegas, USA 6 - 8 June 2005

Nouvelière,L.; Mammar,S.(2003). Experimental vehicle longitudinal control using second order sliding modes, *American control conference,* pp.4705-4710, ISBN 0-7803-7896-2, Denver, Colorado USA, June 4 -6, 2003

Huang,S. Ren,W.(1999). Use of neural fuzzy networks with mixed genetic/ gradient algorithm in automated vehicle control, *IEEE TRANSACTIONS ON INDUSTRIAL ELECTRONICS*, Vol., 46, No., 6, page numbers (1090–1102), ISSN 0278-0046


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When talking about modelling it is natural to talk about simulation. Simulation is the imitation of the operation of a real-world process or systems over time. The objective is to generate a history of the model and the observation of that history helps us understand how the real-world system works, not necessarily involving the real-world into this process. A system (or process) model takes the form of a set of assumptions concerning its operation. In a model mathematical and logical assumptions are considered, and entities and their relationship are delimited. The objective of a model – and its respective simulation – is to answer a vast number of “what-if” questions. Some questions answered in this book are: What if the power distribution system does not work as expected? What if the produced ships were not able to transport all the demanded containers through the Yangtze River in China? And, what if an installed wind farm does not produce the expected amount of energyt? Answering these questions without a dynamic simulation model could be extremely expensive or even impossible in some cases and this book aims to present possible solutions to these problems.

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