

WHEEL AND HEAVY VEHICLE LOAD ESTIMATION USING ROAD PROFILE

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ABSTRACT

Vehicle dynamics are dependent on tire/road contact forces and torques, which are themselves dependent on wheels and vehicle loads. These loads induce normal forces which are very important to calculate road and bridge damage or to assess the LTR (Load Transfer Ratio) used to prevent or forecast rollover of the vehicles. A good evaluation of these loads and normal forces requires an accurate description or modelling of the road profile, which is an input to the heavy vehicle dynamic model.

In this paper, we study the influence of the road profile elevation on the wheel and vehicle loads. Several road profiles are measured and applied as an input to the heavy dynamic model developed in LCPC ([7]). Then the corresponding wheel loads are evaluated on each wheel. The estimation is done using sliding mode observers.

Simulation results from the PROSPER simulator are presented on the estimation of vertical forces, and discussed to evaluate the robustness of our approach and to assess the impact of the road profile.

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1 INTRODUCTION

Tire forces affect vehicle dynamic performance and behavior properties. Thus it is necessary to take into account the contact forces characteristics for vehicles and road safety analysis.

The handling, and the comfort of a vehicle depend on the vehicle's vertical movement. Among the various generalized forces of the vehicle, the vertical forces are mainly at the origin of this vertical movement ([5], [8]). These forces are very important to evaluate the risk of rollover. However, the existing sensors to measure these forces are very expensive and difficult to install. In this study, we propose an observer to estimate the heavy vehicle states and an estimator for vertical forces reconstruction ([6], [12]). The designed observer is based on a sliding mode approach. The main contribution is the on-line estimation of the tire force needed for control of the vehicle. In this work, we deal with a heavy vehicle model coupled with an appropriate wheel road contact model in order to estimate vertical forces using sliding mode observers. Design of such observers requires a dynamic model of heavy vehicle. Then, in a first step, we built up a model for a heavy vehicle which has been validated with the comparison between estimated and measured dynamics response of PROSPER simulator developed by Sera-Cd (see ([10])).

This paper is organized as follows: section 2 deals with the vehicle description and modelling. The design of the observer and the estimation of the vehicle loads are presented in section 3. Some results about the states observation and the vertical forces estimation are presented in section 4. Finally, some remarks and perspectives are given in a concluding section.

2 HEAVY VEHICLE MODELING

Many studies deal with heavy vehicle modelling which represent a complex mechanical system with nonlinear features ([1], [4], [2]). The models used for heavy vehicles are very complicated. Consequently, it is relatively difficult to define the different parameters intervening in these models. The external and internal moments acting on the vehicle follow the axes: longitudinal, lateral, and vertical. In this paper, we consider the tractor-semitrailer model (with 2 axles for the trailer and 1 axle for the semitrailer) presented in the figure (1) which has 12 degrees of freedom (dof). This model is derived using Lagrangian's equations.

In this paper, we are interested in about 8 *dof*. Then, we can define a dynamic model of the vehicle as :

$$M(q) \ddot{q} + C(q, \dot{q}) \dot{q} + K(q) = F_g \quad (1)$$

where $M \in \mathfrak{R}^{8 \times 8}$ is the inertia matrix (Mass matrix), $C \in \mathfrak{R}^{8 \times 8}$ is related to the damping effects, $K \in \mathfrak{R}^8$ is the springs stiffness vector and $F_g \in \mathfrak{R}^8$ is a vector of generalized forces.

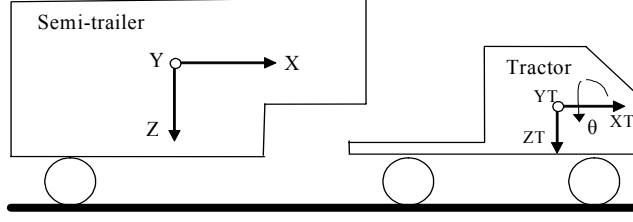


Figure 1: Heavy vehicle model

$q \in \mathfrak{R}^8$ is the coordinates vector defined by:

$$q = [q_1, q_2, q_3, q_4, q_5, q_6, z, \theta]^T \quad (2)$$

θ is the tractor roll angle;

q_1, q_2 are respectively the left and right front suspension deflection of the tractor;

q_3, q_4 are respectively the left and right rear suspension deflection of the tractor;

q_5, q_6 are respectively the left and right suspension deflection of the semi-trailer;

z is the vertical displacement of the tractor sprung mass (center of gravity height).

The suspension is modeled as the combination of a nonlinear spring and damper element as shown in the figure (2). The tractor chassis (with the mass M) is suspended on its axles through two suspension systems. We can also show that the tyre is modeled by springs (represented by k_i in the figure) and damper elements (represented by B_i in the figure). The wheel masses are represented by m_1 and m_2 . At the tyre contact, we have the road profile represented by the inputs u_1 and u_2 , which are considered as the heavy vehicle inputs. z_{r1} and z_{r2} represent

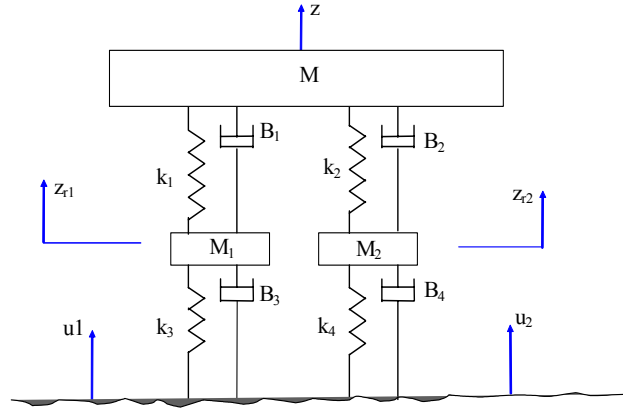


Figure 2: Suspension model

respectively the vertical displacement of the left and right wheel of the tractor front axle. These can be calculated using the following equations:

$$\begin{cases} z_{r1} = z - q_1 - \frac{T_w}{2} \sin(\theta) - r \cos(\theta) \\ z_{r2} = z + q_2 - \frac{T_w}{2} \sin(\theta) - r \cos(\theta) \end{cases} \quad (3)$$

where T_w is the tractor track width and r is the wheel radius.

The main purpose of a tire is to transmit forces between the road and the rim so that the driver can control the vehicle. A lot of studies have been done in the area of modeling tire [9].

The forces generated in the contact between the tires and the road are very important for the dynamic behavior of a road vehicle. Hence, accurate tire models are necessary components of vehicle models aimed at analyzing or simulating vehicle motion in real driving conditions. There are many previous models that describe the tire-forces. Some models are theoretical in the sense that they aim at modeling the physical processes that generate the forces. Other models are empirically oriented and aim at describing observed phenomena in a simple form.

The normal forces F_{ni} , $i = 1..6$ acting on the wheels are calculated using the following expression:

$$F_{ni} = F_{ci} + k_i(u_i - z_{ri}), \quad i = 1..6 \quad (4)$$

where F_{ci} is the static load.

By using the equations (3), we replace z_{ri} by their expression.

We obtain:

$$F_{ni} = F_{ci} + k_i(u_i - z + T_w \sin(\theta) + q_i), \quad i = 1..6 \quad (5)$$

In this study, we assume that the force generated by damping effect is neglected compared to the spring force.

3 VERTICAL LOADS ESTIMATION

In this section, we develop the sliding mode observers to estimate the vertical loads. We rewrite then the dynamic model (1) in the state form as follows:

$$\begin{cases} \dot{x} = f(x) + F_g \\ y = h(x) \end{cases} \quad (6)$$

where the state vector $x = (x_1, x_2)^T = (q, \dot{q})^T$, and $y = q$ is the vector of measured outputs of the system.

Thus, we obtain:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \ddot{q} = M^{-1}(F_g - C(x_1, x_2)x_2 - K(x_1)) \\ y = x_1 \end{cases} \quad (7)$$

Before developing the sliding mode observer, let us consider the following assumptions:

1. The state is bounded ($\|x(t)\| < \infty, \forall t \geq 0$).
2. The system is the inputs bounded (\exists a constant $\mu \in \mathfrak{R}$ such as: $\dot{u}_i < \mu$).
3. The generalized forces F_g are bounded (\exists a constant $\zeta \in \mathfrak{R}$ such as: $F_{gi} < \zeta$).
4. In the vector $C(x_1, x_2)x_2$, we find quadratics elements in x_2 . We can then bound this vector as:

$$\|C(x_1, x_2)x_2\| \leq c_1 \|x_2\|^2 \quad (8)$$

This last assumption is coming from mechanical and physical properties of our system and because of boundness of the real signals (position, speed, acceleration).

In the state form, we rewrite the equation 5 as follows:

$$F_{ni} = F_{ci} + f(x_1), \quad i = 1..6 \quad (9)$$

3.1 OBSERVER DESIGN

In order to estimate the state vector x and to deduce the vertical loads vector F_{ni} , we propose the following sliding mode observer:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + H_1 \text{sign}(\tilde{x}_1) \\ \dot{\hat{x}}_2 = M^{-1}(F_g - C(\hat{x}_1, \hat{x}_2)\hat{x}_2 - K(\hat{x}_1)) + \\ H_2 \text{sign}(\tilde{x}_1) \end{cases} \quad (10)$$

where $\tilde{x}_1 = x_1 - \hat{x}_1$ is the state estimation error. $H_1 \in \mathbb{R}^{8 \times 8}$ and $H_2 \in \mathbb{R}^{8 \times 8}$ represent positive diagonal gain matrices.

Let us now define another observer to estimate the vertical loads vector. It has the following form:

$$\hat{F}_{ni} = F_{ci} + f(\hat{x}_1) \quad (11)$$

3.2 CONVERGENCE ANALYSIS

We can write the dynamics estimation errors as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - H_1 \text{sign}(\tilde{x}_1) \\ \dot{\tilde{x}}_2 = M^{-1}\hat{F}_g - M^{-1}(C(x_1, x_2)x_2 - C(\hat{x}_1, \hat{x}_2)\hat{x}_2) \\ -M^{-1}(K(x_1) - K(\hat{x}_1)) - H_2 \text{sign}(\tilde{x}_1) \end{cases} \quad (12)$$

In order to study the observer stability and to find the gain matrices H_1 and H_2 , first we proceed to prove the convergence of \tilde{x}_1 to the sliding surface $\tilde{x}_1 = 0$, in finite time t_1 . Then, we deduce some conditions about \tilde{x}_2 to ensure its convergence towards 0.

Let us consider the following Lyapunov function:

$$V_1 = \frac{1}{2} \tilde{x}_1^T \tilde{x}_1 \quad (13)$$

The time derivative of this function is given by:

$$\dot{V}_1 = \tilde{x}_1^T (\tilde{x}_2 - H_1 \text{sign}(\tilde{x}_1)) \quad (14)$$

By considering gains matrix $H_1 = \text{diag}(h_{i1})$ with $h_{i1} > |\tilde{x}_{i2}|$, $i = 1 \dots 8$, then $\dot{V}_1 < 0$. Therefore, from sliding mode theory ([3], [11]), the surface defined by $\tilde{x}_1 = 0$ is attractive, leading \hat{x}_1 to converge towards x_1 in finite time t_0 .

Moreover, we have $\dot{\tilde{x}}_1 = 0 \forall t \geq t_0$. Consequently, according to equation system (12), we have (for $t \geq t_0$):

$$\text{sign}_{eq}(\tilde{x}_1) = H_1^{-1} \tilde{x}_2 \quad (15)$$

where $\tilde{x}_2 = x_2 - \hat{x}_2$ and sign_{eq} represents an equivalent form of the sign function on the sliding surface.

Then, equation system (12) can be written as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - H_1 \text{sign}_{eq}(\tilde{x}_1) \rightarrow 0 \\ \dot{\tilde{x}}_2 = M^{-1}\tilde{F}_g - M^{-1}(C(x_1, x_2)x_2 \\ -C(x_1, \hat{x}_2)\hat{x}_2) - H_2 H_1^{-1} \tilde{x}_2 \end{cases} \quad (16)$$

Now, let us consider a (second) Lyapunov function V_2 and its time derivative \dot{V}_2 :

$$\begin{aligned} V_2 &= \frac{1}{2} \tilde{x}_2^T M \tilde{x}_2 \\ \dot{V}_2 &= \tilde{x}_2^T M \dot{\tilde{x}}_2 \end{aligned} \quad (17)$$

Then, from equation (16), \dot{V}_2 becomes:

$$\begin{aligned} \dot{V}_2 &= \tilde{x}_2^T \tilde{F}_g - \tilde{x}_2^T (C(x_1, x_2)x_2 \\ &\quad - C(x_1, \hat{x}_2)\hat{x}_2) - \tilde{x}_2^T M H_2 H_1^{-1} \tilde{x}_2 \end{aligned} \quad (18)$$

Using the assumption 4 and from the equations (8) and (15), we have:

$$\begin{aligned} &C(x_1, x_2)x_2 - C(x_1, \hat{x}_2)\hat{x}_2 \\ &= -C(x_1, x_2)\tilde{x}_2 - C(x_1, \hat{x}_2)\tilde{x}_2 \end{aligned} \quad (19)$$

we obtain:

$$\begin{aligned} \dot{V}_2 &= \tilde{x}_2^T \tilde{F}_g - \tilde{x}_2^T (-C(x_1, x_2) + C(x_1, \hat{x}_2)) \\ &\quad + M H_2 H_1^{-1} \tilde{x}_2 \end{aligned} \quad (20)$$

Recalling that M and H_1 are positive definite matrices, F_g bounded and by choosing $h_{2i} > \zeta$, we obtain $\dot{V}_2 < 0$. Therefore, the surface $\tilde{x}_2 = 0$ is attractive, leading \hat{x}_2 to converge towards x_2 .

According to equations (5) and (11), after convergence of the position x_1 , we can estimate from the equation (3), the vertical displacements z_{ri} of the wheels and consequently, we deduce the vertical forces F_{ni} .

4 ESTIMATION RESULTS

In this section, we give some results in order to test and validate the robustness of our approach. Many tests are done with the PROSPER simulator. These simulation results are considered as references in our work and are used to compare with our estimation using sliding mode observers. The heavy vehicle inputs are represented by the road profile which is measured by the LPA instrument (see figure (3)). The figure (4) shows an example of the left and right road profile.

In the first two subplots on top of the figure (5), we represent respectively the measured and estimated suspension deflection and the roll angle and the respectively speeds. It is shown that these displacements are observed to be good and converge well towards the actual ones in finite time. In the bottom of this figure, the estimated and observed velocities are represented.

We notice that these unmeasured signals are well reconstructed with some errors concerning the roll rate.

The figure (6) shows that the observed wheel's vertical displacements are quite close to the true ones (estimated by the PROSPER simulator).

The convergence of the states is very fast and the estimation is high quality. The accurate reconstruction of these states allows estimation of the vertical forces. Then in figure (7), we present the behavior of this estimation.

We can observe that the estimated signals are accurate with respect to the true signals with errors near 0.



Figure 3: Longitudinal Profile Analyser (APL in French)

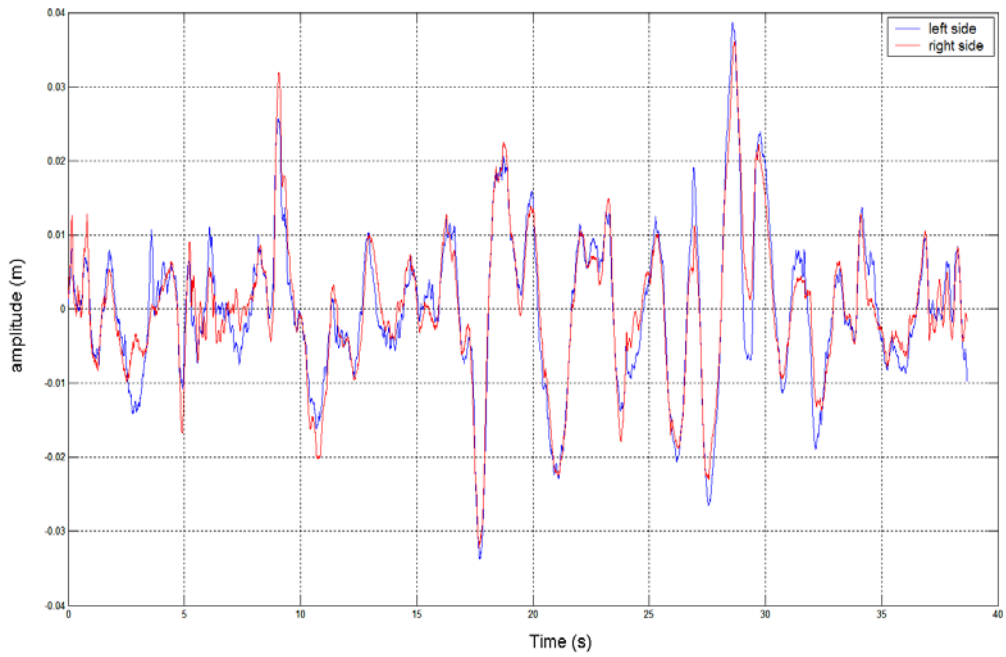


Figure 4: Road profile inputs

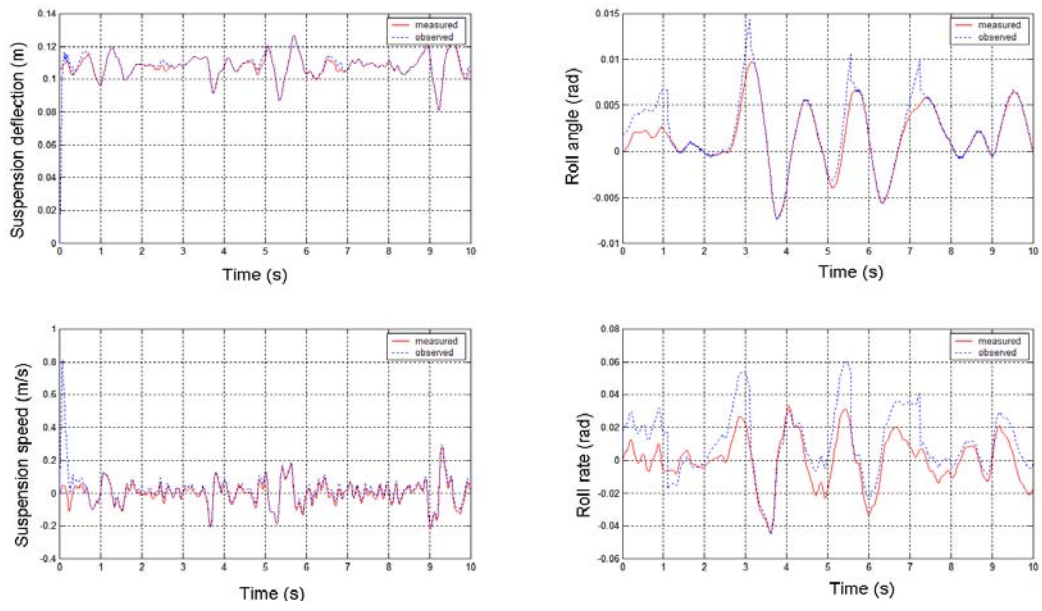


Figure 5: Estimated and measured suspension deflection

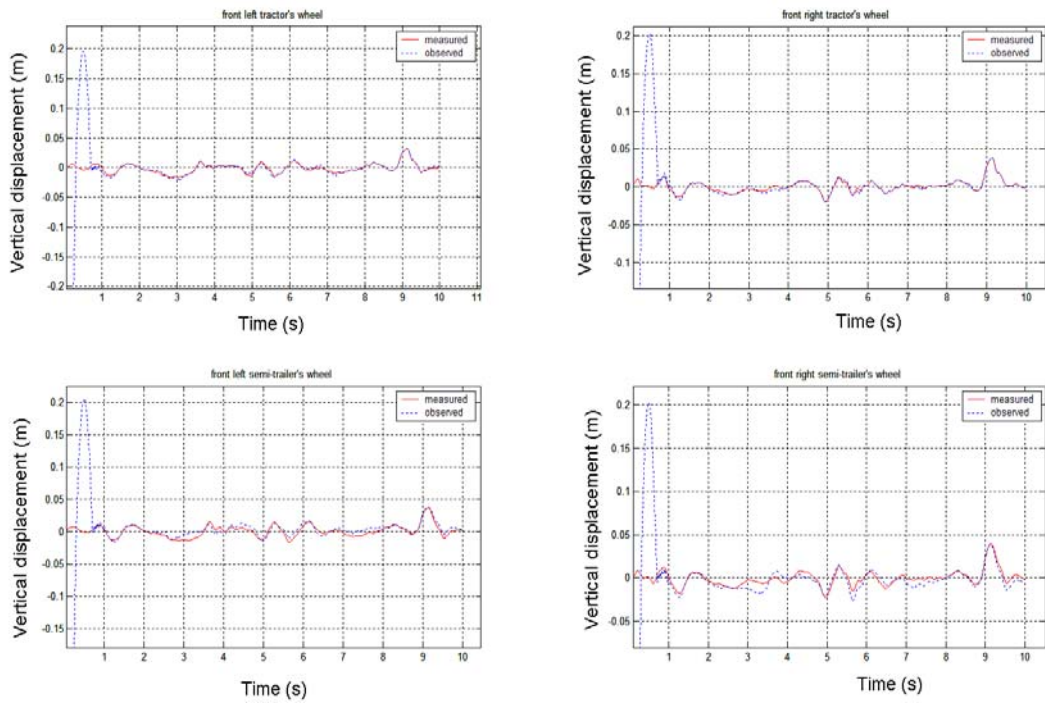


Figure 6: Estimated and measured wheel's vertical displacements

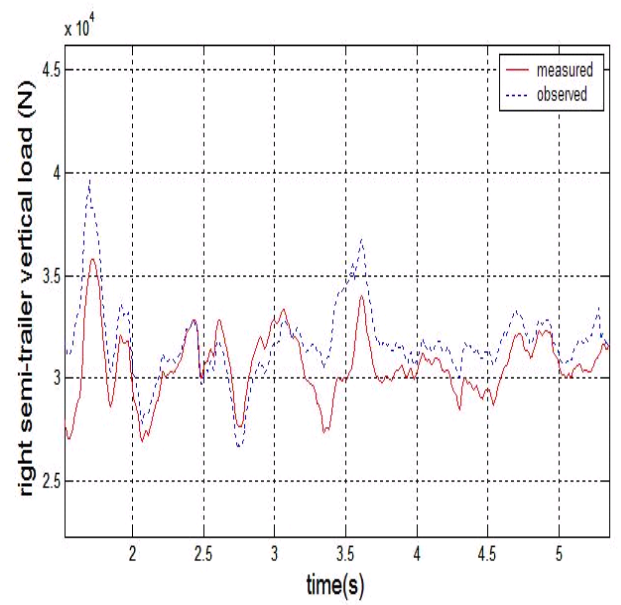
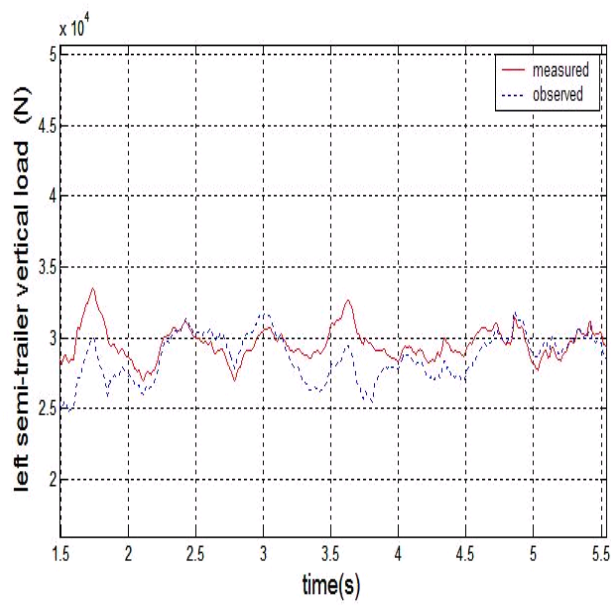
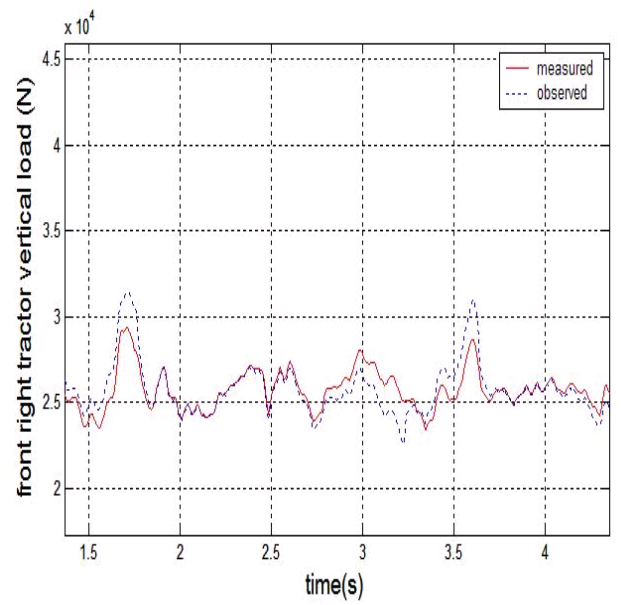
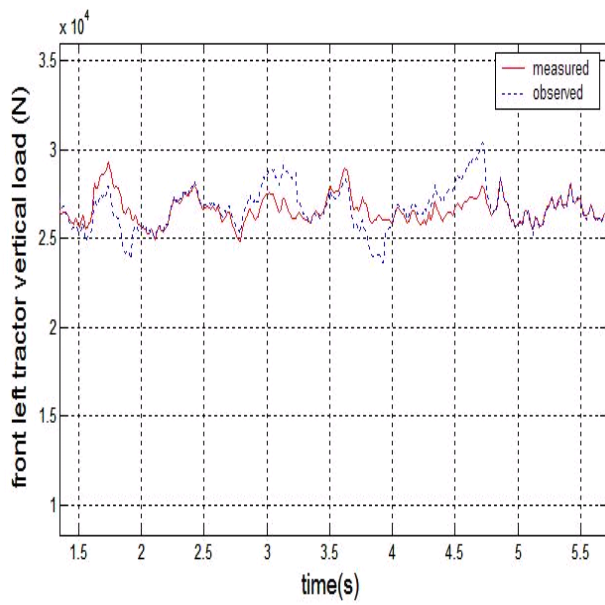


Figure 7: Estimated and measured vertical forces

5 CONCLUSION

In this paper, an original method is presented to estimate vertical forces acting on the wheels. A model of a heavy vehicle taking into account road profile input is developed. This model is validated using the PROSPER simulator. Then, in the second part, a sliding mode observer is built to estimate the heavy vehicle states and at the same time, the vertical loads. The states are well observed with errors near zero and the vertical forces are estimated and compared to true ones coming from the PROSPER simulator. Since the comparison with vertical forces by PROSPER is satisfying and the error is near zero, we can conclude that our method is interesting.

In future work, we will apply our method to actual heavy vehicles to estimate on board these vertical forces.

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References

- [1] M. Bouteldja, "Modélisation des Interactions Dynamiques Poids Lourd/Infrastructures Pour la Sécurité et les Alertes", PHD thesis, l'Université de Versailles Saint Quentin en Yvelines, 2005.
- [2] B. C. Chen and H. Peng, "A Real-time Rollover Threat Index for Sports Utility Vehicles", American Control Conference, San Diego, USA, pp. 1233-1237, June 1999.
- [3] S. V. Drakunov, "Sliding-Mode Observers Based on Equivalent Control Method", Proc. 31st IEEE Conf.
- [4] I.M. Ibrahim, "Design of reducing the generated dynamic tyre loads of the articulated tanker vehicles", Heavy Vehicle Systems, Int. J. of Vehicle Design, Vol. 11, No. 1, pp. 86-99, 2004
- [5] H. Imine, Y. Delanne and N. K. M'Sirdi, "Road Profiles Inputs to evaluate loads on the wheels", International Journal of Vehicle System Dynamics, Vol 43, pp. 359-369, Supplement, November 2005.
- [6] H. Imine, "Observation d'états d'un véhicule pour l'estimation du profil dans les traces de roulement". PHD thesis, l'Université de Versailles Saint Quentin en Yvelines, 2003.
- [7] V. Legeay and P. Daburon and C. Gourraud, Laboratoire Central des Ponts et Chaussées, DGER/IRVAR, "Comparaison de mesures de l'uni par L'Analyseur de Profil en Long (APL)"

- [8] Misun, Vehicle System Dynamics, pp. 237-253, Simulation of the interaction between vehicle wheel and the unevenness of the road surface, Vol. 19, 1990.
- [9] H.B. Pacejka. "The tire as a vehicle component", Proceeding of the 26th FISITA, Congress'96 , 1996.
- [10] PROSPER, Sera-Cd. Technical report, www.sera-cd.com.
Decision and Control, Tucson, Arizona, pp. 2368-2369, 1992.
- [11] J. J. E. Slotine, J. K. Hedrick & E. A. Misawa. "On Sliding Observer of Nonlinear Systems".
Journal of Mathematical System, Estimation and Control, No. 109, pp 245-259, 1987.
- [12] 13. V. I. Utkin and S. Drakunov, "Sliding Mode Observer", IEEE conference on Decision and Control, pp. 3376-3378, Orlando, Florida USA, 1995.