

ROLLOVER RISK PREDICTION OF HEAVY VEHICLES USING SLIDING MODE OBSERVER

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ABSTRACT

This paper presents a rollover risk prediction system of heavy vehicles using sliding mode observers. This system comprises several sensors which measure the dynamics of the vehicle. Initially, the vehicle states are predicted and estimated using observers, and then, knowing infrastructure characteristics, the rollover's risk is assessed using the lateral acceleration of the vehicle, the center of gravity height, the LTR (Load Transfer Ratio) and the load on each wheel.

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1 INTRODUCTION

In recent years, many authors have become interested in the problem of the rollover of heavy vehicles ([4], [8]). However the interaction with the infrastructure is not taken into account. The originality of our method developed in this paper is to consider the infrastructure data such as road profile, radius of curvature, lateral inclination of the road, etc., to show their influence on the proposed warning system. In this study, we present an observer to estimate the dynamics of heavy vehicle in interaction

with the infrastructure and an estimator for vertical loads. The designed observer is based on a sliding mode approach. If a high risk is detected, a warning is given to the driver to prevent the rollover.

Several simulations with various scenarios are presented, done with the PROSPER simulator ([7]), to assess the accuracy of our method and then the efficiency and reliability of the whole system. Simulations are presented on the estimation of vertical forces, and discussed to evaluate the robustness of our approach and to assess the impact of the infrastructure.

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

2 MODEL DESCRIPTION

Various studies have dealt with the heavy vehicle modelling without taking into account the infrastructure data base ([3], [5], [1]). However, these data are very important to study the different risks such as rollover or jackknifing. In this work we develop a heavy vehicle model using the infrastructure data (road profile, radius of curvature, lateral road inclination) as inputs (see figure (1)). These data are coming from measures done at Lyon (in France) within the framework of an European project called SAFEMAP. In this paper, we consider the tractor-semitrailer model (with 2 axles for the trailer and 1 axle for the semitrailer) with 12 degrees of freedom. This model is derived using Lagrangian's equations. In this paper, we neglect the relative yaw angle and the longitudinal movement of the heavy vehicle.

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 $M \in \mathfrak{R}^{11 \times 11}$ is the inertia matrix (Mass matrix), $C \in \mathfrak{R}^{11 \times 11}$ is related to the damping effects, $K \in \mathfrak{R}^{11}$ is the springs stiffness vector and $F_g \in \mathfrak{R}^{11}$ is a vector of generalized forces.

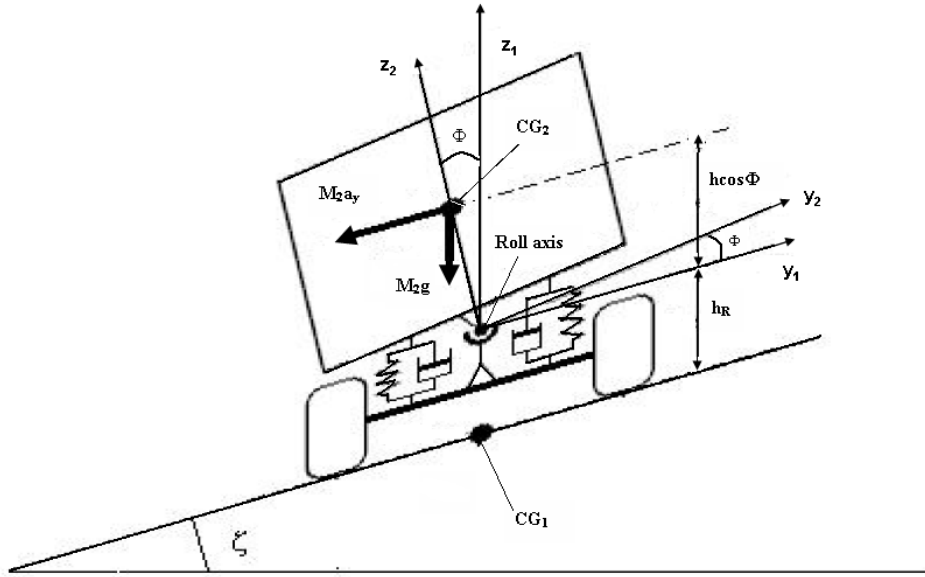


Figure 1: Heavy vehicle model on the road with the lateral inclination

$q \in \mathfrak{R}^{11}$ is the coordinates vector defined by:

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

\dot{q} and \ddot{q} represent respectively the speed and acceleration vector, ϕ is the tractor roll angle, ψ is the yaw angle of the tractor, 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 (height of the gravity center), y is the lateral displacement of the tractor.

The suspension is modeled as the combination of a nonlinear spring and damper elements. The tractor chassis (with the mass M) is suspended on its axles through two suspension systems. We can also shown that the tyre is modeled by spring and damper elements. The road profile inputs are considered as the heavy vehicle inputs. z_{ri} $i = 1..6$ represent the vertical displacements of the wheels with respect to coordinate system fixed to the ground (road). These can be calculated using the following equations:

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

where T_w is the tractor track width and r is the wheel's radius. The lateral road inclination is represented by ς .

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

On the other hand, the dynamic rolling of the vehicle is described using the following differential equations.

$$I_{xx}\ddot{\phi} = ma_y h + mg \sin(\phi + \varsigma) - C_R \dot{\phi} - K_R \phi \quad (5)$$

where I_{xx} is the inertia moment in the roll axis, C_R represents a damping coefficient of the roll motion, K_R is spring coefficient of the roll motion, $\dot{\phi}$ is the roll rate and $\ddot{\phi}$ is the roll acceleration according to the road. h is the center height of gravity in relation to the roll axis (see figure (??)). g represents the gravity acceleration and a_y is the lateral acceleration of the center of gravity height. This can be calculated by the following equation:

$$a_y = v\dot{\psi} + g \sin(\phi + \varsigma) \quad (6)$$

where v is the vehicle speed and $\dot{\psi}$ is the yaw rate.

3 ROLLOVER RISK ESTIMATION

In this section, we develop the triangular sliding mode observers to evaluate the heavy vehicle states and the vertical loads. This observer allows in finite time for the convergence of the positions and speeds (see more details in [2]).

The LTR (Load Transfer Ratio) is dependent on the load on each wheel and calculated as follows:

$$LTR = R = \frac{F_{nr} - F_{nl}}{F_{nr} + F_{nl}} \quad (7)$$

Many studies ([3]) have shown that this coefficient can be approximated (we assume that the roll angle ϕ is small) by the following expression:

$$R \simeq \frac{2(h + h_R)}{T} \frac{a_y}{g} + 2\frac{h}{T}\phi \quad (8)$$

where h_R is the roll axis height in relation to the ground (see figure (??)).

When $F_{nr} = 0$ ($F_{nl} = 0$) the right (left) wheels lift off the road and the rollover coefficient takes on the value $R = -1$ ($R = 1$). For straight driving on a horizontal road for the tire vertical loads, it holds that $F_{nr} = F_{nl}$ which means that $R = 0$.

The rollover occurs when the moment created by the lateral acceleration acting on the vehicle exceeds a threshold value a_{yc} . This value depends on the vehicle parameters, center of gravity height and infrastructure data base.

In this paper, the critical LTR value is about 0.9.

Then a_{yc} is calculated from the equation (8) as follows:

$$a_{yc} = \frac{(0.9Tg - 2gh\phi)}{2(h + h_R)} \quad (9)$$

To evaluate the *LTR*, we must first estimate the heavy vehicle states and the vertical loads. Then we write the dynamic model (1) in the state form as follows:

$$\begin{cases} \dot{x} = f(x) + F_g \\ y = h(x) \end{cases} \quad (10)$$

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:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \ddot{q} = M^{-1}(F_g - C(x_1, x_2)x_2 - K(x_1)) \\ y = x_1 \end{cases} \quad (11)$$

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: $u_i < \mu$).
3. The generalized forces F_g are bounded (\exists a constant $\eta \in \mathfrak{R}$ such as: $F_{gi} < \eta$).
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:

$$\|C(x_1, x_2)x_2\| \leq c_1 \|x_2\|^2 \quad (12)$$

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

3.1 TRIANGULAR OBSERVER DESIGN

In order to estimate the state vector x and to deduce the vertical loads vector F_{ni} , we propose the following triangular sliding mode observer [6]:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + H_1 \text{sign}(\tilde{x}_1) \\ \dot{\hat{x}}_2 = M^{-1}(F_g - C(\hat{x}_1, \hat{x}_2)\hat{x}_2 - K(\hat{x}_1)) \\ \quad + H_2 \text{sign}(\bar{x}_2 - \hat{x}_2) \end{cases} \quad (13)$$

where \hat{x}_i represents the observed state vector and:

$$\bar{x}_2 = \hat{x}_2 + H_1 \text{sign}(\tilde{x}_1) \quad (14)$$

$H_1 \in \mathfrak{R}^{11 \times 11}$ and $H_2 \in \mathfrak{R}^{11 \times 11}$ represent positive diagonal gain matrices.

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

$$\hat{F}_{ni} = F_{ci} + f(\hat{x}_1) \quad (15)$$

3.2 CONVERGENCE ANALYSIS

We can write the dynamics estimation errors as follows:

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

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 $V_1 = \frac{1}{2}\tilde{x}_1^T\tilde{x}_1$. The time derivative of this function is given by:

$$\dot{V}_1 = \tilde{x}_1^T(\tilde{x}_2 - H_1 \text{sign}(\tilde{x}_1)) \quad (17)$$

By considering gains matrix $H_1 = \text{diag}(h_{i1})$ with $h_{i1} > |\tilde{x}_{i2}|$, $i = 1...8$, then $\dot{V}_1 < 0$. Therefore, from sliding mode theory [9], 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 and according to equation (14), we have the convergence of \tilde{x}_2 towards x_2 .

Then, equation system (16) can be written as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - H_1 \text{sign}_{eq}(\tilde{x}_1) \rightarrow 0 \\ \dot{\tilde{x}}_2 = M^{-1}\tilde{F}_g - M^{-1}(C(x_1, x_2)x_2 \\ - C(x_1, \hat{x}_2)\hat{x}_2) - H_2 \text{sign}(\tilde{x}_2) \end{cases} \quad (18)$$

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

$$\dot{V}_2 = \tilde{x}_2^T M \dot{\tilde{x}}_2 \quad (19)$$

Then, from equation (18), \dot{V}_2 becomes:

$$\dot{V}_2 = \tilde{x}_2^T \tilde{F}_g - \tilde{x}_2^T (C(x_1, x_2)x_2 - C(x_1, \hat{x}_2)\hat{x}_2) \quad (20)$$

$$- \tilde{x}_2^T M H_2 \text{sign}(\tilde{x}_2) \quad (21)$$

Using the assumption 4, we have:

$$C(x_1, x_2)x_2 - C(x_1, \hat{x}_2)\hat{x}_2 = -C(x_1, x_2)\tilde{x}_2 - C(x_1, \hat{x}_2)\tilde{x}_2 \quad (22)$$

Finally the equations (20), becomes:

$$\begin{aligned} \dot{V}_2 = & \tilde{x}_2^T \tilde{F}_g - \tilde{x}_2^T (-C(x_1, x_2) + C(x_1, \hat{x}_2)) \\ & + M H_2 \text{sign}(\tilde{x}_2) \end{aligned} \quad (23)$$

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 in finite time t_1 .

According to equations (4) and (15), 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} .

Finally, we obtain the convergence of all system states (positions, center of gravity height, speeds and accelerations) and we can calculate the LTR coefficient using the equations (7) or (8).

4 ROLLOVER RISK PREDICTION

One of the most important and difficult parameters is the detection of the risk in order to prevent rollover of heavy vehicles. Then the knowledge of the time to rollover (TTR) is very interesting. In this paper, we choose this time as 3 seconds.

As in the previous section, to calculate the LTR , we use the values of the infrastructure data base at the time $(t + 3)$ second. From equation (9), and if the risk is detected ($LTR \geq 0.9$), we calculate the critical lateral acceleration of the heavy vehicle and we can deduce the maximal speed of the vehicle. This information can be sent to the driver to prevent the rollover.

5 MAIN RESULTS

In this section, we give some results in order to test our approach. These simulation results are considered as reference in our work to compare with estimation using mode observers. Some infrastructure data bases are used in simulation.

We begin with a heavy vehicle rolling on a straight line. In the figure (2), we notice that the center of gravity height is observed to be acceptable.

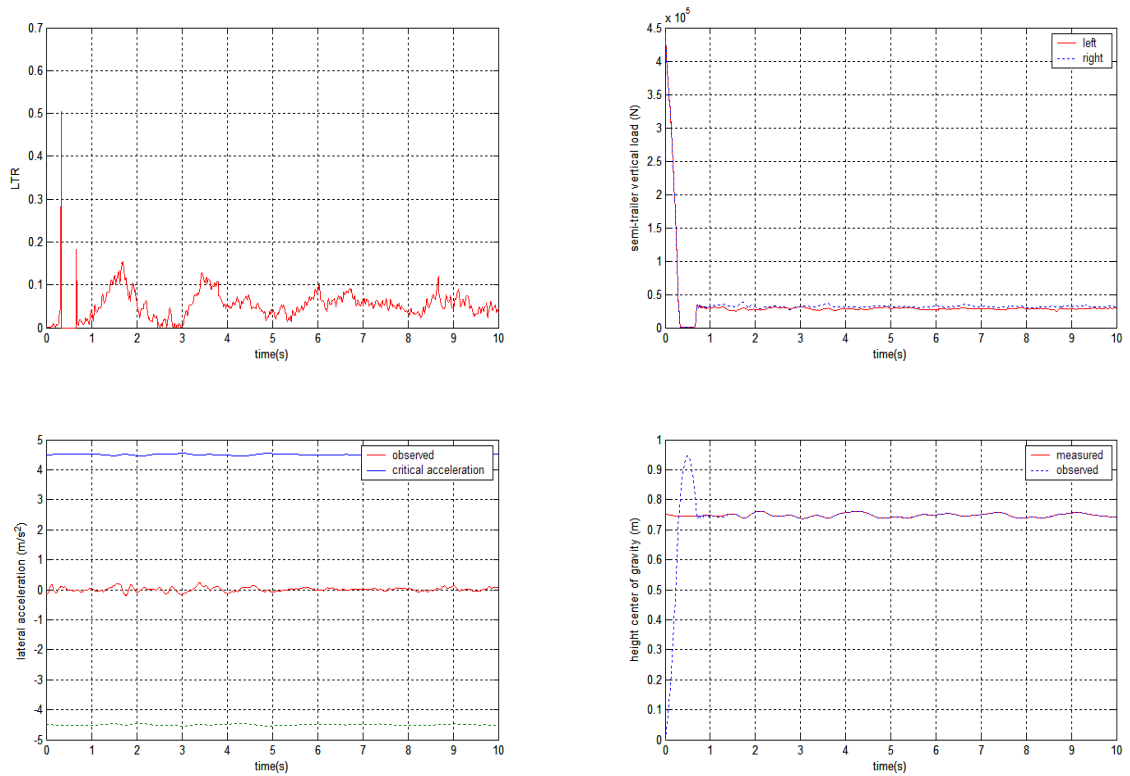


Figure 2: LTR comparing to the lateral acceleration and vertical load

We also show that the vertical load of the semi-trailer is quite close to the true one estimated by the PROSPER simulator. This allow us to determine the LTR. Since this coefficient does not reach the value 0.9, we can conclude that no risk is detected. This result is confirmed by the estimated lateral acceleration which stays very far from the critical lateral acceleration.

The figure (3) shows the calculated LTR , using the equation 7:

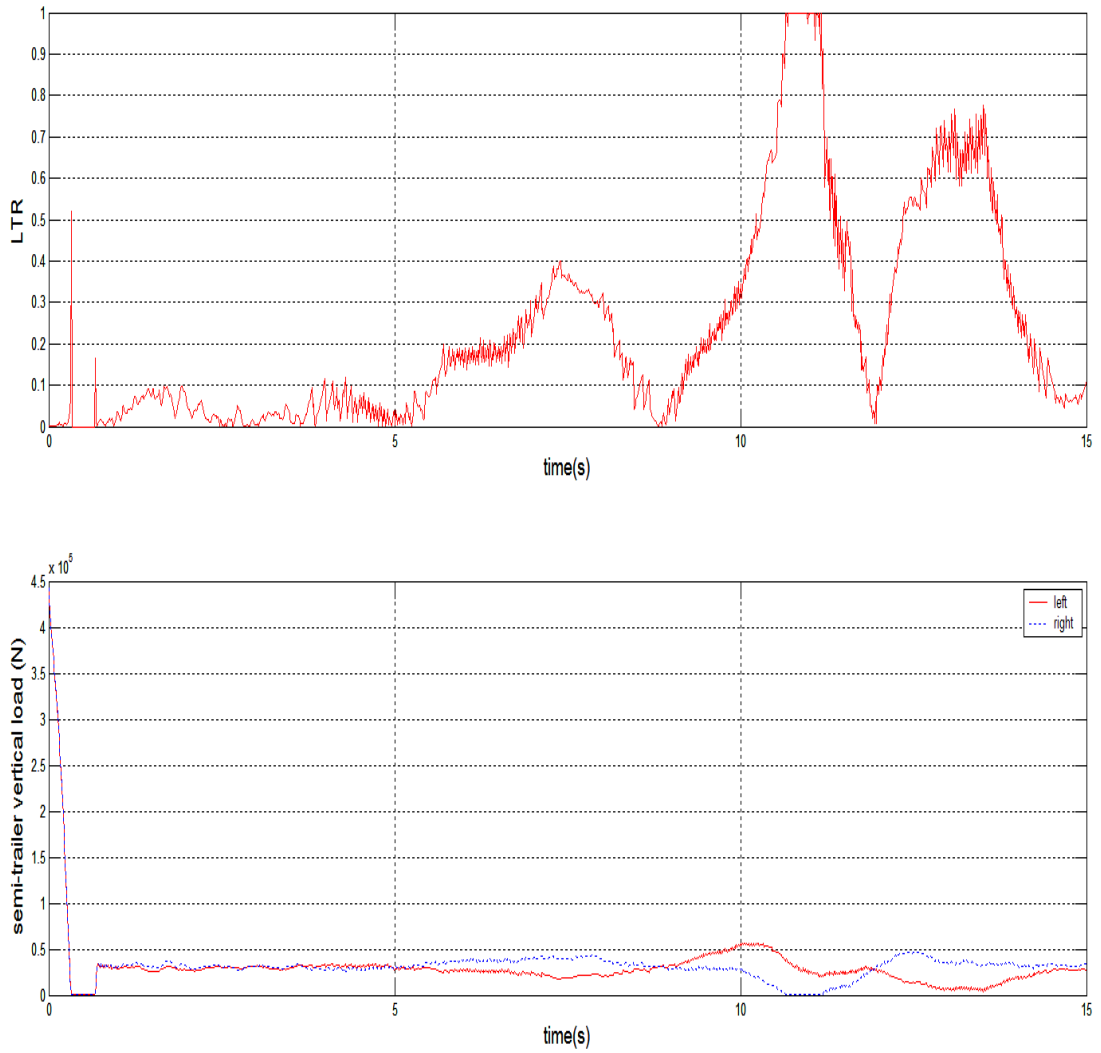


Figure 3: LTR comparing to the vertical load

This figure shows that the left vertical load of the semi-trailer is near to 0 at time 11s. We notice that at this same time, the LTR is near to 0.9.

In the figure (4), we show the good estimation of the center of gravity height.

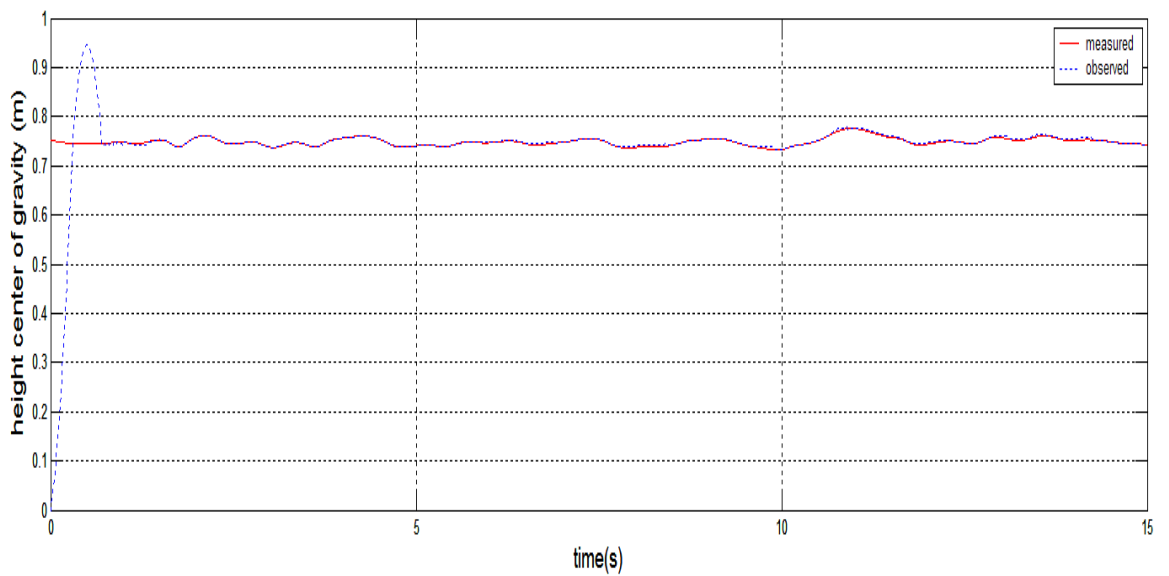
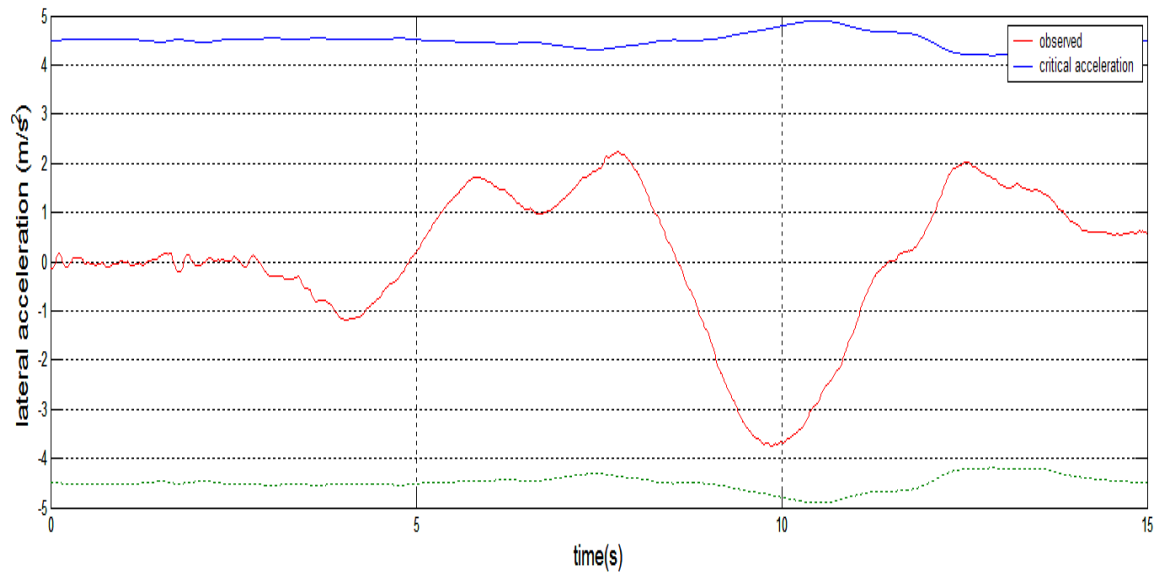


Figure 4: Critical lateral acceleration and estimated height center gravity

In the top, the lateral accelerations are compared. We notice that the estimated lateral acceleration converges toward the limited one (critical one) after only 10 seconds.

This means that, using the acceleration, the risk is detected before the *LTR* method (11s).

6 CONCLUSION

In this paper, we present an original method about rollover risk detection and estimation. A model of a heavy vehicle taking into account the infrastructure data base (road profile input, curvature radius, road lateral inclination) is developed and tested using the PROSPER simulator. The sliding mode observers able to estimate heavy vehicle dynamics and the vertical loads are developed and the convergence is studied. We show through different results that the states are observed to be acceptable and the center of gravity height is estimated. Then we estimate the vertical loads in order to calculate *LTR* coefficient. In the first example, when a heavy vehicle is rolling on a straight line, we have seen that no risk is detected and the *LTR* stays lower than 0.9. In the second example, the vehicle is rolling on the curvature. The results show that a risk is detected and the left vertical load of the semi-trailer is near to 0 while the vertical load of the tractor stays different to 0. We can then conclude that the semi-trailer rolls over first, before the tractor.

An other important result is the risk detection using the lateral acceleration. We have seen that the estimated lateral acceleration reaches its limit in 10 seconds that is before the *LTR* reaches 0.9. In this paper, we choose arbitrarily the time to rollover (*TTR*). In the next works, we will test our approach using sliding mode observers to estimate this *TTR*.

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