DRIVING SITUATION ANLYSIS BASED ON MULTI-MODEL OBSERVER: APPLICATION OF RISK ACCIDENT OF HEAVY VEHICLES

Graduated of the Blida University of Algeria, speciality in Mechanical Engineering, 2000. Completed a PhD in 2005 on Heavy Vehicle Dynamics and Road Interaction in the Université de Versailles St Quentins en Yvelines in France. Currently researcher at team n°12 about "Skid Resistance and Heavy Vehicle Safety" at the Laboratoire Régional des Ponts et Chaussées in Lyon since 2008.

Graduated of the National School of Public Works of Lyon, speciality in Civil Engineering, 2000. Completed a PhD in 2005 on materials and construction in the Doctoral School Mechanic, Energetic, Civil Engineering and Acoustic in Lyon. Currently responsible from the research team n°12 about "skid resistance and safety" at the Laboratoire régional des Ponts et Chaussées in Lyon since 2008.

M. BOUTELDJA **V. CEREZO**

Laboratoire Régional des Ponts et Chaussées de Lyon
25, Avenue François Mitterrand
69500 Bron-France

Abstract:
The vehicles stability analysis is often based on mechanical models, which can be very complex. Indeed, the complete modeling of heavy vehicle is very difficult to design and to use because of uncertainties on the model parameters. The multi-model approach (Sequencing of Situations of Driving) can be an alternative to reduce the complexity of the vehicle models and at the same time to focus on the request dynamics. In this paper, a heavy vehicle stability analysis on a real route (ramp) for which we defined a whole driving situations is proposed. For each situation, one associates a model representative of the requested dynamics. Thereafter, the unknown dynamic state reconstructed at every moment by using the multi-observer, based on the multi-model and the driving situation. The observation approach used for the reconstruction of the vehicle dynamics is based on the sliding mode technique. The computed information leads to detect the accident risky situation. Simulation of observation and rollover detection results are presented to show the effectiveness of the proposed method.

Key words: Rollover, Heavy Vehicle, Modeling, Observer, Stability Analysis and Detection.

1. Introduction

Accidents involving heavy vehicles have serious consequences for road users, and incidents induce major congestions or damage to the environment or the infrastructure at disproportionate economic costs. Thus, it is essential to study the origin of these accidents and to prevent them. Several driving situations can directly or indirectly induce the accidents, such as an excessive speed when entering a curve, a severe lane change, an obstacle avoidance manoeuvres, when the height gravity center (CoG) is too high or when there is external disturbance impact like side wind on the truck (Winkler, 1999), (Dhalberg, 1999).

However, more and more studies and new active safety systems are developed and installed on vehicle in order to real-time monitoring and controlling the dynamic stability (EBS, ABS, ESP). So, these systems are generally designed to help avoid accidents or at least reducing their severity (Chen, 2001). Moreover, these systems need to access, in real time, to information concerning the vehicle dynamics interacting with its environment. In some cases, the real-time acquisition of this information by sensors embedded or installed in vehicles is very difficult. Thus, the use of an overall model of vehicle for the reconstruction of these information also creates some problems of handiness and management of many parameters. The multi-model approach (depending on driving situations) can be an alternative to reduce the complexity of the vehicle models and at the same time focusing on the request dynamics.

In this paper, we propose an analysis of a route of heavy vehicle for which we defined relevant situations (sequencing of driving situations). The vehicle during a phase of control meets several control situations (straight line, turn, braking and acceleration). For each situation, a model representative of requested dynamics is associated. The movement of vehicle can be divided into 6 traditional situations (speed constant, deceleration and acceleration on straight line or turn). Three models are presented: a pure longitudinal model which translates in this case longitudinal dynamics by taking account of the slope of the road and the aerodynamic loads; a pure lateral model which translates the lateral dynamics of the vehicle in a turn at constant speed; a coupled model which represents deceleration or acceleration in a turn with weak lateral requests (Gillespie, 1992). The route is seen like a succession of simple unit trajectories. Their association makes it possible to rebuild a real route.

Thereafter, to rebuild the dynamic state at any moments, an observer multi-model is elaborated in adequacy with the driving situation. This approach is based on the technique of the observers to sliding modes (Bouteldja, 2005). This technique allows determining the non-measurable variables of state of the vehicle as well as the unknown parameters using the available measurements (Utkin, 1995). We thus call upon non-linear observers for each driving situation (model). The structure of obtained total observation are named “multi-model observers”, thus makes up of observers particular to each situation of driving. The change from one observer to another is ensured by an automat (switch).

Finally, an approach detecting risky situations is proposed. The method includes an algorithm for risk evaluation, using stability criteria. These criteria are based on relevant information on the dynamics of the vehicle (state variables estimated by observer) and its environment. The risky situations considered and analysed in this work are rollover of heavy vehicle.

In this study, some simulation results are presented to show the effectiveness of the proposed approach. Simulations are carried out using the Matlab/Simulink software. As it's not possible to have experimental measurements, the simulation data and the simulation results are compared with results provided by a heavy vehicle dynamic simulator (PROSPER).

2. Stability and Driving Situations Analysis

2.1 Stability Analysis

The heavy vehicle accidents related to driving conditions and occur when the vehicle is beyond the stability limits (Bouteldja, 2003). The return to stability is a complicated task for the driver then the accident is inevitable for an inexperienced driver or devoid of intelligent system. In particular, the rollover statistics of heavy vehicle, shown that the majority of these accidents type crossing a have been produced during driving situations like braking manoeuvres, lane change driving, round about or vehicle skidding (Dahalberg 1999). The main influence comes from the vehicle velocity, the road pavement, and the friction changes.

In this section, a stability analysis of heavy vehicle due to parameters variation mentioned above is presented. To simplify this analysis, the simulations were performed on a simple model, where the heavy vehicle is presented by one axle.

Variation of Road Curve

The steering angle is a parameter strongly related to the mobilization of the lateral forces. Figure 1 shows that the stability strongly decreases with speed when the steering angle varied. Thus, more and driving speed, steering angle decreases. Thus, more and driving speed, steering maneuvers are limited to maintain stability of the vehicle (no sudden change).

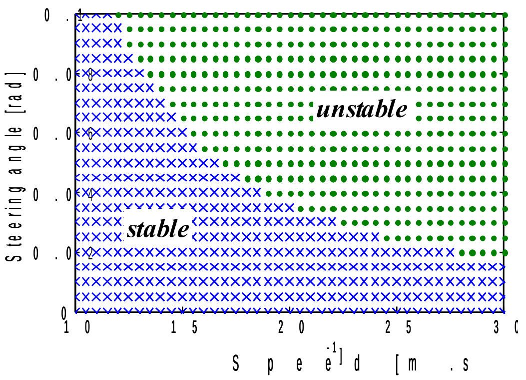

Figure 1 Stability steering/speed plane

Variation of Skid resistance/Friction

Adhesion was one of the most important parameters in the rollover accident. Figure 2 represents the stable and unstable domains. When adhesion decreases, the heavy vehicle is stable. The rollover main characteristic (the strong mobilization of the pneumatic contact) is represented: when adhesion is lower than 0.4, which corresponds to a wet roadway, the vehicle does not turn over. However, other problems appear like trajectory loss while slipping.

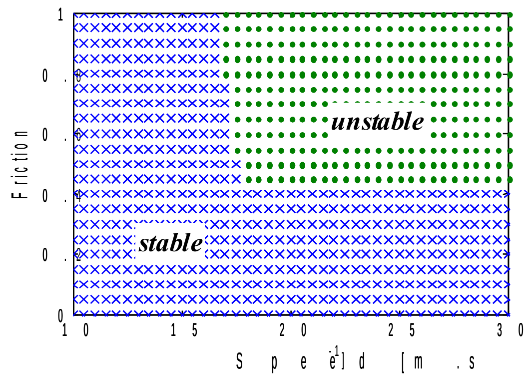

Figure 2 Stability adhesion/speed plane

Variation of Mass and Height of the Gravity Center

When the sprung mass increases at a constant height, the efforts that suspensions must generate are more important. For this purpose, the suspensions deflection should be high, involving an increase in roll angle, and therefore instability of heavy vehicle. However the effect of the suspended mass variation on the overall stability of the heavy vehicle is low, as shown in figure 3. In this configuration, stop the arrival of a suspension is neither model nor simulated (Bouteldja, 2004).

Figure 3 shows the stability domain of heavy vehicle compared of the height of gravity center (CoG) and longitudinal speed variations. We note that an increase of the height of CoG strongly decreases the maximum speed at which a heavy vehicle can approach to a turn. Therefore this height has a great influence in particular for the low height of CoG between 0.5 *m* and 1*m*, critical acceleration passes from 6.3 $m.s^{-2}$ to 4.3 $m.s^{-2}$, against 3.4 $m.s^{-2}$ for 1.5 *m*.

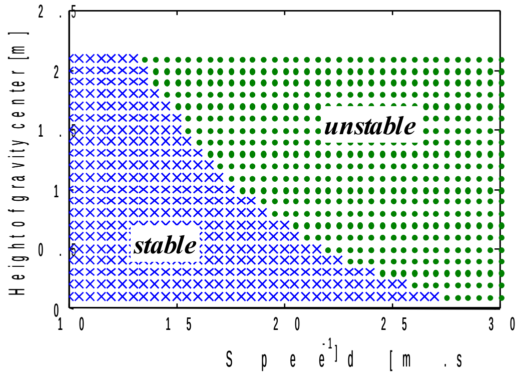

Figure 3 Stability height of gravity center/speed plane

Other Factors Influencing Roll Stability
We studied the combined variations of several suspension parameters (stiffness and damper viscosity) on the critical lateral acceleration. We note that, the increase of these parameters greatly reduces the heavy vehicle roll angle. Figure 4 shows the increased instability risk with the increase of these parameters. However, increasing (or reducing) the suspensions stiffness or viscosity is not a solution because the vehicle becomes uncomfortable and controllability problems could arise.

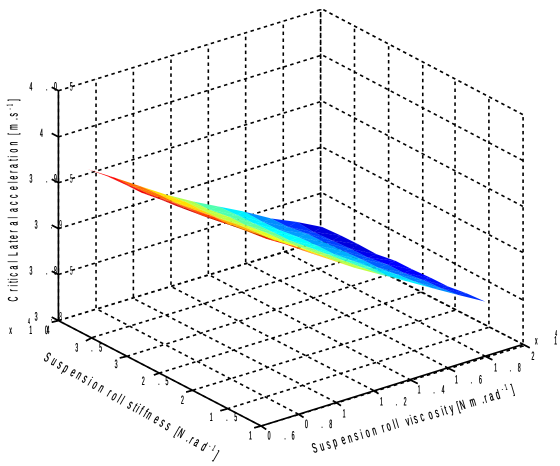

Figure 4 Stability CoG/speed plane

Finally, torsional compliance of the vehicle frames also reduces the rollover threshold. For example, a flexible trailer frame rolls to a greater angle under the influence of lateral acceleration, thus increasing the magnitude of the destabilising lateral displacement moment. Furthermore torsional compliance of the tractor frame reduces the ability of the tractor steer axle to provide a stabilising moment which resist to the roll motion of the payload.

Torsional compliance of the vehicle couplings reduces roll stability in a similar way. Note that the roll stiffness of a conventional fifth wheel coupling decreases with articulation angle, and the roll moment that can be transmitted through a fifth wheel coupling saturates at an included angle of around 2° (Bouteldja, 2004).

2.2 Driving Situation Analysis

Criterion of trajectory
In this section the vehicle motion on the route is decomposed into 6 simple situations (table 1). The route is seen like a succession of simple unit trajectories. Their associations make it possible to rebuild a real route. The transition from one situation to another requires the

definition of criteria allowing identifying each situation. These driving situations are directly drawn from the vehicle motion analysis interacting with its environment.

Table 1- Description of the driving situation

Trajectory	Mode		
Straight line	Constant speed	Braking	Acceleration
Turn	Constant speed	Braking	Acceleration

The criteria of change of driving situation are defined by the longitudinal and lateral accelerations; $\dot{v}_x$ and $\dot{v}_y$. For example, lateral acceleration is not null in the turn, whereas it is null of the straight line. Then, when the longitudinal speed is constant, the longitudinal acceleration is null. Once the transitions from passage between the various established driving situations, we define a management structure using an automat (table 2).

Table 2 Management structure (switch)

States	State N°1	State N°2	State N°3	State N°4
State N°1		$\dot{v}_x \neq 0, \dot{v}_y = 0$	$\dot{v}_x \neq 0, \dot{v}_y \neq 0$	$\dot{v}_x = 0, \dot{v}_y \neq 0$
State N°2	$\dot{v}_x = 0, \dot{v}_y = 0$		$\dot{v}_x \neq 0, \dot{v}_y \neq 0$	-
State N°3	$\dot{v}_x = 0, \dot{v}_y = 0$	$\dot{v}_x \neq 0, \dot{v}_y \neq 0$		$\dot{v}_x = 0, \dot{v}_y \neq 0$
State N°4	$\dot{v}_x = 0, \dot{v}_y = 0$	-	$\dot{v}_x \neq 0, \dot{v}_y \neq 0$	

Scenario

To associate a model to every driving situation and develop observers, a scenario representing the driving situations is proposed. Within this framework, a cutting in 5 driving situations is realised to describe the trajectory of this scenario (figure 5).

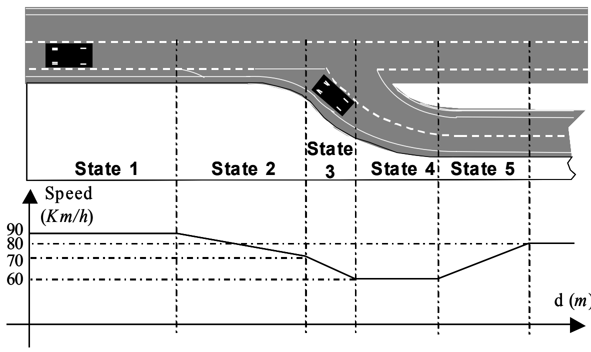

Figure 5 Scenario of driving situations

First situation, the heavy vehicle is launched in straight line with a constant speed of 90 km/h, then it decelerates in the approach a exit (braking in straight line) during a distance approximately 80 m to start the turn with a speed of 70 km/h very continuously to slow down in a moderate way. Braking continues during in curve (braking in curve) until the speed of 60 km/h, which it preserves during the major part of the turn out (constant speed in curve). Thereafter an acceleration phase starts at the end of the curve (acceleration in curve) and continues to accelerate in straight line until the speed of 80 km / h, and then continues its trajectory has the same speed (constant speed).

3. Driving Situation Observer

In this section, we seek to reconstruct the non-measurable dynamics variables of the vehicle and to estimate the unknown parameters using available measurements directly in the vehicle. For this, an observation method based on sliding mode approach is applied

(Filippov, 1988), (Utkin, 1995). This method is known to be robust versus parametric uncertainties, modeling errors and disturbances. We associate to each driving situations presented previously an observer. These observers are developed from driving situation models. The operation of all the models and observers are ensured by an automat (cf. Table 2). The total structure observation is named "observers multi-model". These allow evaluating rollover risk of heavy vehicle in each driving situation.

3.1 Pure Longitudinal Dynamics Observer

Dynamic model

Fig. 6 illustrates a simple model of the vehicle, which will be used for the considerations in this driving situation. This model described the longitudinal dynamics of vehicle in a straight line, where m is vehicle mass, F_{xf} et F_{xr} are respectively the longitudinal forces at the rear and front wheel. F_d is the resistance force of wind. γ is the road slope. The vehicle displacement is described by the following equation:

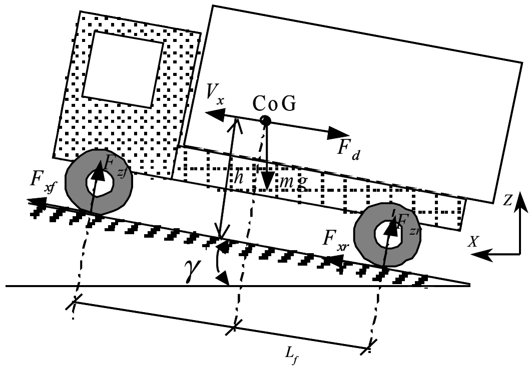

Figure 6 - Vehicle longitudinal model

$$
\begin{aligned}
&m\dot{v}_x = F_x - F_d - mg\sin(\gamma) \\
&F_x = F_{xf} + F_{xr} \ and \ F_d = \frac{1}{2}C_d\rho A v_x^2 sign(v_x)
\end{aligned}
\quad (1)
$$

where A is the vehicle frontal section, ρ is the air density, and C_d is the aerodynamic coefficient. L_f and L_r are the distances from the front and rear wheels to the gravity center.

The wheel dynamics is taken into account by the following equations:

$$
\begin{aligned}
&I_w\dot{w}_f = -rF_{xf} + T_f sign(w_f) \\
&I_w\dot{w}_r = -rF_{xr} + T_r sign(w_r)
\end{aligned}
\quad (2)
$$

where r is the wheel radius, I_w is the inertia moment of the wheel, T_f and T_r are the couples of traction/braking applied respectively on the wheel front and rear. ω_f and ω_r represent the angular velocity of wheel front and rear.

Observer design

In this part, a sliding mode observer is developed which is used to estimate the dynamics states of heavy vehicle (Utkin, 1995), (Bouteldja, 2005). From the model (Eq.1 and Eq.2), the observation structure is proposed by:

$$
\begin{cases}
\dot{\hat{v}}_x = \frac{1}{2m}\left[C_d\rho A\hat{v}_x^2 sign(\tilde{v}_x) + 2\Gamma_x sign(\tilde{v}_x)\right] \\
I_w\dot{\hat{w}}_f = \frac{T_f}{I} + \Gamma_w sign(\tilde{w}_f),\ I_w\dot{\hat{w}}_r = \frac{T_r}{I} + \Gamma_w sign(\tilde{w}_r)
\end{cases}
\quad (3)
$$

where $\tilde{v}_x = v_x - \hat{v}_x$, $\tilde{w}_f = w_f - \hat{w}_f$ and $\tilde{w}_r = w_r - \hat{w}_r$ are the observation errors and Γ_x and Γ_w are the observer gains. For convergence proof and more details, the reader can see (Utkin, 1995), and the observer gains must satisfy the following inequalities

$$\Gamma_x > \left|\frac{F_{x\_max}}{m} + g\right|,\ \Gamma_w = \left|\frac{r}{I_w} F_{x\_max}\right|$$

3.2 Pure Lateral Dynamics Observer

Dynamic model

In this configuration, the heavy vehicle is represented by non-linear bicycle model, which is obtained by approximating the front and rear pairs of wheels as single wheels. Assuming that the forward speed is constant. In order to integrate roll motion in this model, it will now be extended by a suspended mass attached to the chassis through the suspension. The gravity center is at a height h above the rolling axis. ϕ is the roll angle of the suspended mass. The motion equations of model illustrate in figure 7 are given by:

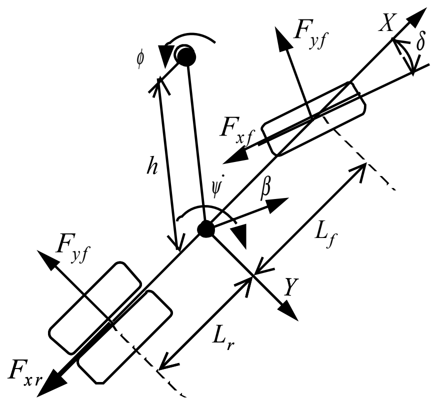

Figure 7 - Vehicle lateral model

$$\begin{aligned}
& m(\dot{v}_y + v_x\dot{\psi}) = F_{yf} + F_{yr} + mg\sin(\varphi) \\
& I_{zz}\ddot{\psi} - I_{xz}\ddot{\phi} = L_f F_{yf} - L_r F_{yr} \\
& I_{xx}\ddot{\phi} + k_\phi \phi + c_\phi \dot{\phi} - I_{xz}\ddot{\psi} = mh(\dot{v}_y + v_x\dot{\psi}) + M_x
\end{aligned} \tag{4}$$

where, m is total mass of vehicle. I_i are the moment of inertia. The roll damping and the roll stiffness coefficients of the suspension system are k_ϕ and c_ϕ. F_{yf} et F_{yr} are respectively the longitudinal forces at the rear and front wheel. φ is the road crossfall and $M_x = h(F_{yf} + F_{yr})$.

Observer design

In this section, a second order sliding mode observer is proposed to observe the non-measurable dynamics and to identify the height gravity center. For that reason, the equations of vehicle model (4) are written in the following form:

$$\dot{x} = A(x)^{-1} F(x,u) \tag{5}$$

where $x = [v_y, \dot{\psi}, \dot{\phi}]$, $A(x) = \begin{bmatrix} m & 0 & 0 \\ 0 & I_{zz} & -I_{xz} \\ -mh & -I_{xz} & I_{xx} \end{bmatrix}$, $F(x,u) = \begin{bmatrix} -mv_x\dot{\psi} + F_{yf} + F_{yr} + mg\sin(\varphi) \\ L_f F_{yf} - L_r F_{yr} \\ M_x + mhv_x\dot{\psi} - k_\phi \phi - c_\phi \dot{\phi} \end{bmatrix}$

Let us take $\theta = \dot{\phi}$, the model (Eq. 5) can be written in the following state form:

$$\begin{bmatrix} \dot{\phi} \\ \dot{\theta} \\ \ddot{\psi} \\ \dot{v}_y \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{I_{zz}}{A}k_\phi & -\frac{I_{zz}}{A}c_\phi & 0 & 0 \\ -\frac{I_{xz}}{A}k_\phi & -\frac{I_{xz}}{A}c_\phi & 0 & 0 \\ 0 & 0 & -v_x & 0 \end{bmatrix} \begin{bmatrix} \phi \\ \theta \\ \dot{\psi} \\ v_y \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ \frac{mghI_{zz}}{A} & \frac{2hI_{zz} + I_{xz}L_f}{A} & \frac{2hI_{zz} + I_{xz}L_r}{A} \\ \frac{mghI_{xz}}{A} & \frac{2hI_{xz} + I_{xx}L_f}{A} & \frac{2hI_{xz} + I_{xx}L_r}{A} \\ g & 1/m & 1/m \end{bmatrix} \begin{bmatrix} \sin(\varphi) \\ F_{yf} \\ F_{yr} \end{bmatrix} \tag{6}$$

where $A = I_{zz}I_{xx} + I_{xz}^2$. For the observation of the state of this system we proceed by stages. Initially we consider the observation of the roll dynamics described by:

$$\begin{cases} \dot{\phi} = \theta \\ \dot{\theta} = \dfrac{I_{zz}}{A}(k_\phi \phi + c_\phi \theta) + H \end{cases} \tag{7}$$

where $H = A^{-1}(mghI_{xz}\sin(\varphi) + 2I_{xz}M_x + I_{xx}(L_f F_{yf} + L_r F_{yr}))$. As described in (Bouteldja, 2005), the proposed observer has the form:

$$\begin{cases} \dot{\hat{\phi}} = \hat{\theta} + \lambda_\phi sign(\phi - \hat{\phi}) \\ \dot{\hat{\theta}} = \dfrac{I_{zz}}{A}(k_\phi \phi + c_\phi \bar{\theta}) + \alpha\,[\lambda_\theta sign(\bar{\theta} - \hat{\theta})] \end{cases} \tag{8}$$

$$\bar{\theta} = \hat{\theta} + \lambda_\theta sign_{moy}(\phi - \hat{\phi}) \tag{9}$$

λ_ϕ is the observer gain. The additional variable θ is introduced to provide an estimate in finished time the roll speed. $sign_{moy}(\phi - \hat{\phi})$ represents the filtering of the function *sign*(.) (Utkin, 1995). The parameter α allows activating the observation of the roll speed $\hat{\theta}$ $(\alpha = 0 \text{ or } 1)$. For the study of convergence of the observer, we consider $\tilde{\phi} = \phi - \hat{\phi}$, the observation dynamics error is given by the following equation:

$$\dot{\tilde{\phi}} = \tilde{\theta} + \lambda_\phi sign(\tilde{\phi}) \tag{10}$$

with $\tilde{\theta} = \theta - \hat{\theta}$ who represents the error of observation of the roll speed.

Step 1 ($\alpha = 0$, convergence in a finite time t_1 of ϕ_1 to $\hat{\phi}_1$): For sufficiently large $\lambda_\phi > |\tilde{\theta}|$ the error of observation towards the sliding surface $\tilde{\phi} = 0$ is ensured in a finished time t_1 (in this step $\alpha = 0$; the switch term in second equation of the observer is not activate and θ is well limited). On the sliding surface, $\dot{\tilde{\phi}} = 0$, consequently the equation (10) becomes:

$$0 = \tilde{\theta} + \lambda_\phi sign_{eq}(\tilde{\phi}) \tag{11}$$

However the estimate of the average value $sign_{eq}(\tilde{\phi})$ can be obtained by filtering the high frequencies with a low-pass filter (Filippov, 1988), (Utkin, 1995).
In this case, $sign_{eq}(\tilde{\phi}) = sign_{moy}(\phi - \hat{\phi})$ thus starting from the equations (9) and (11) it is shown that $\bar{\theta} = \theta$. This estimate is obtained in finished time t_1.

Step 2 ($\alpha = 1$, convergence in a finite time $t > t_1$ of θ to $\hat{\theta}$): In this step, we suppose that conditions of attractively of the sliding surface $\tilde{\phi} = 0$ are always true and the observation then the roll speed is activated ($\alpha = 1$). In this step one thus has $\hat{\phi} = \theta$ and $\bar{\theta} = \theta$. The dynamics of the observation error of roll speed $\tilde{\theta}$ can be written:

$$\dot{\tilde{\theta}} = H - \lambda_\theta sign(\tilde{\theta}) \qquad (11)$$

where λ_θ is the observer gain. For values sufficiently large of $\Gamma_\phi > |H|$ then the observation error $\tilde{\theta} \to 0$ in finished time t_2. After $t > t_1$, the variables available are $\phi, \theta, \dot{\psi}$ and v_x. The model (6) becomes:

$$\begin{cases} \ddot{\psi} = A^{-1}(-I_{xz}k_\phi\phi - I_{xz}c_\phi\theta + mghI_{xz}\sin(\varphi) + (2hI_{xz} + I_{xx}L_f)F_{yf} - (2hI_{xz} + I_{xx}L_f)F_{yr}) \\ \dot{v}_y = -v_x\dot{\psi} + \dfrac{F_{yf} + F_{yr}}{m} + mg\sin(\varphi) \end{cases} \qquad (12)$$

We can thus propose the following observer:

$$\begin{aligned} \hat{\ddot{\psi}} &= A^{-1}(-I_{xz}k_\phi\phi - I_{xz}c_\phi\bar{\theta}) + \lambda_\psi sign(\dot{\psi} - \hat{\dot{\psi}}) \\ \dot{\hat{v}}_y &= -v_x\dot{\psi} + \lambda_{v_y} sign(v_y - \hat{v}_y) \end{aligned} \qquad (13)$$

To ensure the convergence of this observer in finite time, the gains are chosen respectively as follows: $\lambda_\psi > \left|A^{-1}(2hI_{xz} + I_{xx}L_f)F_{yf} - (2hI_{xz} + I_{xx}L_f)F_{yr}\right|$ and $\lambda_\psi > \left|\dfrac{F_{yf} + F_{yr}}{m} + mg\sin(\varphi)\right|$

3.3 Coupled Dynamics Observer

Dynamic model

In this configuration (see figure 8), The model used has 4 degrees of freedom, and 5 states are needed. The motions considered are: v_x (longitudinal velocity), v_y (lateral velocity), ψ (yaw), and ϕ (roll). The front wheel axle is denoted as f, and the rear wheel axle is denoted r. The equations of motion are described in (14), and the parameters are shown in figure 8. The road crossfall φ integrates in the model.

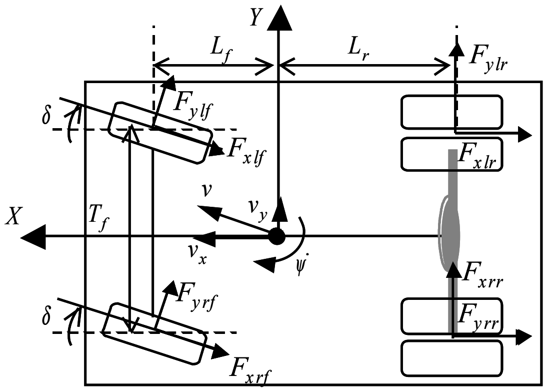

Figure 8 - Vehicle coupled model

$$A(x)\dot{x} = F(x,u) \qquad (14)$$

where $x = [v_x, v_y, \dot{\psi}, \dot{\phi}]$ and $u = [\delta, \varphi, F_{x,T}, F_{y,T}, M_T]$

$$A = \begin{bmatrix} m & 0 & -mh\phi & 0 & 0 \\ 0 & m & 0 & mh & 0 \\ -mh\phi & 0 & I_{zz} & I_{zz}\varphi - I_{xz} & 0 \\ 0 & mh & I_z\varphi - I_{xz} & I_{xx} - mh^2 & c_{\phi f} + c_{\phi r} \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix}, \quad F = \begin{bmatrix} F_{x,T} + m\dot{\psi}v_y + 2mh\dot{\phi}\dot{\psi} + mg\sin(\varphi) \\ F_{y,T} - m\dot{\psi}v_x + mh\phi\dot{\psi}^2 \\ M_T - mh\phi v_y\dot{\psi} \\ M \\ \phi \end{bmatrix}$$

and $F_{x,T} = F_{xf,l} + F_{xf,r} + F_{xr,l} + F_{xr,r} - (F_{yf,l} + F_{yf,r})\delta$, $F_{y,T} = F_{yf,l} + F_{yf,r} + F_{yr,l} + F_{yr,r} + (F_{xf,l} + F_{xf,r})\delta$

$$M_T = (F_{yf,l} + F_{yf,r})L_f + (F_{yr,l} + F_{yr,r})L_r + (F_{xf,l} + F_{xf,r})\delta L_f + (F_{xf,l} + F_{xr,l} + F_{yf,r}\delta - F_{xf,r} + F_{xr,r} + F_{yf,l}\delta)L$$

$$M = -mhv_x\dot{\psi} + (mh^2 + I_{xx} - I_{zz})\phi\dot{\psi}^2 - (c_{\phi f} + c_{\phi r})\dot{\phi} - (k_{\phi f} + k_{\phi r} - mgh)\phi$$

Observer design

The construction of this observer will be made in the same manner than in the preceding case, and we propose the structure of following observation (by posing $\theta = \dot{\phi}$):

$$\begin{cases} \dot{\hat{\phi}} = \hat{\theta} + \lambda_{\phi} sign(\tilde{\phi}) \\ \begin{bmatrix}\dot{\hat{v}}_x \\ \dot{\hat{v}}_y \\ \dot{\hat{\psi}} \\ \dot{\hat{\theta}}\end{bmatrix} = \begin{bmatrix} m & 0 & -mh\phi & 0 \\ 0 & m & 0 & mh \\ -mh\phi & 0 & I_{zz} & I_{zz}\varphi - I_{xz} \\ 0 & mh & I_z\varphi - I_{xz} & I_{xx} - mh^2 \end{bmatrix}^{-1} \begin{bmatrix} m\dot{\psi} v_y + 2mh\bar{\theta}\dot{\psi} \\ -m\dot{\psi} v_x + mh\phi\dot{\psi}^2 \\ mh\phi v_y \dot{\psi} \\ \bar{M} \end{bmatrix} + \alpha\,\lambda\, sign(\tilde{x}) \end{cases} \qquad (15)$$

with $\begin{vmatrix} \bar{\theta} = \hat{\theta} + \lambda_{\theta}\, sign_{moy}(\phi - \hat{\phi}) \\ \bar{M} = -mhv_x\dot{\psi} + (mh^2 + I_{xx} - I_{zz})\phi\dot{\psi}^2 - (c_{\phi f} + c_{\phi r})\bar{\theta} - (k_{\phi f} + k_{\phi r} - mgh)\phi \end{vmatrix}$.

The same steps described above were followed to prove the convergence in a finite time for this observer (15), with, $\alpha = diag\{\alpha_1, \alpha_2, \alpha_3, \alpha_4\}$, $\Gamma = diag\{\lambda_{v_x}, \lambda_{v_y}, \lambda_{\psi}, \lambda_{\theta}\}$ and $sign(\tilde{x}) = [sign(v_x - \hat{v}_x), sign(v_y - \hat{v}_y), sign(\dot{\psi} - \dot{\hat{\psi}}), sign(\bar{\theta} - \hat{\theta})]$.

4. Rollover Detection

Rollover detection and prediction is a topical area of research in the automotive industry (see, for example, (Dahlberg, 1999), (Winkler, 1999) for a good introduction to the problem) and several studies have recently been published. In order to detect rollover risk, it is necessary to know when a rollover is imminent on a route and every driving situation (Eger, 2000). Thus, some kind of rollover measure is needed. The approach used is inspired by Ackermann (Ackermann, 1998), and is based on load transfer ratio.

Load transfer occurs in connection with rollover events, the load transfer ratio is often used for rollover detection. The case of $R = \pm 1$, corresponding to the point at which one wheel begins to lose contact with the road surface, is often used as a critical situation which should be avoided in order to prevent rollover. It follows the same approach as in (Ackermann, 1998), and takes into account the road crossfall, the coefficient can be approximated by the following expression:

$$R = \frac{2(\dot{v}_y(h\cos(\phi + \varphi) + gh\sin(\phi + \varphi))}{T_r g \cos(\varphi)} \qquad (14)$$

It is based on different variables observed above, this criterion allows detecting or evaluating the risk of rollover.

5. Discussion and Simulation Results

To rebuild a total behavior of the vehicle starting from the observer based multi-models approach, the algorithm of program-ming of the System under Matlab/Simulink is presented in the form given figure (9).

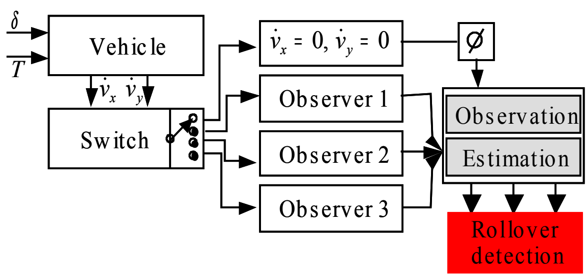

Figure 9 – milti-observer structure

In order to test and validate the robustness of our approach the some simulation results are presented and compared by data from another simulator "PROSPER". This simulator has already been validated experimentally. This is why, the PROSPER's simulation results are considered as references in our work. The model inputs are the road data and the steering angle of the heavy vehicle (see figure (10)).

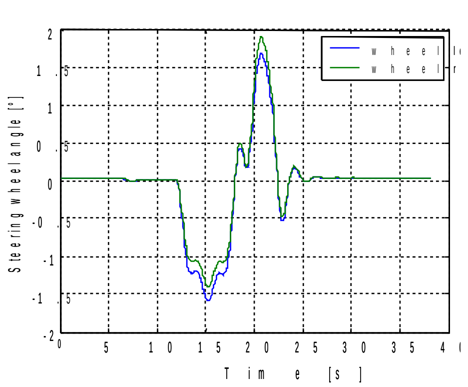

Figure 10 – Steering angle (input)

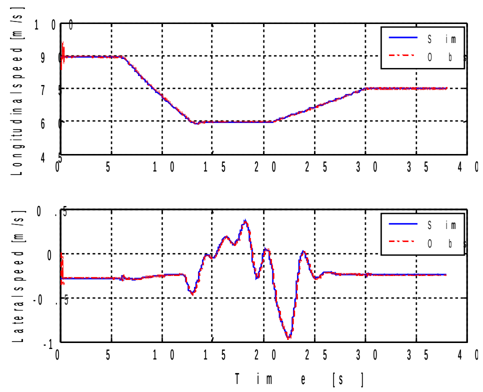

Figure 11 – longitudinal and lateral speed

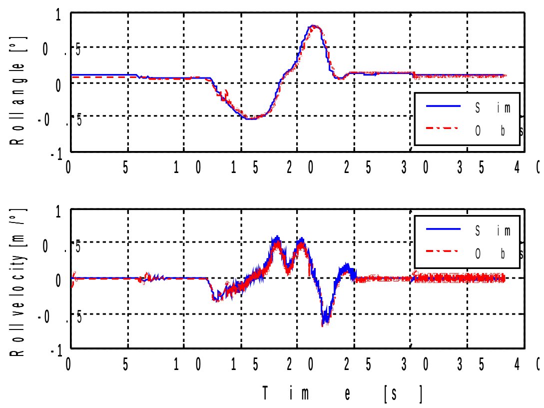

Figure 12 – Roll angle and roll ratio

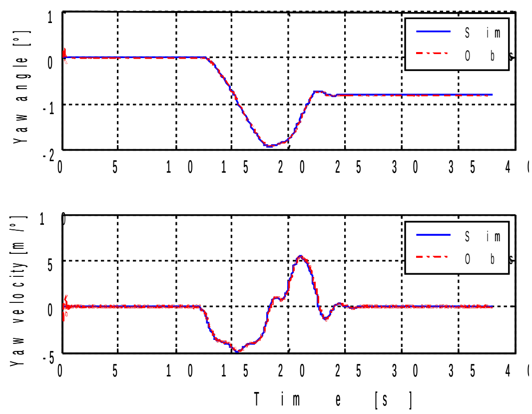

Figure 13 – Yaw angle and yaw ratio

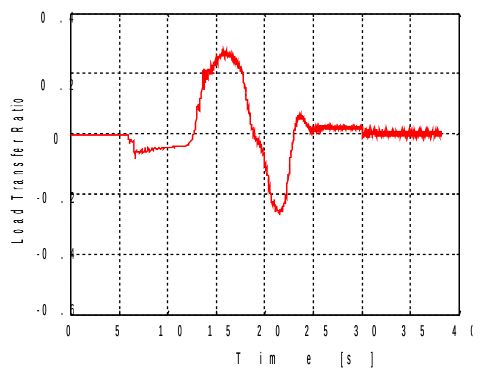

Figure 14 – *R* for the first test scenario

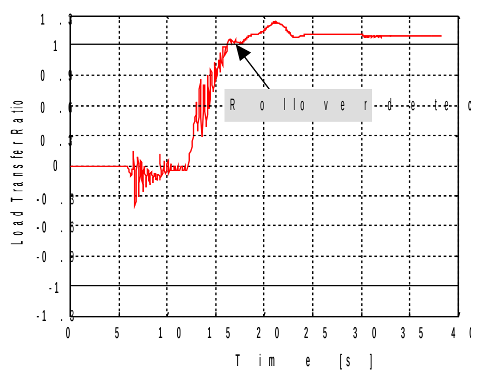

Figure 15 – *R* for the second test scenario

On the figure (11) we present the evolution of longitudinal and lateral speed during the simulation of the scenario. We find well the speed situations sequence described in the scenario.

Figures 12 show the comparison of the roll angles and roll velocity measured by PROSPER and observed by milti-obserder, also we find in figure 13 the yaw angles and yaw velocity. On these figures, we note that the observer multi-model reconstruct perfectly the dynamics of these variables. This allows us to determine the load transfer ratio (R) represented in figure 14. Since the R coefficient does not reach the value 1, we can conclude that no rollover risk is detected.

Now, to test the effectiveness of this approach for rollover detection, let us suppose a second scenario where the driver rolls on the same road at initial speed 110*km/h*. Figure 15 shows the evaluation of R. At this initial speed, the heavy vehicle overturns at the beginning of the curve.

6. Conclusion

In this paper, a stability analysis of heavy vehicle is realised towards the rollover risk. This study was completed by an analysis of driving situation on a route in accidents. Then a modeling of a sequence of driving situations is proposed taking into account infrastructure inputs. Thereafter, we associated each driving situation (model) an observer in order to estimate in real time the dynamics non-measurable and evaluate the rollover risk. These observers are based on sliding mode approach, known to be robust versus parametric uncertainties, modeling errors and disturbances. The passage (management) between different driving situations (models and observers) is provided by an automat. This approach is validated using the PROSPER simulator. The results obtained present a very weak error between variables simulated and those estimated. Then, from a rollover criterion (load transfer ratio) and dynamic estimated, the rollover risk of vehicle can be detected.

7. References

- Winkler, C. B., Ervin, R. D., (1999) “Rollover of Heavy Commercial Vehicles”, UMTRI-99-19, The University of Michigan, Transportation Research Institute, Michigan 48 109.
- Dahlberg, E., (1999), “Commercial Vehicle Rollover Prediction Using Energy Consideration”, Proc. of the European Automotive Congress, STA N° 99C203.
- Chen, B. C., Peng, H., (2001), “Differential Braking Based Rollover Prevention for Sport Utility Vehicles With Human in the Loop Evaluations”, Veh. Syst. Dynamics, Vol. N° 36.
- Ackermann, J., Odenthal, D., (1998), “Robust Steering Control for Active Rollover Avoidance of Vehicle With Elevated Center of Gravity”, Proc. International Conference of Advances in Vehicle Control and Safety, PP 118-123.
- Bouteldja, M., (2005) “Modélisation des interactions dynamiques poids lourd/ infrastructures pour la sécurité et les alertes”, PhD, Univ. de Versailles, France. 215pp.
- Bouteldja, M., M’Sirdi, N.K., Glaser, S., Dolcemascolo V., (2004) “Stability Analysis and Rollover Scenario Prediction For Tractor Semi-trailer”, International Conference on Advances in Vehicle Control and Safety AVCS04, Paper N° 136, Italy.
- Bouteldja, M., Dolcemascolo, V., Glaser, S., Jacob, B., (2003), “Generation of Warning for Rollover Avoidance”, IEEE, ITS200,Paper N° 2141, Spain.
- Eger, R., (2000), “Uwe Kiencke. Modeling of Rollover Sequences.1st IFAC Conference on Mechatronic Systems, Darmstadt.
- Gillespie, T., (1992) “Fundamentals of Vehicle Dynamics”, S. of Automotive Engineers.
- Filippov, A.F., (1988) “Differential Equations with Discontinuous Right-hand Sides” Dordrecht, The Netherlands: Kluwer Academic Publishers.
- V.I. Utkin and S. Drakunov, (1995), “Sliding mode observer". tutorial. IEEE Conf. on Decision and Control, pages 3376-3378.