Steering Control for Rollover Avoidance of Heavy Vehicle

H. Imine, L. Fridman and T. Madani

***Abstract*—The aim of presented work consists in developing an active steering assistance system in order to avoid the rollover of heavy vehicle. The proposed approach is applied on single body model of heavy vehicle presented in this paper. An estimator based on the high order sliding mode observer is developed in order to estimate the vehicle dynamics such as lateral acceleration limit and centre height of gravity. The lateral position and lateral speed are controlled using twisting algorithm in order to ensure the stability of the vehicle and avoid the accident. In the same time, the identification of unsprung masses and suspension stiffness parameters of the model has been computed in order to increase the robustness of the method. Some simulation and experimental results are given in order to show the quality of the proposed concept.**

***Index Terms*—Vehicle modelling, Rollover, Sliding mode observers, Sliding mode control, Estimation, Identification, Twisting algorithm, Steering control.**

I. INTRODUCTION

ROLL stability of the Heavy Vehicle (HV) is affected by the center height of gravity, the track width and the kinematics properties of suspensions. More destabilizing moment arises during the cornering maneuver when the center of gravity of the vehicle shifts laterally. The roll stability of the vehicle can be guaranteed if the sum of the destabilizing moment is compensated during a lateral manoeuvre.

Recently, several solutions have been proposed in order to reduce the number of HV rollover accidents. Several systems have been developed in order to assist the driver to avoid the rollover. Some of them are installed in the infrastructure before a dangerous curvature. In case of overspeed leading to rollover in cornering, a warning is sent to the driver in order to decrease the speed ([1], [2]).

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Other systems have been installed onboard the truck. They use informative measures coming from sensors, such as vehicle speed and lateral acceleration in order to send a warning signal to the driver when he goes beyond some risk thresholds ([3], [4]).

In case of active systems, the concept is to minimize the lateral acceleration by braking action, steering action, suspension action, by anti-roll action or combination of all ([5], [6], [7], [8], [9]).

However, most of existing methods require full information on the state that may limit their practical utility. In fact, even if all the state measurements are possible, they are typically corrupted by noises. Moreover, the increased number of sensors makes the overall system more complex in implementation and expensive in realization. Thus, design of an observer becomes an attractive approach to estimate the unmeasured states of the HV. A second order sliding mode observers, providing theoretically exact states observation, parameter identification and impact forces estimation, are presented in this work.

In most of recent researches, the parameters of the vehicle are supposed known and constant. Some of them were given by vehicles constructors and manufacturers, and others unknown parameters are taken from literature. In present work, suspensions stiffness and unsprung masses have been identified. This allows to improve the quality of the proposed system.

The impact forces affect vehicle dynamic performance and behaviour properties. These forces are very important to evaluate the rollover risk of heavy vehicle by computing the Load Transfer Ratio (LTR), study and evaluate the damage of the vehicle on the road or bridges and in order to control the weights of the vehicles ([10], [11], [12]). These forces can also be used in the control systems in order to assist the driver. However, the exiting sensors to measure these forces are very expensive and difficult to install [13].

Comparing to the previous works, the proposed method is based on sliding mode observer in order to estimate the vertical forces and in the same time identify the dynamic parameters of the vehicle ([14], [15], [16], [17], [18]).

Design of such observers requires a dynamic model of heavy vehicle which is build up in a first step of this work.

This model is coupled with an appropriate wheel road contact model. It has been validated using simulator [19] and by real measurements carried out with an instrumented tractor [20].

In this paper, an observer-controller law using the sliding mode technique is developed.

The proposed method has been adopted for different reasons:

- achieve good tracking of desired trajectories by ensuring the convergence of the lateral acceleration of the vehicle toward the estimated acceleration limit. This allows the limitation of the load transfer between the right and the left side of the vehicle to its limited value which is set to *0.9*.
- estimate the non measurable states of the HV.
- identify some unknown parameters of HV.

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 vertical forces and parameters identification are presented in section 3. Section 4 is devoted to steering control method design. Some experimental results are presented in section 5. Finally, some remarks and perspectives are given in a concluding section.

II. VEHICLE MODELLING

The vehicle studied in this paper and developed by [5] is a non-articulated heavy vehicle with two axles. Therefore, some assumptions were considered: the roll angle is assumed to be small, and the suspension and the tire dynamics are assumed to be linear. The corresponding model has five degrees of freedom and used to represent the roll and lateral dynamics as shown in figure 1.

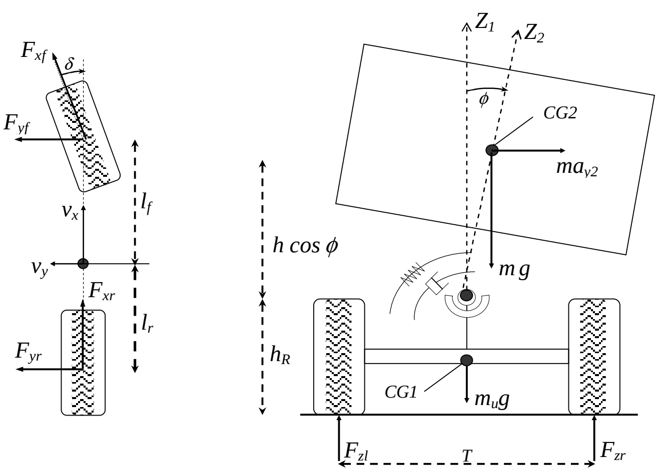

Fig.1. Heavy vehicle model

The vehicle consists of two bodies. Body 1 which is composed of two axles is represented by the unsprung mass m_u and center of gravity *CG1*. Body 2 is the sprung mass m and centre of gravity *CG2*. The centre CG1 is assumed to be in the road plane above *CG2*.

In this case, the lateral forces acting on the system can be linearized and computed as follows:

$$\begin{cases} F_{yr} = \mu\, c_r\, \alpha_r \\ F_{yf} = \mu\, c_f\, \alpha_f \end{cases} \tag{1}$$

Where c_f and c_r are respectively the front and the rear cornering stiffness after assuming the equality of the cornering stiffness for the two front wheels and the rear ones. The road adhesion is represented by the variable μ. In this work, the dry road is assumed ($\mu = 1$). The parameters α_f and α_r are respectively the front and rear tire slip angle computed by using the following formula:

$$\begin{cases} \alpha_r = -(\beta - l_r \dfrac{\dot\psi}{v}) \\ \alpha_f = \delta - (\beta + l_f \dfrac{\dot\psi}{v}) \end{cases} \tag{2}$$

where β is the side slip angle which is assumed equal for the two front wheels and the rear ones, l_f and l_r are respectively the distance from *CG1* to front axle and rear axle, v is the vehicle speed, $\dot\psi$ is the yaw rate and δ is the front wheel steering angle.

The model is derived using Lagrangian's equations:

$$M(q)\ddot q + C(q,\dot q)\dot q + Kq = F_g(q,u) \tag{3}$$

where $q = [q_1, q_2, y_l, \phi, \psi]^T$ represents the coordinate's vector composed of respectively left and right front suspension deflection, lateral displacement, roll angle and yaw angle, $M \in \Re^{5\times5}$ is the inertia matrix, $C \in \Re^{5\times5}$ is the matrix related to the damping effects, $K \in \Re^5$ is the springs stiffness vector, $F_g \in \Re^3$ is a vector of generalized forces and u is the system input.

The suspension is modelled as the combination of spring and damper elements as shown in the figure 2.

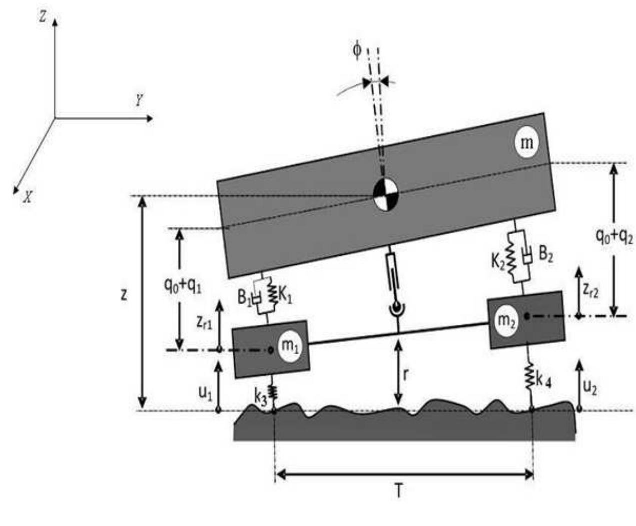

Fig.2. Front view of heavy vehicle model

The vertical acceleration of the chassis is obtained as following:

$$\ddot{z} = (k_1 q_1 + k_2 q_2 + (k_1 - k_2)\frac{T}{2}\sin(\phi))/m + (B_1\dot{q}_1 + B_2\dot{q}_2 - (B_1 - B_2)\frac{T}{2}\cos(\phi)\dot{\phi})/m \quad (4)$$

where ϕ is the tractor roll angle and T is the tractor track width, q_1 and q_2 are respectively the left and right front suspension deflection of the tractor, z is the vertical displacement of the tractor sprung mass (center of gravity height). The tractor chassis with the mass m is suspended on its axles through two suspension systems.

The suspensions stiffness and damping coefficients are represented respectively by $k_i, i = 1,2$ and $B_i, i = 1,2$ in the figure 2.

The tire is modelized by springs represented by $k_i, i = 3,4$ in the figure 2. The left and right wheels masses are respectively m_1 and m_2. At the tire contact, the road profile is represented by the variables u_1 and u_2 which are considered as heavy vehicle inputs. The variables z_{r1} and z_{r2} correspond respectively to the vertical displacement of the left and right wheel of the tractor front axle, which can be computed by means of using the following equations:

$$\begin{cases} z_{r1} = z - q_1 - \frac{T}{2}\sin(\phi) \\ z_{r2} = z - q_2 + \frac{T}{2}\sin(\phi) \end{cases} \quad (5)$$

The lateral and yaw accelerations are computed as follows:

$$\begin{cases} \ddot{y}_l = -\frac{(c_f + c_r)}{mv}\dot{y}_l + \frac{(-c_f l_f + c_r l_r) - mv^2}{mv^2}\dot{\psi} + \frac{c_f}{m}\delta \\ \ddot{\psi} = -\frac{\left(c_f l_f^2 + c_r l_r^2\right)}{I_z v}\dot{\psi} + \frac{l_f c_f}{I_z}\delta + \frac{(-c_f l_f + c_r l_r)}{I_z}\beta \end{cases} \quad (6)$$

where I_z is the inertia along the Z axis.

The vertical accelerations of the wheels are given by:

$$\begin{cases} \ddot{z}_{r1} = (k_1 q_1 - k_1\frac{T}{2}\sin(\phi) + B_1\dot{q}_1)/m_1 \\ -(B_1\frac{T}{2}\cos(\phi)\dot{\phi} + F_{zl})/m_1 \\ \ddot{z}_{r2} == (k_2 q_2 + k_2\frac{T}{2}\sin(\phi) + B_2\dot{q}_2)/m_2 \\ +(B_2\frac{T}{2}\cos(\phi)\dot{\phi} - F_{zr})/m_2 \end{cases} \quad (7)$$

The accelerations of suspensions are given by the following equation system:

$$\begin{cases} \ddot{q}_1 = (\frac{1}{m} - \frac{1}{m_1})k_1 q_1 + \frac{k_2 q_2}{m} + (k_1 - k_2)\frac{T}{2}\sin(\phi) \\ +(\frac{1}{m} - \frac{1}{m_1})B_1\dot{q}_1 + \frac{B_2\dot{q}_2}{m} - ((B_1 - B_2)\frac{T}{2}\cos(\phi)\dot{\phi})/m - \frac{F_{zl}}{m_1} \\ \ddot{q}_2 = (\frac{1}{m} - \frac{1}{m_2})k_2 q_2 + \frac{k_1 q_1}{m} + (k_1 - k_2)\frac{T}{2}\sin(\phi) \\ +(\frac{1}{m} - \frac{1}{m_2})B_2\dot{q}_2 + \frac{B_1\dot{q}_1}{m} - ((B_1 - B_2)\frac{T}{2}\cos(\phi)\dot{\phi})/m - \frac{F_{zr}}{m_2} \\ \ddot{\phi} = (k_1 q_1 - k_2 q_2 - (k_1 + k_2)\frac{T}{2}\sin(\phi) + B_1\dot{q}_1 \\ -B_2\dot{q}_2 - (B_1 + B_2)\frac{T}{2}\cos(\phi)\dot{\phi})\frac{T}{2}/I_x \end{cases} \quad (8)$$

where I_x is the inertia moment in the roll axis, F_{zl} and F_{zr} are respectively the vertical forces of the left and the right wheel, which are considered as unknown perturbations to be estimated.

These forces can be computed by using the following:

$$\begin{cases} F_{zl} = k_3(z_{r1} - u_1) \\ F_{zr} = k_4(z_{r2} - u_2) \end{cases} \quad (9)$$

III. Sliding Mode Observer Design

This section is devoted to observer design using sliding mode technique. In order to develop it, let us rewrite the equation (3) in state form as follows:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = f(x_1, x_2) + F_g(x_1, u) \\ y = x_1 \end{cases} \quad (10)$$

where $x = (x_1, x_2)^T = (q, \dot{q})^T \in \Re^{10}$ is the state vector and $y = q \in \Re^5$ is the measured outputs vector of the system, f is a vector of nonlinear analytical function and F_g is an unknown input vector computed as follows:

$$F_g = \begin{bmatrix} -F_{zl}/m_1 \\ -F_{zr}/m_2 \\ 0 \end{bmatrix}.$$

Before developing the sliding mode observer, one supposes:

1. The state is bounded.
2. The inputs of the system are bounded.

A. States observation

The second order observer developed in ([21], [22], [23]) has been adapted to the presented model in order to estimate in finite time states and to identify parameters.

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It has the following form:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + z_1 \\ \dot{\hat{x}}_2 = f(\hat{x}_1, \hat{x}_2) + z_2 \end{cases} \quad (11)$$

where $\hat{x}_1$ and $\hat{x}_2$ are respectively the estimations of vectors x_1 and x_2.

The correction variables z_1 and z_2 represent the output injections of the form:

$$\begin{cases} z_1 = \lambda\Delta(\tilde{x}_1)Sign(\tilde{x}_1) \\ z_2 = \alpha Sign(\tilde{x}_1) \end{cases} \quad (12)$$

where $\tilde{x}_1 = x_1 - \hat{x}_1 \in \Re^5$ is a vector of the state estimation error. The gains matrices (λ and α)$\in \Re^{5\times5}$, the matrix $\Delta(\tilde{x}_1)$ and the vector "*Sign*" are defined as follows:

$$\begin{cases} \lambda = diag\{\lambda_1, \lambda_2, ...\lambda_5\} \\ \alpha = diag\{\alpha_1, \alpha_2, ...\alpha_5\} \\ \Delta(\tilde{x}_1) = diag\left\{ \left|\tilde{x}_{1_1}\right|^{1/2}, \left|\tilde{x}_{1_2}\right|^{1/2}, ...\left|\tilde{x}_{1_5}\right|^{1/2} \right\} \\ Sign(\tilde{x}_1) = \left[sign(\tilde{x}_{1_1}), sign(\tilde{x}_{1_2}), ...sign(\tilde{x}_{1_5})\right]^T \end{cases} \quad (13)$$

The dynamics estimation errors are calculated as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - \lambda\Delta(\tilde{x}_1)Sign(\tilde{x}_1) \\ \dot{\tilde{x}}_2 = f(x_1, x_2) - f(\hat{x}_1, \hat{x}_2) \\ + F_g(x_1, u) - \alpha Sign(\tilde{x}_1) \end{cases} \quad (14)$$

where $\tilde{x}_2 = x_2 - \hat{x}_2 \in \Re^5$ is the estimation error of the vector x_2.

Considering the accelerations of the system as bounded, the elements of the diagonal matrix α can be selected, satisfying the following inequality:

$$\alpha_{ii} > 2\left|\dot{\hat{x}}_{2_i}\right|, \; i=1...5 \quad (15)$$

On the other hand, from [24], the elements of the diagonal matrix λ can be selected as:

$$\lambda_{ii} > \sqrt{\frac{2}{\alpha_{ii} - 2\left|\dot{\hat{x}}_{2_i}\right|} \frac{(\alpha_{ii} + 2\left|\dot{\hat{x}}_{2_i}\right|)(1+p_i)}{1-p_i}}, i = 1,..,5 \quad (16)$$

where p_i □]0, 1[are some constants to be chosen (see proof in [24]).

In order to study the observer stability, first, the convergence of $\tilde{x}_1$ in finite time t_0 is proved. Then, some conditions about $\tilde{x}_2$ in order to ensure its convergence to *0* are deduced. Therefore, for $t \geq t_0$, the surface $\tilde{x}_2 = 0$ is attractive, leading $\hat{x}_2$ to converge towards x_2 satisfying the inequalities (15) and (16).

The convergence proof of the second order observer can be found in [24].

B. Vertical forces estimation and parameters identification

In order to estimate the vertical forces and to identify parameters of the system, let us rewrite the system (10) by reducing its order, as following:

$$\begin{cases} \dot{x}_{11} = x_{21} \\ \dot{x}_{21} = a_1\varphi(x_1, x_2) - F_{zl} / m_1 \\ \dot{x}_{12} = x_{22} \\ \dot{x}_{22} = a_2\varphi(x_1, x_2) - F_{zr} / m_2 \end{cases} \quad (17)$$

with $x_{11} = q_1, x_{21} = \dot{q}_1, x_{12} = q_2, x_{22} = \dot{q}_2$.

The unknown vectors of parameters represented by a_1 and a_2 and the vector φ are defined as follows:

$$a_1 = \left[k_1\left(\frac{m_1 - m}{m_1}\right) \quad k_2 \right]; a_2 = \left[k_1 \quad k_2\left(\frac{m_2 - m}{m_2}\right) \right];$$

$$\varphi = \left[\frac{q_1}{m} \quad \frac{q_2}{m} \right]^T.$$

To simplify the system, the vector of unknown parameters a is computed as following:

$$a = \begin{cases} = [a_{11} \quad a_{21}] = \left[k_1\left(\frac{m_1 - m}{m_1}\right) \quad k_2 \right] \\ = [a_{12} \quad a_{22}] = \left[k_1 \quad k_2\left(\frac{m_2 - m}{m_2}\right) \right] \end{cases}$$

In order to estimate the vertical forces and to identify the unknown inputs, the second order sliding mode observer defined in (11) is rewritten as:

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 + \lambda\Delta(\tilde{x}_1)sign(\tilde{x}_1) \\ \dot{\hat{x}}_2 = \bar{a}\varphi(\hat{x}_1, \hat{x}_2) + \alpha sign(\tilde{x}_1) \end{cases} \quad (18)$$

where $(x_1, x_2) = (x_{11}, x_{21})$ or $(x_1, x_2) = (x_{12}, x_{22})$.

The variable $\bar{a}$ represents a vector of the nominal values of the vector parameters a.

In this case, the dynamic estimation errors are calculated as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - \lambda\Delta(\tilde{x}_1)sign(\tilde{x}_1) \\ \dot{\tilde{x}}_2 = \tilde{a}\varphi(x_1, \tilde{x}_2) + F_{gi}(x_1, u) - \alpha sign(\tilde{x}_1) \end{cases} \quad (19)$$

where $\tilde{a} = a - \bar{a}$ is the estimation error of the vector parameters a, $F_{gi} = -F_{zl} / m_1$ or $F_{gi} = -F_{zr} / m_2$.

After convergence of the observer (18), the variable $\tilde{x}_2$ convergences towards 0 in finite time $t \geq t_0$.

In this case, and from (19), one obtains:

$$z_2 = \alpha sign(\tilde{x}_1) = \tilde{a}\,\varphi(x_1, \tilde{x}_2) + F_{gi}(x_1, u) \quad (20)$$

Theoretically, the equivalent output injection is the result of an infinite switching frequency of the discontinuous term $\alpha sign(\tilde{x}_1)$. Nevertheless, the realization of the observer produces a high switching frequency which makes the application of a filter necessary.

To eliminate the high frequency component, a filter of the following form is used:

$$\tau\dot{\bar{z}}_2(t) = \bar{z}_2(t) + z_2(t) \quad (21)$$

where $\tau \in R$ and $h << \tau << 1$, being h a sampling step.

The variable z_2 is then rewritten as follows:

$$z_2(t) = \bar{z}_2(t) + \xi(t) \quad (22)$$

where $\bar{z}_2(t)$ is the filtered version of $z_2(t)$ and $\xi(t) \in R$ is the difference caused by the filtration.

Nevertheless, as it is shown in ([25], [26]) that:

$$\lim_{\substack{\tau \to 0 \\ h/\tau \to 0}} \bar{z}_2(\tau, h) = z_2(t) \quad (23)$$

Thus, it is possible to assume that the equivalent output injection is equal to the output of the filter.

1) Vertical forces estimation

In order estimate the vertical force $F_{gi}(x_1, u)$, the vector of parameters a is supposed to be known. In this case $\tilde{a} = 0$. Therefore and using the equation (20), the vertical force is obtained as follows:

$$F_{gi} = \alpha sign(\tilde{x}_1) \quad (24)$$

One reminds that this vector is composed of the forces F_{zl} or F_{zr} which can be computed using the system equation (9). One can then mention the advantages of the proposed method as following:

1. The measuring of the road profiles u_1 and u_2 is not necessary.
2. The estimation of the vertical displacements of the wheels and its derivative are also not necessary to obtain.

2) Parameters identification

To identify the parameters of the system, the impact forces vector is supposed to be zero $F_g(x_1, u) = 0$. That means that the road profile is supposed to be very small. There are no irregularities on the road that can affect vertically the vehicle. In this case, the vertical displacements of the wheels are close to zero.

Using the equation (20), it follows:

$$z_2 = \tilde{a}\,\varphi(x_1, x_2) = \alpha sign(\tilde{x}_1) \quad (25)$$

Considering the unknown parameters vector $\tilde{a}$ as a constant vector, and in order to identify it, a linear regression algorithm, namely the least square method is applied ([17], [18]).

The time integration is given by:

$$\frac{1}{t}\int_0^t z_2(\sigma)\varphi(\sigma)^T d\sigma = \tilde{a}\,\frac{1}{t}\int_0^t \varphi(\sigma)\varphi(\sigma)^T d\sigma \quad (26)$$

The vector $\tilde{a}$ is then estimated by:

$$\hat{\tilde{a}} = \left[\int_0^t z_2(\sigma)\varphi(\sigma)^T d\sigma\right]\left[\int_0^t \varphi(\sigma)\varphi(\sigma)^T d\sigma\right]^{-1} \quad (27)$$

where $\hat{\tilde{a}}$ is the estimation of $\tilde{a}$.

Let us define $\Gamma = \left[\int_0^t \varphi(\sigma)\varphi(\sigma)^T d\sigma\right]^{-1}$

Its derivative gives $\dot{\Gamma} = -\Gamma\,\varphi(\sigma)\varphi(\sigma)^T\,\Gamma$.

The derivative of the vector $\hat{\tilde{a}}$ using the equation (27) gives:

$$\dot{\hat{\tilde{a}}} = \left[\int_0^t z_2(\sigma)\varphi(\sigma)^T d\sigma\right]\dot{\Gamma} + z_2\varphi^T\Gamma \quad (28)$$

Replacing $\dot{\Gamma}$ by its value given before, and using the equation (28), it follows:

$$\begin{aligned}\dot{\hat{\tilde{a}}} &= -\hat{\tilde{a}}\,\varphi\varphi^T + z_2\varphi^T\Gamma \\ &= (-\hat{\tilde{a}}\,\varphi + z_2)\varphi^T\Gamma\end{aligned} \quad (29)$$

(29)

This ensures the asymptotic convergence of $\hat{\tilde{a}}$ toward $\tilde{a}$ and therefore, this allows the identification of the real value of the vector a. In order to obtain the unsprung masses and after identification of $k_1=a_{12}$ and $k_2=a_{21}$, the expression of the variable a defined earlier is used here.

The identified values are then as follows:

$a_{11}=k_1(m_1-m)/m_1$ and $a_{22}=k_2(m_2-m)/m_2$. Finally, the unsprung masses are deduced as following:

$m_1 = mk_1/(k_1 - a_{11})$ and $m_2 = mk_2/(k_2 - a_{22})$.

IV. STEERING CONTROL

Rollover risk evaluation is based on load transfer ratio (LTR) which corresponds to the difference in tire normal forces acting on each side of the vehicle. It can be computed as follows:

$$LTR = \frac{F_{zr} - F_{zL}}{F_{zr} + F_{zl}} = \frac{2m_2}{mT}\left((h_R + h\cos\phi)\frac{a_{y2}}{g} + h\sin\phi\right) \quad (30)$$

where $a_{y2} = a_y - h\ddot{\phi}$ is the lateral acceleration of the sprung mass and $a_y = v(\psi + \beta)$ is the lateral acceleration of the unsprung mass and T is the track width, F_{zl} and F_{zr} are normal forces acting on respectively the left and the right side of the vehicle. When *LTR* is equal to *0*, the heavy vehicle has a stable roll dynamic. The risk becomes higher when this indicator goes towards *±1*. Both extreme values characterize wheel lift-off. The same model developed before has been used in this section. However, in order to perform the controller, only the lateral part of the model is important. Therefore, the suspension deflections variables are not used here.

In this section, an active steering control is developed in order to assist the vehicle in case of rollover risk. In addition to the steering angle commanded by the driver and noted δ_d, an auxiliary steering angle δ_a is set by an actuator. Therefore, the control input $u = \delta = \delta_a + \delta_d$. In this work, the limit value of LTR is set to *0.9*. This chosen value is used arbitrary, less than the limit *1*, in order to give sufficient time to the controller/driver to react, before one of wheels lifts-off the road. In this case, the controller has time to avoid the rollover before to obtain high values of lateral accelerations. In the case of small roll angle, one can assume that:

$$h\sin\phi < \left((h_R + h\cos\phi)\frac{a_{y2}}{g}\right)$$

From equation (30), the acceleration limit can be obtained and approximated by:

$$a_{y2\lim} \approx \frac{0.9Tgm}{2m_2 H}; H = h + H_R \quad (31)$$

The aim of the developed steering control is to ensure the convergence of the lateral acceleration a_{y2} of the vehicle to its acceleration limit a_{y2lim}. This will allow the limitation of load transfer between the right and the left side of the vehicle to its limited value *0.9*.

The steering control diagram is shown in the figure 3:

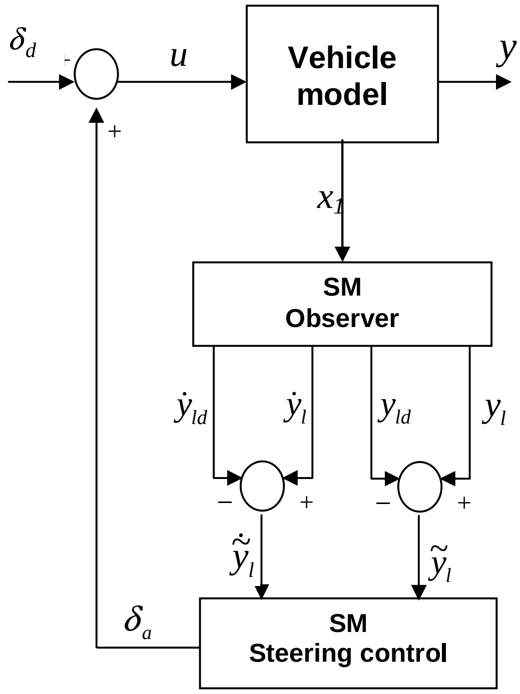

Fig.3. Controller diagram

In order to be able to achieve the aim, let us consider the following sliding mode surface:

$$S = \dot{\tilde{y}}_l + \eta\tilde{y}_l \quad (32)$$

where $\dot{\tilde{y}}_l = \dot{y}_l - \dot{y}_{ld}$ and $\tilde{y}_l = y_l - y_{ld}$ are respectively the lateral speed error and lateral offset, η is a positive constant, $\dot{y}_{ld}$ and y_{ld} are respectively the desired velocity and the desired lateral displacement obtained respectively by first and double integration of the acceleration limit a_{y2lim}, computed earlier using the equation 31.

Deriving the variable *S* and using the equation 6, it follows:

$$\dot{S} = \ddot{\tilde{y}}_l + \eta\dot{\tilde{y}}_l = \frac{c_f}{m}u + (\eta + a_{55})\dot{\tilde{y}}_l \quad (33)$$

The equivalent control u_{eq} *is* obtained by resolving the equation $\dot{S} = 0$ as follows:

$$u_{eq} = -\frac{m}{c_f}(\eta + a_{55})\dot{\tilde{y}}_l - \delta_d \quad (34)$$

The proposed control is based on Super-twisting algorithm. That algorithm has been developed to control dynamic systems in order to avoid chattering.

In this case, the trajectories are characterized by twisting around the origin, as shown in the figure 4.

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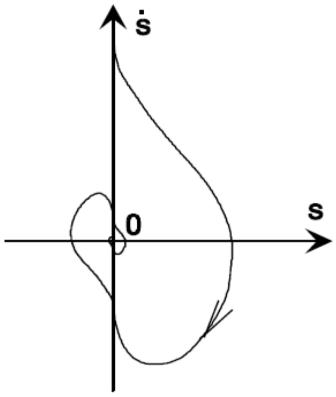

Fig.4. Super twisting algorithm trajectory

The continuous control law u is composed of two terms. The first is defined by means of its discontinuous time derivative, while the other is a continuous function of sliding variable.

Therefore, the proposed control law is defined as follows:

$$\begin{cases} \delta_a = u_{eq} - G_1 |S|^{1/2} sign(S) + u_1 \\ \dot{u}_1 = -G_2 sign(S) \end{cases} \tag{35}$$

where G_1 and G_2 are the positive control gains.

The convergence proof and analysis of the used super twisting algorithm can be found in ([14], [15]).

V. Experimental results

The tractor of the figure (5) has been instrumented on behalf of VIF project [27]. The vehicle was equipped with different sensors to measure the dynamic states of the vehicle such as gyrometers, accelerometers, LVDT, Lasers…etc as shown in the figure 6. The installation and positions of these sensors in the tractor are illustrated in the figure 7.

Fig.5. Instrumented vehicle

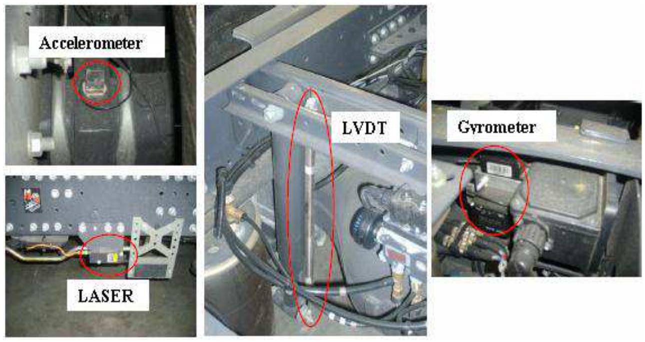

Fig. 6. Sensors in the vehicle

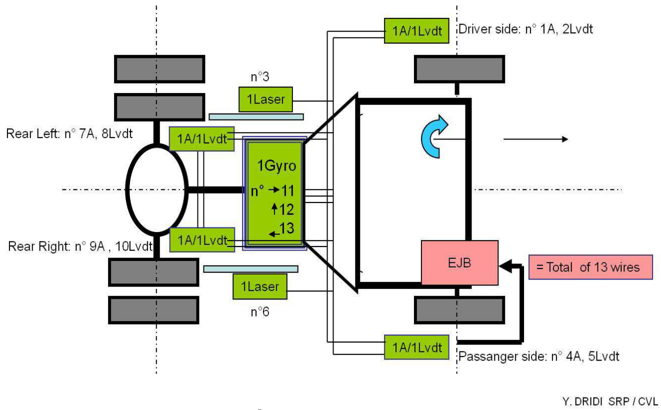

Fig.7. Sensors positions in the tractor

As shown in the figure 6, different sensors have been installed in order to validate and calibrate the whole system:

- four accelerometers installed on the chassis in order to measure the vertical accelerations of wheels,
- four sensors LVDT in order to measure the suspensions deflections (q_1 and q_2 have been used in the observer).
- three axial gyroscopes installed on the chassis, to measure the angular rates (roll, pitch and yaw rates). The roll angle is deduced from integration of the measured roll rate or by computation using the following formula:

$$\phi = \arcsin\left(\frac{q_1 - q_2}{T}\right)$$

In this work, this last method was preferred and used in order to avoid errors which can result from roll rate integration.

- two lasers in order to measure the vertical displacements of the chassis.

Many tests and scenarios have been performed with the instrumented vehicle driving at various speeds. In this work, the results obtained from zigzag and ramp tests are presented in order to show the robustness of the estimation and identification. While, the presented results for steering control to avoid rollover are obtained from simulation.

The nominal dynamic parameters and the static vertical forces are measured before the tests. The static values of front left and front right vertical forces are respectively *24200N* and *25250N*. The static values of rear left and rear right vertical forces are respectively *9450N* and *12050N*.

The nominal values of the unsprung masses m_1 and m_2 are respectively *100kg* and *95kg* and the nominal values of suspensions stiffness k_1 and k_2 are respectively *194680N/m* and *188540 N/m*.

A. Zigzag test

Zigzag Test is presented in this section. This test is very important in order to show the reaction of controller when the driver changes at short time, often and brutally the direction of the vehicle. The simulation results are shown in the figure 8. One remarks that the rollover risk appears at *14s* and lasts approximately*1s*.

The value of the steering angle at this time is about *-0.06rad*. This value decreases until *-0.25rad* when the controller is OFF (dashed line). Otherwise, and when the controller is ON (solid line), this value is decreased to *-0.15rad*. At the critical time *14s*, and in order to stabilize the value of the LTR at its limit *0.9*, the computed lateral acceleration limit is around $-8m/s^2$. When the controller is activated (solid line), the lateral acceleration is still equal to $-8m/s^2$. Otherwise, if the controller is still OFF, the lateral acceleration decreases until $-12m/s^2$ (dashed line). During the risk time interval [14, 15]s, the controller is activated and the load transfer ratio (LTR) is stabilised to *-0.9* (solid line). Without control, the rollover risk decreases and the LTR tends towards *-1.8* (dashed line). The same situation occurs at the time interval [16, 17]s. In this case, the LTR is also stabilised to the limited value *0.9* when the controller is active.

Otherwise, the LTR tends towards *1.5*. The use of sliding mode observer allowed a well and quick estimation of roll angle as presented in the figure 8. Without control (dashed line), the absolute value of this variable increases from *0.06* to *0.11rad* during the first time interval [14, 15]s and from *0.06* to *0.09rad* in the time interval [16, 17]s. In the figure 9, the identification results are shown. The suspensions stiffness k_1 and k_2 are identified with success. In effect, compared to their nominal values of respectively *194680N/m* and *188540 N/m*, these parameters have been identified with errors which can be neglected as shown in the figure 10. The percentage of the error is less than *0.025%*. However, one notes some variations at the time interval [13, 20]s. This is due to the fact that at this time, the driver changes abruptly the direction of the vehicle.

In the right side of the figure 10, the unsprung masses m_1 and m_2 have been identified with a percentage of error less than *0.1%* compared to their nominal values of respectively *100kg* and *95kg*.

B. Ramp test

The results of ramp test are presented in this section. The steering angle increased until a maximum value of *0.5rad*, during *5s* before to stabilize, as shown in the figure 11.

During the time interval [0, 1.5] s, there is no rollover risk, as one can remark in the left side of the figure 11. Although during this time, the estimated steering angle coming from control block is the same than that coming from the model without control. The LTR in this case is less than *1* as shown in the right side of the figure 11.

After *1.5s*, the risk appears and the LTR reaches the limit value *1*, which corresponds to the situation where one of the wheels of the same axle lifts-off. In order to avoid this risk situation, the controller is then activated in order to avoid the rollover of the vehicle.

With the active controller, the steering angle is reduced and its value becomes less than the original one coming from the model. In this case, the lateral acceleration limit is estimated and shown in the figure 11 (solid line). Without control (dashed line), the lateral acceleration increases until $4m/s^2$. Otherwise, when the control is activated (solid line), this acceleration does not exceed $3m/s^2$. Therefore, the value of LTR is reduced and it becomes less than *1*. On the other hand, the sliding mode observer allows to estimate in finite time and quickly the different variables of the system.

In the figure 11, one notices also the well estimation of side slip angle when it's compared to the variable coming from the model. The identification results are shown in the figure 12. In this test, the suspensions stiffness k_1 and k_2 have been identified better than in the first one. Indeed, one notes that the signals are smooth and only small variations of these identified values occur around their nominal values of respectively *194680N/m* and *188540N/m*.

In the right side of the figure 12, the identified unsprung masses m_1 and m_2 are shown. The variations of these parameters occur around their nominal values respectively of *100kg* and *95kg*, with an error quite close to *0* as shown in the figure 13.

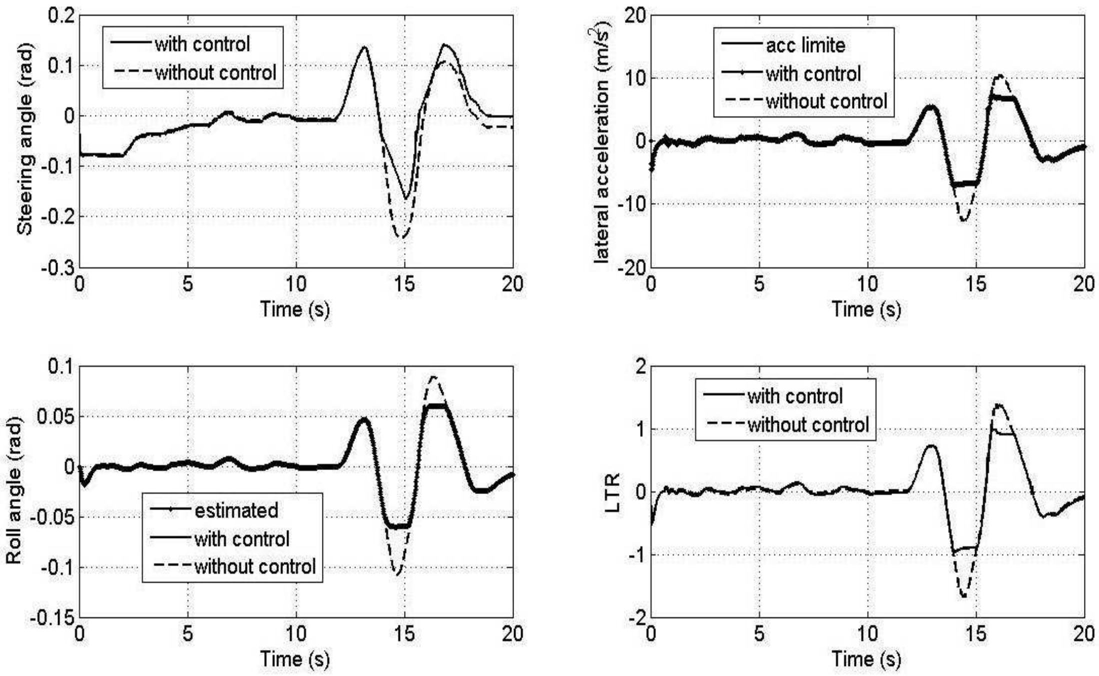

Fig.8. Simulation results for a driver steering input zigzag

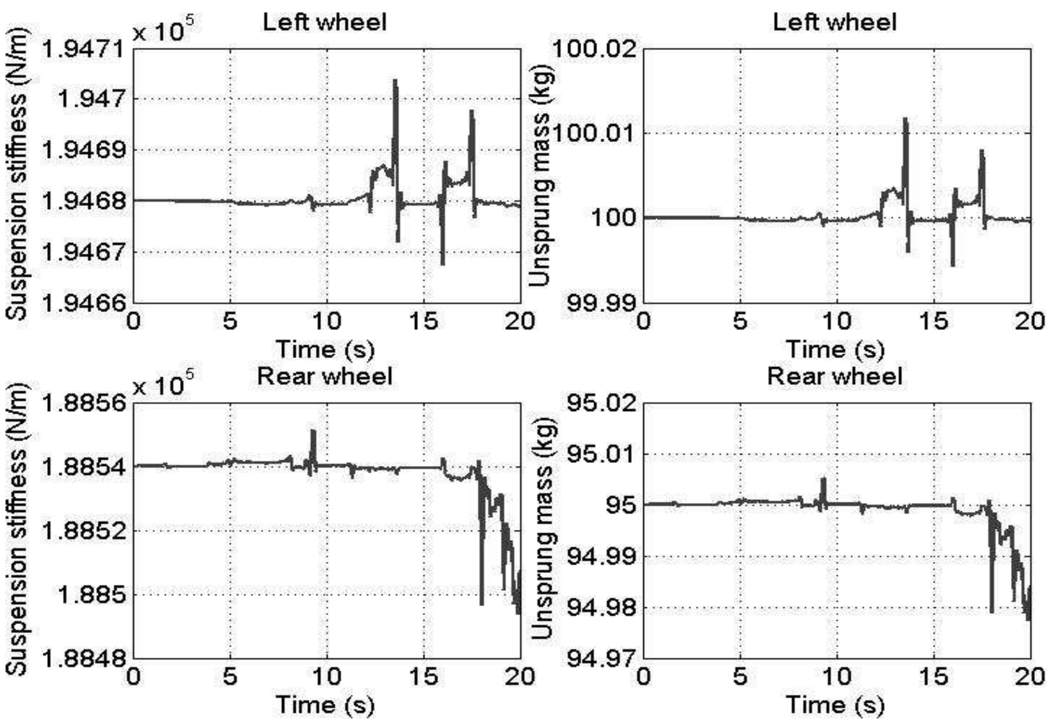

Fig.9. Identification results for a driver steering input zigzag

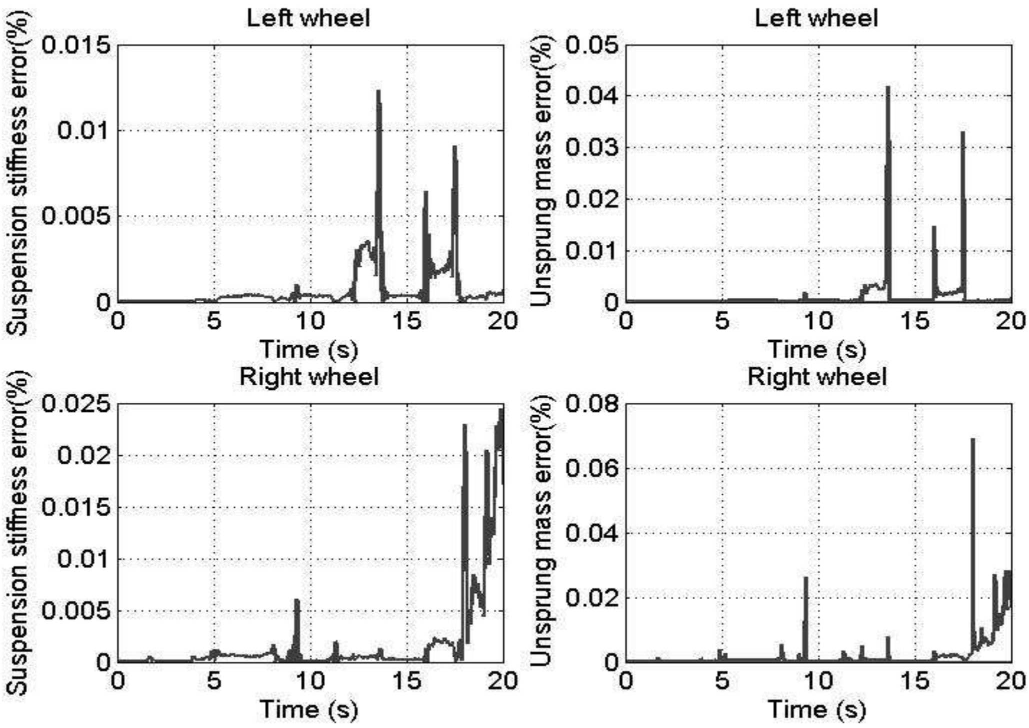

Fig.10. Identification errors for a driver steering input zigzag

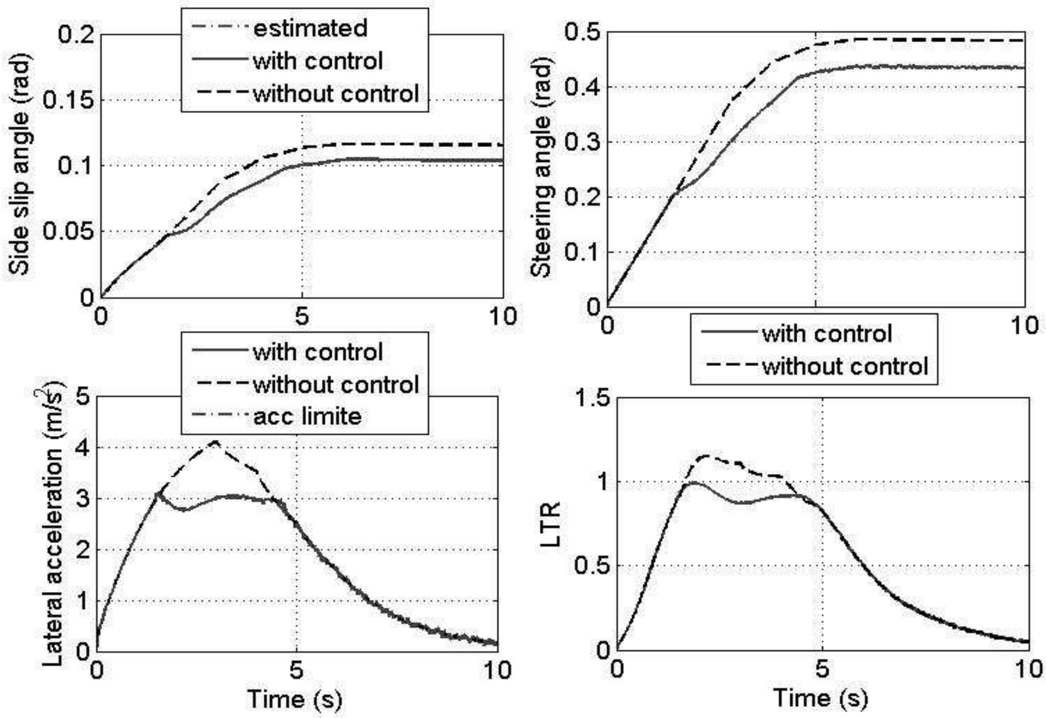

Fig.11. Simulation results for a driver steering input ramp

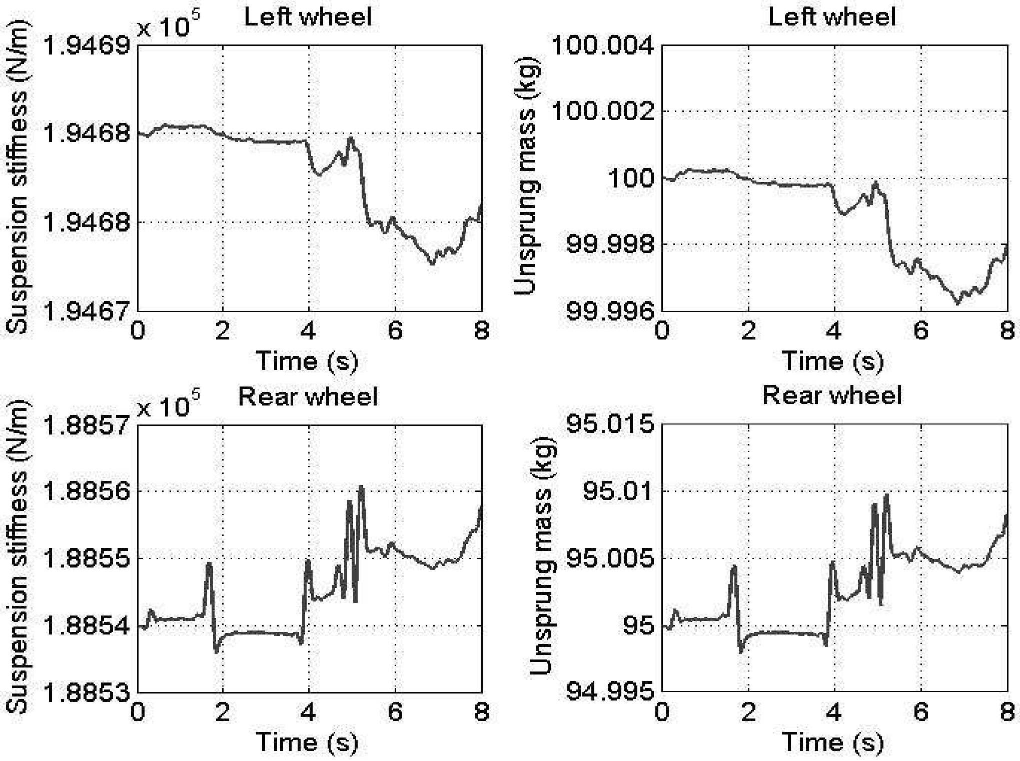

Fig.12. Identification results for a driver steering input ramp

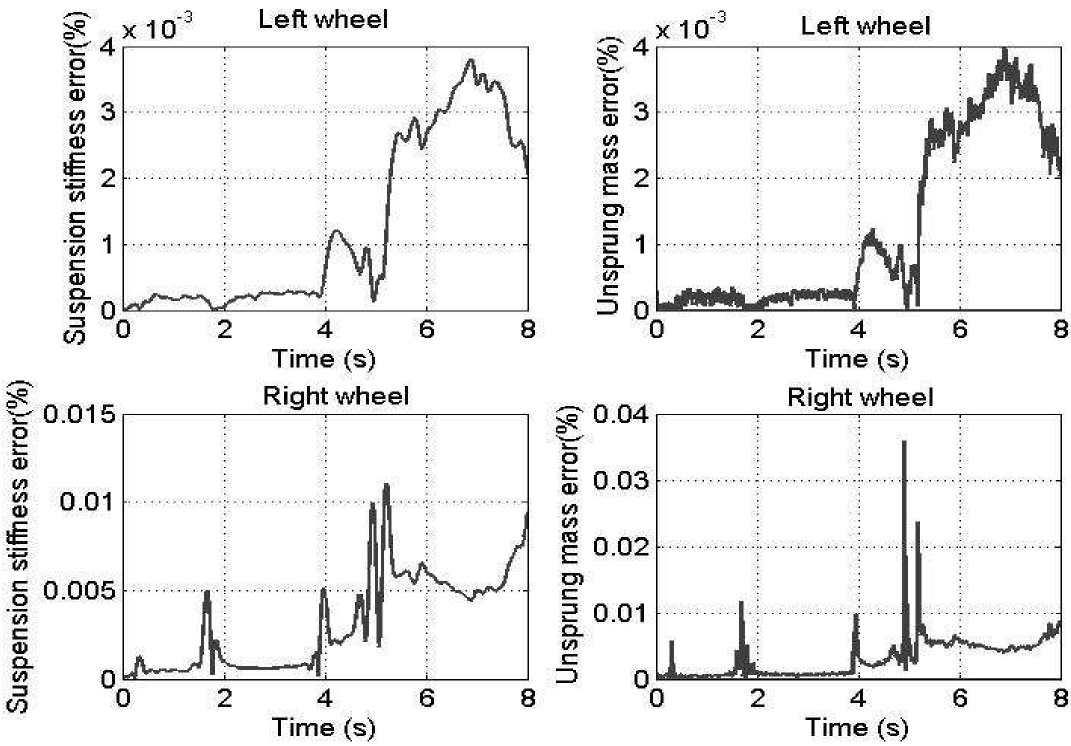

Fig.13. Identification errors for a driver steering input ramp

VI. CONCLUSION

In this paper, a steering control system was developed in order to avoid the rollover of heavy vehicle. It is based on control of lateral acceleration of the vehicle. Previously, an estimator based on sliding mode was implemented. It made it possible to estimate the non measured dynamics of the vehicle. Lateral acceleration and roll angle estimation are presented in this work. Comparing to existing method, the proposed approach is based on robust controller-estimator.

The identification of unsprung masses and suspension stiffness has been computed in order to increase the robustness of the controller.

A real zigzag scenario is tested and presented. The results showed that this system is effective and made it possible to control the vehicle and to avoid its rollover. The lateral acceleration limit is stabilized. Therefore, the steering angle of the vehicle is modified in order to force this latter to stay on a safety trajectory. The LTR is maintained around a maximum value of 0.9. The experimental tests done on an instrumented truck showed the quality of this approach since the convergence of the observer is quick and it is done in finite time.

The identification of the parameters was a success with small variations around the nominal values at the time interval [13 20]s. These variations are due to the abrupt direction change of the driver.

A second ramp test is presented in this paper. Compared to the first one, the identification process has been carried out with errors quite close to zero and the controller is activated quickly in order to avoid the accident.

The originality of this approach is the use of the equivalent control, which provides a linear regression algorithm in order to identify the unknown parameters of the system. An example of identification of the unsprung mass and the stiffness is given in this paper. The result is of quality.

In the future work, it will be interesting to test this approach in real time with an instrumented tractor semi-trailer and using the dynamo wheel in order to measure the impact forces, which will be the reference for a better validation of the estimation. In this case, the controller will be applied in order to stabilize the semi trailer which is, in this type of heavy vehicle, the first unit that can lose the control.

In this paper, the unsprung masses and the suspension stiffness have been identified. In the future work, one will focus on the identification of other important dynamic parameters, namely damping coefficients, roll and yaw inertia moments.

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