Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

Vehicle Parameter Estimation and Stability Enhancement ⋆

Hassan Shraim * **Mustapha Ouladsine** * **Leonid Fridman** **

* *Laboratory of sciences of informations and of systems, LSIS UMR 6168 University of Paul Czanne, Aix-Marseille III Av escadrille de Normandie Niemen 13397 Marseille Cedex 20*
** *Department of Control, Division of Electrical Engineering,Engineering Faculty, National Autonomous University of Mexico(UNAM), Ciudad Universitaria, 04510,D.F, Mexico*

Abstract: In this paper, high order sliding mode observers are used to estimate tires longitudinal forces, vehicle side slip angle and velocities. Two types of high order sliding mode observers are used. A third order one is used to estimate the longitudinal forces, while for the estimation of the vehicle side slip and velocities, an observer based on the super twisting algorithm is proposed. Validations with the simulator VE-DYNA pointed out the good performance and the robustness of the proposed observers. Controller design for the braking is accomplished using a reduced state space model representing the movement of the vehicle center of gravity in the (X, Y) plane. Driver's reactions are taken into account. The performance of the closed loop system is carried out by means of simulation tests.

1. INTRODUCTION

In the recent years, important research has been undertaken to investigate the safe driving conditions in both normal and in critical situations. Safe driving requires the driver to react extremely quickly in dangerous situations, which is generally very difficult unless for experts, that result the instability of the system. Consequently, the improvement of the vehicle dynamics by active chassis control is necessary for such catastrophic situations. Increasingly, commercial vehicles are being fitted with micro processor based systems to enhance the safety and to improve driving comfort.

Considerable attention has been given to the development of the control systems over the past few years, authors have investigated and developed different methods and different strategies for enhancing the stability and the handling of the vehicle such as, the design of the active automatic steering You et al. [1999] and the wheel ABS control Petersen [2003], Unsa et al. [1999], or the concept of a four-wheel steering system (4WS) which has been introduced to enhance vehicle handling.

In terms of vehicle safety, and in order to develop a control law for the vehicle chassis, accurate and precise tools such as sensors should be implemented on the vehicle, to give a correct image of its comportment. Difficulties in measuring all vehicle states and forces, due to high costs of some sensors, or the non existence of others, make the design and the construction of observers necessary. In the field of automotive engineering, the estimation of vehicle side slip angle and wheel interaction forces with the ground are very important, because of their influence on the stability of the vehicle. Many researchers have studied and estimated vehicle side slip angle, using a bicycle model as in Stephant [2004], or by using an observer with adaptation of a quality function as in Von Vietinghoff et al. [2005] which requires a certain linearised form of the model. Moreover, an extended Kalman filter is used for the estimation of wheel forces Samadi et al. [2001].

The problem of observation has been actively developed within Variable Structure Theory using sliding mode approach. Sliding mode obsevers (see for example Edwards et al. [1998], V. Utkin et al. [1999] and Poznyak [2003]) are widely used due to their attractive features;

- Insensitivity (more than robustness) with respect to unknown inputs;
- Possibilities to use the values of the equivalent output injection for the unknown inputs identification (V. Utkin et al. [1999]);
- Finite time convergence to the reduced order manifold.

In Levant [2003] robust exact differentiators were designed ensuring finite time convergence to the real values for derivatives, as an application of super-twisting algorithm. New generation of observers based on the high order sliding mode differentiators are recently developed (see Barbot et al. [2004] and Bejarano et al. [2007]). Those observers:

- provide a convergence to the exact values of states variables;
- allow the exact identification the unknown inputs without filtration;
- guarantee the best possible accuracy of the state estimation w.r.t. to the sampling steps and deterministic noises.

⋆ This work was supported by ST Microelectronics, Rousset-France and the Conseil Gnral des Bouches-du-Rhne.).

In this paper we propose a robust sliding mode controller together with robust high order sliding mode observers for estimation of the states, forces and parameters, and for the stabilization of the vehicle on a reference trajectory in order to ensure safety in the critical situations.

In the estimation part of this paper, the model is decoupled into two parts in order to insure the observability. First, the longitudinal forces are estimated by a high order sliding mode observer based on the dynamical equations of the wheels. These forces are then injected, and supposed as known inputs, in the reduced model of the vehicle center of gravity. Based on this reduced model, the vehicle side slip angle and velocity are estimated by a sliding mode observer based on the super twisting algorithm.

The paper is organized as follows: In section 2, estimation of longitudinal forces is proposed using high order observers. In this section two cases are considered depending on the wheel measurements. In Section 3, the estimation of vehicle side slip angle and vehicle velocity is shown. In section 4, a controller design is presented. In section 5 simulations are shown and finally a conclusion is presented.

In this study it is supposed that we can measure:

- angular positions of the wheels (in the case that they are not measured and the angular velocities are measured we have also proposed a solution)
- front wheel angle;
- yaw rate

and it is required to estimate:

- angular velocity of the wheels;
- contact forces;
- vehicle velocity;
- vehicle side slip angle.

It is important to note that the torques applied at the wheels are estimated and then injected in the model as known input.

2. ESTIMATION OF LONGITUDINAL FORCES

In order to identify the longitudinal forces, we use the following model for the dynamics of the wheels (figure 1) :

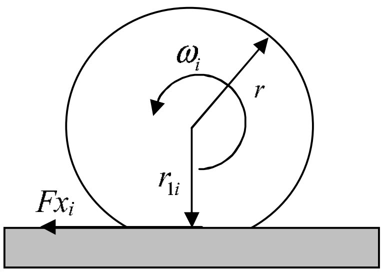

Fig. 1. Wheel and its contact with the ground

$$I_{ri}\dot{\Omega}_i = -r_{1i}Fx_i + torque_i i = 1:4 \quad (1)$$

In fact, the design of an observer for the estimation of the longitudinal forces is related to the available measurements. For this reason we have two possibilities

2.1 If the wheel angular position is available

If the wheels angular positions are available then, the dynamical equation of the wheel can be written in the following form:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= f(x_1, x_2, u) \end{aligned} \quad (2)$$

where x_1 and x_2 are respectively θ_i (which is measured) and Ω_i (to be observed) (appears implicitly in Fx_i), and u is $torque_i$. In fact this torque may be measured as shown in Rajamani et al. [2006], and it can also be estimated by estimating the motor and the braking torque, the motor torque may be estimated as in Khiar et al. [2006], while the braking torque is estimated by measuring the hydraulic pressure applied at each wheel (existing on the most of the vehicles).

A Third Order sliding mode Observer The proposed third order sliding mode observer has the following form:

$$\begin{aligned} \dot{\hat{\theta}}_i &= \underbrace{\hat{\Omega}_i + \lambda_0|\theta_i - \hat{\theta}_i|^{2/3}\, sign(\theta_i - \hat{\theta})}_{v_o} \\ \dot{\hat{\Omega}}_i &= \frac{torque_i}{I_{ri}} + z_1 \end{aligned} \quad (3)$$

where $\hat{\theta}_i$ and $\hat{\Omega}_i$ are the state estimations of the angular positions and the angular velocities of the four wheels respectively, z_1 is the correction factor based on the super-twisting algorithm having the following forms:

$$\begin{aligned} z_1 &= \underbrace{\lambda_1|\hat{\Omega}_i - v_o|^{1/2}\, sign(\hat{\Omega}_i - v_o) + Z_1}_{v_1} \\ \dot{Z}_1 &= \lambda_2\, sign(Z_1 - v_1). \end{aligned} \quad (4)$$

with$\lambda_2 = 3\sqrt[3]{\lambda_0}$, $\lambda_1 = 1.5\sqrt{\lambda_0}$ et $\lambda_0 = 2f^+$. The sliding occurs in (32) in finite time (see Levant [2003]). In particular, this means that the longitudinal forces can be estimated in a finite time without the need for low pass filter of discontinuous output injections.

2.2 If the wheel angular velocity is available

If the angular position is not available, then we propose a sliding mode observer based on the super twisting algorithm. This observer has the following form :

$$\begin{cases} \dot{\hat{\Omega}}_i = \dfrac{torque_i}{I_{ri}} + \left|\Omega_i - \hat{\Omega}_i\right|^{1/2} sign(\Omega_i - \hat{\Omega}_i) + Z_1 \\ \dot{Z}_1 = \Delta\, sign(\Omega_i - \hat{\Omega}_i) \end{cases} \quad (5)$$

2.3 Estmation of the side slip angle and the velocity of the vehicle

In this part, a sliding mode observer based on the super-twisting algorithm is used to estimate vehicle velocity and side slip angle. The model of the vehicle is a non-linear model and it can be written as follows :

$$\dot{v}_{COG} = \frac{1}{M}\left(cos(\beta)\sum F_L - sin(\beta)\sum F_S\right) \quad (6)$$

$$\dot{\beta} = \frac{1}{Mv_{COG}}\left(cos(\beta)\sum F_S - sin(\beta)\sum F_L\right) - \dot{\psi} \quad (7)$$

With

$\sum F_L = F_{xwind} + cos(\delta_f)(Fx_1 + Fx_2) + cos(\delta_r)(Fx_3 + Fx_4) - sin(\delta_f)(Fy_1 + Fy_2) - sin(\delta_r)(Fy_3 + Fy_4)$

And

$\sum F_S = sin(\delta_f)(Fx_1 + Fx_2) + sin(\delta_r)(Fx_3 + Fx_4) + cos(\delta_f)(Fy_1 + Fy_2) + cos(\delta_r)(Fy_3 + Fy_4)$

$$\begin{aligned} I_Z\ddot{\psi} = t_f\,&\{cos(\delta_f)(Fx_2 - Fx_1) + sin(\delta_f)(Fy_1 - Fy_2)\} + \\ L_1\,&\{sin(\delta_f)(Fx_2 + Fx_1) + cos(\delta_f)(Fy_1 + Fy_2)\} + \\ L_2\,&\{sin(\delta_r)(Fx_3 + Fx_4) - cos(\delta_r)(Fy_3 + Fy_4)\} + \\ t_r\,&\{cos(\delta_r)(Fx_4 - Fx_3) + sin(\delta_r)(Fy_3 - Fy_4)\} \end{aligned} \tag{8}$$

This model can be rewritten as a state space model having the following form :

$$\dot{x} = f(x,u) = A(x) + B(x)u \tag{9}$$

where

$$x = \begin{bmatrix} v_{COG} & \beta & \dot{\psi} \end{bmatrix} \tag{10}$$

the input:

$$u = [F_{x1} \quad F_{x2} \quad F_{x3} \quad F_{x4}] \tag{11}$$

and the measurement vector

$$y = \begin{bmatrix} \dot{\psi} \end{bmatrix} \tag{12}$$

Before the design of the sliding mode observer for the model of equation 9, the observability of the model must be investigated and tested. The observability definition is local and uses the Lie derivative. It is a function of the state trajectory and the inputs to the model. For the system described by equation 9 the jacobian of the observability function has a full rank and hence the system is observable.

So, the proposed sliding mode observer based on the hierarchical super twisting algorithm is:

$$\begin{cases} \dot{\hat{x}} = f(\hat{v}_{COG}, \hat{\beta}, \dot{\psi}, u) + Z_2 \\ Z_2 = \Delta_1 |(\dot{\psi} - \dot{\hat{\psi}})|^{1/2}\, sign(\dot{\psi} - \dot{\hat{\psi}}) + Z_1 \\ \dot{Z}_1 = \Delta\, sign(\dot{\psi} - \dot{\hat{\psi}}) \end{cases} \tag{13}$$

Where Δ and Δ_1 are the gains of the sliding mode observer.

SIMULATIONS RESULTS The estimated variables, $\hat{v}_{COG}$ and $\hat{\beta}$ are compared to that provided by VE-DYNA. The operation conditions are given by a variation in δ_f figure 10 and $torque_i$ figure 11, which constitute a significant driving situation. Fx_i are estimated from the third order sliding mode observer presented in the previous section and injected in this model. In figures 12, 13, 14 and 15, we see the observed $\hat{v}_{COG}$, $\hat{\beta}$ and $\hat{\dot{\psi}}$ and those provided by VE-DYNA. The rapid convergence pointed out the good performance of the proposed observer.

3. CONTROLLER DESIGN

In this section a sliding mode control strategy is presented. For the design process, the model presented by equation (9) is used. As we have described, the state vector is $x = [v_{COG}, \beta, \dot{\psi}]^T$ and the input $u = [Fx_1, Fx_2, Fx_3, Fx_4]^T$ (the steering angle is supposed given by the driver). As the only available actuators are the brakes, then only braking

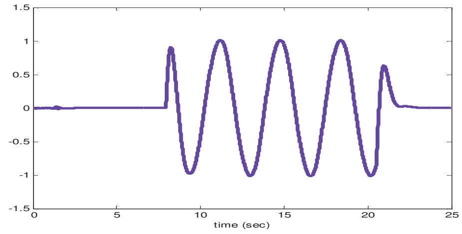

Fig. 2. Front steering angle (rad)

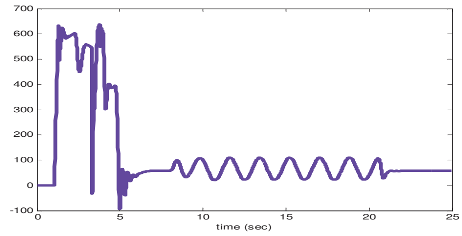

Fig. 3. Motor and braking torque (N.m) applied at the two rear wheels

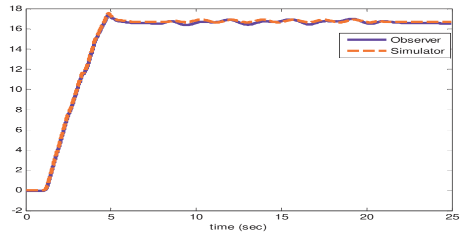

Fig. 4. Estimated vehicle velocity (m/sec) and that of the simulator VE-DYNA

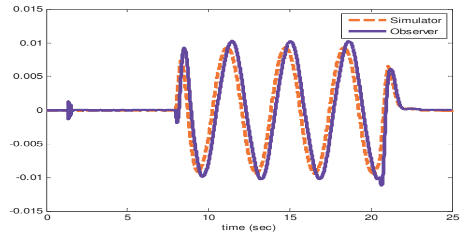

Fig. 5. Estimated side slip angle (rad) using sliding modes and that of the simulator VE-DYNA

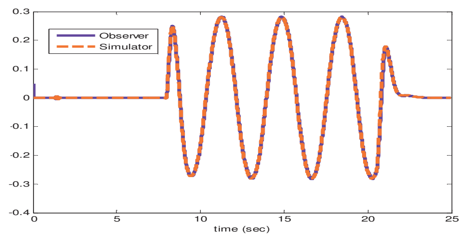

Fig. 6. Reconstructed yaw rate (rad/sec) and that of the simulator VE-DYNA

torques can be generated by the controller Uwe et al. [2005]. These braking torques are directly related to the hydraulic pressure applied at each wheel Uwe et al. [2005]:

$$torque_{controller} = -r_{1i} k_{bi} p_{BRi} \qquad i = 1:4 \tag{14}$$

To design a sliding mode controller, a sliding surface is proposed as:

$$s = x - x_{ref} \tag{15}$$

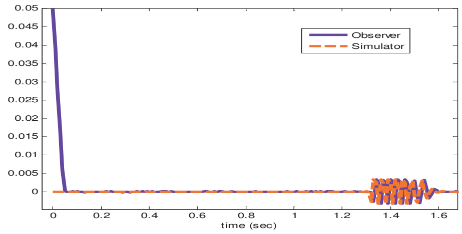

Fig. 7. Zoomed reconstructed yaw rate (rad/sec) and that of the simulator VE-DYNA

With $x_{ref} = [v_{COGref}, \beta_{ref}, \dot{\psi}_{ref}]^T$ represents the reference states which are derived as shown in Uwe et al. [2005].

The control objective is to derive the state vector x to the reference state vector x_{ref}. In order to ensure the stability, let us suppose that Lyapunov candidate is given by V. Utkin [1992]:

$$V = \frac{1}{2} s^T s > 0 \tag{16}$$

its derivative can be written as:

$$\dot{V} = s^T \dot{s} \tag{17}$$

where:

$$\dot{s} = \dot{x} - \dot{x}_{ref} \tag{18}$$

by substituting (9) into (18) we get:

$$\dot{s} = A(x) + B(x)u - \dot{x}_{ref} \tag{19}$$

Then we have:

$$\dot{V} = s^T \left(A(x) + B(x)u - \dot{x}_{ref} \right) \tag{20}$$

let us suppose the following control law:

$$u = -K sign(B(x)^T s) \tag{21}$$

substituting u in the above equation, we get: $\dot{V} = s^T \left(A(x) - B(x) K sign(B(x)^T s) - \dot{x}_{ref} \right)$.
In order to satisfy the conditions of stability in the Lyapunov sense, $\dot{V}$ should be negative, that means:
$s^T \left(A(x) - B(x) K sign(B(x)^T s) - \dot{x}_{ref} \right) < 0$
since K is scalar, we can write:

and then $k > \frac{s^T (A(x) - \dot{x}_{ref})}{|s^T (B(x))|}$
a necessary condition for the existence of the control law is that $s^T B(x) \neq 0$. Where $sgn(s)$ is a sign function which equals to 1 when $s > 0$ and -1 if $s < 0$. The chattering of the function $sgn(s)$ may be reduced by $sat(\frac{s}{\Phi})$, where Φ is a design parameter denoting the boundary layer thickness Slotine et al [1991].

4. SIMULATION RESULTS

In this section, computer simulations are carried out to verify the effectiveness of the proposed observers and controller. Simulation is made, in which driver's inputs are given by the simulator VE-DYNA. The driver wants to move on a 'chicane' (double lane) trajectory described by driver's steering angle figure 8 and wheel torques figure 9. The inputs of the driver are considered and the reference trajectory, to know if we are in the safe region is found as in Uwe et al. [2005]. Based on the driver's inputs and the calculated reference trajectory, we found that the in some situations the vehicle goes its safe region. The controller reacts as shown in figure 10. In this figure 10, we see three curves, the reference yaw rate, the yaw rate for the system without the controller and that with the sliding mode controller, it is seen how the controller pushes the controlled yaw rate to its reference value. In figure 11, three curves also are shown for the side slip angle which are: the reference side slip angle, the side slip angle without the controller and that with the sliding mode control. In figure 12, the response of the controller is shown, it is seen that in the normal cases where the side slip angle and the yaw rate are in their safety regions, the controller gives zero, other wise, when they exceed their limits, the controller tries to regulate this problem giving different torques on the different wheels.

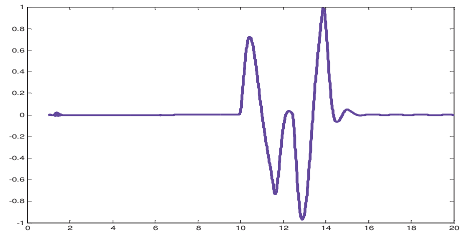

Fig. 8. Steering angle (double lane trajectory)

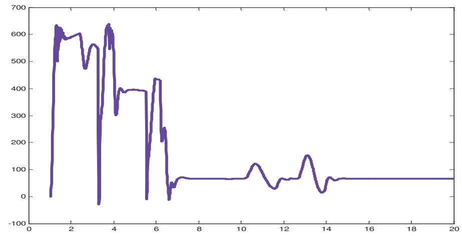

Fig. 9. Torque applied at the rear wheels (double lane trajectory)

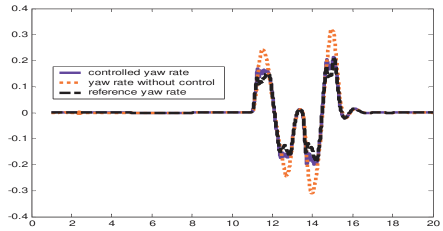

Fig. 10. Yaw rate with and without the controller and the reference yaw rate

5. CONCLUSIONS

Sliding mode observers are proposed in this work to estimate vehicle parameters and states which are not easily measured. These observers have shown a short time of convergence and robustness in the automotive applications that we have proposed. The validation of the proposed observers is realized by comparing the observers output with the outputs of the simulator VE-DYNA, reasonable and acceptable results have been shown. In the second

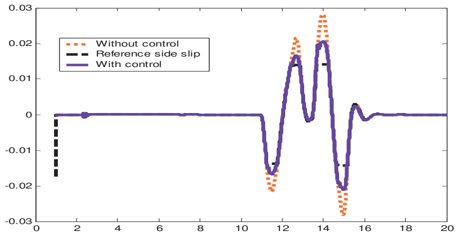

Fig. 11. Side slip angle with and without the controller and the reference side slip angle

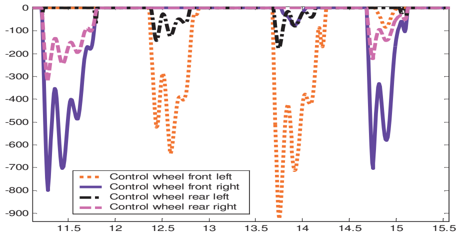

Fig. 12. The four outputs of the controller

part of this work, sliding mode controller is designed. This controller shows its strong and rapid reactions on the braking systems in the critical situations where we need the controller to work.

REFERENCES

Yi et al. Adaptive emergency braking control with underestimation of friction coefficient *IEEE Trans. on Control Systems Technology*, volume 10, pages 381–392, 2002.

K. D. Young, V. I. Utkin, and U. Ozguner. A Control Engineer's Guide to sliding mode Control *IEEE Trans on Control Systems Technology*, pages 328–342, 1999.

S.-S. You and Y.-H. Chai. Multi-objective control synthesis: an application to 4WS passenger vehicles *IEEE Trans on Control Systems Technology*, volume 7, pages 363-390, 1999.

A. Von Vietinghoff, M. Hiemer and K. Uwe. Non-linear observer design for lateral vehicle dynamics *16th IFAC World Congress*, Prague, Czech Republic, 2005.

K. Uwe, and L. Nielsen. Automotive control system *Springer-Verlag*, Berlin, Germany, 2nd edition, 2005.

V. Utkin. Sliding mode control design principles and application to electrical drives *IEEE Trans. Ind. Electron.*, pages 23-36, 1992.

V. Utkin, J. Guldner and J. Shi. Sliding Modes in Electromechanical Systems *Taylor and Francis*, London, 2nd edition, 1999.

J. Stephant. Contribution l'tude et la validation exprimentale d'observateurs appliqus la dynamique du vhicule *Thesis presented to have the doctor degree from the UTC,University of technology Compigne*, 2004.

C. Unsa and P. Kachroo. Sliding mode measurement feedback control for antilock braking systems *IEEE Transactions on Control System Technology*, volume 7, 1999.

J. J. Slotine and W. Li. Applied nonlinear control *Prentice-Hall,Inc.,Englewood Cliffs*, NJ, USA, 1991.

H. Shraim, M. Ouladsine, M. El. Adel and H. Noura. Modeling and simulation of vehicles dynamics in presence of faults *16th IFAC World Congress*, Prague, Czech Republic, pages 687–692 , 2005.

H. Shraim, M. Ouladsine and M. El. Adel. A New Nonlinear Control Strategy for a vehicle trajectory tracking in the presence of faults *44th IEEE Conference on Decision and Control and European Control Conference*, pages 1994-1999 , 2005.

R. Rajamani, D. Piyabongkarn, J., Y. Lew and J. A. Algorithms for real time estimation of individual wheel tire road friction coefficients *Proceedings of the American Control Conference (ACC)*, pages 4682–4687 , 2006.

B. Samadi, R. Kazemi, K. Y. Nikravesh and M. Kabganianl. Real-time estimation of vehicle state and tire friction forces *Proceedings of the American Control Conference (ACC)*, pages 3318-3323 , 2001.

A.S. Poznyak. Stochastic output noise effects in sliding mode estimations *International Journal of Control*, volume 2, pages 986–999 , 2003.

C. Edwards and S.K. Spurgeon. Sliding Mode Control *Taylor and Francis*, London, 1998.

I. Petersen. Wheel slip control in ABS brakes using gain scheduled optimal control with constraints *thesis submitted for the degree of doctor engineer Department of Engineering Cybernetics, Norwegian University of Science and Technology Trondheim*, Norway, 2003.

L. Fridman and A. Levant. Higher order sliding modes *W.Perruquetti and J.P. Barbot, editors, Sliding Mode Control in Engineering*,New York:Marcel Dekker, pages 986–999 , 2002.

M. Hiemer et al. Cornering stiffness adaptation for improved side slip angle observation *Proceedings of the First IFAC Symposium on advances in Automotive Control (AAC04)*,New York:Marcel Dekker, pages 685–690 , 2004.

D. Khiar, J. Lauber, T. Floquet, T. Guerra, G. Coline and Y. Chamillard. Estimation of the instantaneous torque of a gasoline engine *CIFA, In french*, 2006.

A. levant. Higher order sliding modes, differentiation and output control *International Journal of Control*, volume 76, pages 924–941, 2003.

J. Barbot and T. Floquet. A sliding mode approach of unknown input observers for linear systems *43 th IEEE Conference on Decision and Control*, volume 76, pages 1724-1729, 2004.

F. J. Bejarano, L. Fridman and A. Poznyak. Exact State Estimation for Linear Systems with Unknown Inputs based on a hierarchical super twisting algorithm *International Journal of Robust And Nonlinear Control.*, in press, 2007.

Table .1. Nomenclature.

Symbol	Physical signification
Ω_i	angular velocity of the wheel
M	total mass of the vehicle
r_i	radius of the wheel i
COG	center of gravity of the vehicle
r_{1i}	dynamical radius of the wheel i
Fx_i	longitudinal force applied at the wheel i
Fy_i	lateral force applied at the wheel i
$torque_i$	Engine torque-Braking torque
I_Z	moment of inertia around the Z axis
ψ	yaw angle
$\dot{\psi}$	yaw velocity
δ_f	front steering angle
δ_r	rear steering angle
Vx	longitudinal velocity of the center of gravity
Vy	lateral velocity of the center of gravity
I_{ri}	moment of inertia of the wheel i
v_{COG}	total velocity of the center of gravity
L_1	distance between COG and the front axis
L_2	distance between COG and the rear axis
L	$L_1 + L_2$
h_{COG}	height of COG
t_f	front half gauge
t_r	rear half gauge
l	$t_f + t_r$
F_{xwind}	air resistance in the longitudinal direction
F_{ywind}	air resistance in the lateral direction
A_L	front vehicle Area
ρ	air density
C_{aer}	coefficient of aerodynamic drag
β	side slip angle at the COG
p_{BRi}	braking pressure at the wheel i
k_{bi}	brake coefficient of the wheel i

Table .2. parameters.

Parameter	Value	Parameter	Value
M	1296 Kg	L_1	0.97 m
L_2	1.53 m	r_{1i}	0.28 m
I_Z	1750	H	0.52 m
I_{ri}	0.9 Kg.m^2	t_f	0.7 m
t_r	0.75 m	l_o	-0.03 m
l_1	0.12 m	ρ	1.25 Kg/m^3
C_{ij}	50000 N/rad	A_L	2.25 m^2