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Estimation of Vehicle Longitudinal Tire Force with *FOSMO* & *SOSMO*

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Abstract. *In this paper, we propose two SM observer to estimate the longitudinal tire forces (F_x) using only wheel angular positions measurement. The robust estimations are based on First and Second Order Sliding Mode Observer (FOSMO, SOSMO). The estimation performance is tested and validated using a car driving simulator SCANeR^{TM}-Studio (www.oktal.fr). The simulation results show effectiveness and robustness of the proposed method.*

Keywords. Vehicle dynamic, Sliding Mode Observer, Inputs Estimation, Tire Forces Estimation.

1 Introduction

The estimation of tire road force has become an intensive research area as the interest in information technology in vehicles grows. More and more new safety technologies and approaches are introduced in the automotive environment. Therefore, the problem of traction control for ground vehicles is of enormous importance to automotive industry.

In the literature, many studies deal with observers for vehicle dynamics. The objective may be comfort analysis, design or increase in safety and by means of this controllability of the car on the road [1][2][3]. Various tire models for automotive vehicle are proposed for estimation of involved variables [4][5][6][7]. In [8], Bakker describe two analytical models that are intensively used by researchers in the field. In [9], Eiichi Ono estimate friction force characteristics using XBS (extended braking stiffness) method by applying online least-squares method. This method is applied also in [10][11]. Kiencke presents in [12] a procedure for real-time estimation of adherence μ ($\mu = \frac{Friction force}{Normal force}$). He develops a relation between μ and the wheel slip s. In [13] and [14], Ray estimates the forces on the tires with an extended Kalman filter. Villagra presents in [15] an estimation

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of the road maximum adherence μ based on algebraic approach using ALIEN algorithms. In [16], Gustafsson derives a scheme to identify different classes of roads. He assumes that combining the slip and the initial slope of the road adhesion versus curve it is possible to distinguish between different road surfaces. Recently, many studies have been performed on estimation of the frictions and contact forces between tires and road [17][18][19][20][21]. However, these inputs are difficult to get and not easy to measure. There are some difficulties to have them in real time. Then, it is necessary to estimate contact forces by robust observers with fast convergence.

In this paper, we use SCANeR^{TM}-Studio, a professional driving simulator proposed by Oktal (see www.oktal.fr), to validate the developed control systems using in the field of autonomous vehicles. This driving simulator can be used for prototyping and for design of vehicle functionalities. It will help us to evaluate performance of our observers.

This paper is organized as follows. Section 2 presents, after this introduction, the problem formulation and the models we have used to design our observers. The design of the First Order Sliding Mode Observer is then considered in section 3. This provide robust and fast estimations which are good indications for vehicle controllability despite the chattering effect which may appear. The Second Order sliding Mode Observer and its convergence analysis are presented in Section 4 to tackle the chattering problems. The description of the car driving simulator environment SCANeR^{TM}-Studio and some results about the states observation and longitudinal tire force estimation by means of proposed observer are presented in Section 5. Finally, some remarks and perspectives are given in a concluding section.

2 Vehicle Road Interaction and Problem Formulation

2.1 bakground

In this part, we use of a simple nominal partial model for description of wheels dynamics (see Figure (1)) [11][20][21]. This model is then parametrized for state observation and force estimation. The dynamic equations of the vehicle dynamics have been studied in [20][21][23] assuming the car body rigid and pneumatic contact permanent (see Figure(1)).

The generalized coordinate vector $q \in R^{16}$ is defined as follows for he Nominal Global Dynamical Model (see [21] for details):

$$\tau = M(q)\ \ddot{q} + C(q,\dot{q})\ \dot{q} + V(q,\dot{q}) + \eta_o(t,\ q,\ \dot{q}) \quad (1)$$

$$\tau = T_e + \Gamma = T_e + J^T F \quad (2)$$

The vectors $\dot{q}, \ddot{q} \in R^{16}$ are velocities and corresponding accelerations respectively. The $M(q)$ and $C(q,\dot{q})$ are the inertia and the Coriolis and Centrifugal matrices respectively. The vector $V(q,\dot{q})$ summarize the suspensions and gravitation forces with K_v and K_p are respectively the damping and the stiffness matrices, $G(q)$ is the gravity term. $\eta_o(t,q,\dot{q})$ represents the uncertainties and the neglected dynamics.

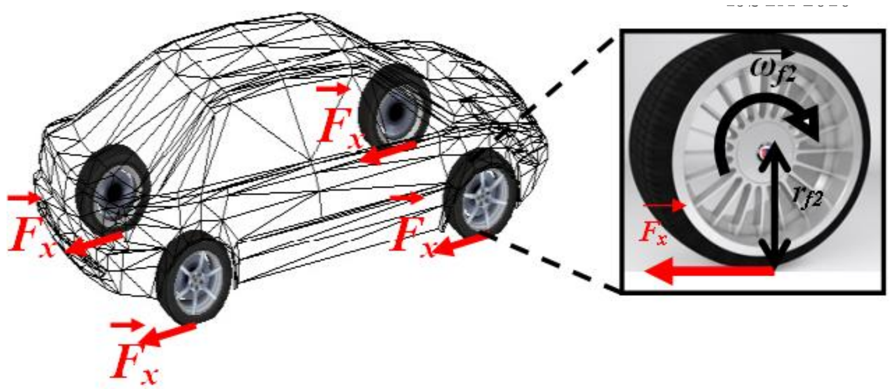

Fig. 1. Longitudinal Tire Forces F_x of Vehicle

2.2 Wheels Contact models

From equation (1) the wheels dynamic can be extracted in a partial model. The corresponding coordinate vector is x_1, velocities x_2 and longitudinal forces x_3 are:

$$x_1 = (x_{11}\ x_{12}\ x_{13}\ x_{14})^T = (\varphi_{f1}\ \varphi_{f2}\ \varphi_{r1}\ \varphi_{r2})^T \tag{3}$$

$$x_2 = (x_{21}\ x_{22}\ x_{23}\ x_{24})^T = (\omega_{f1}\ \omega_{f2}\ \omega_{r1}\ \omega_{r2})^T \tag{4}$$

$$x_3 = (x_{31}\ x_{32}\ x_{33}\ x_{34})^T = (F_{xf1}\ F_{xf2}\ F_{xr1}\ F_{xr2})^T \tag{5}$$

- x_1 are the angular positions of the four wheel.
- x_2 are the angular velocities of the four wheel.
- x_3 are the longitudinal forces applied to wheels
- $y = (\varphi_{f1}\ \varphi_{f2}\ \varphi_{r1}\ \varphi_{r2})^T$ ($y \in \mathbb{R}^4$) is the vector of measured system outputs.

The wheel angular motion is given by:

$$\begin{cases} \dot{\omega}_{f1} = \frac{1}{J_{rf1}} (C_m - T_{f1} - r_{f1}F_{xf1}) - \mu_{f1} \\ \dot{\omega}_{f2} = \frac{1}{J_{rf2}} (C_m - T_{f2} - r_{f2}F_{xf2}) - \mu_{f2} \\ \dot{\omega}_{r1} = \frac{1}{J_{rr1}} (-T_{r1} - r_{r1}F_{xr1}) - \mu_{r1} \\ \dot{\omega}_{r2} = \frac{1}{J_{rr2}} (-T_{r2} - r_{r2}F_{xr2}) - \mu_{r2} \end{cases} \tag{6}$$

Where ω_{f1}, ω_{f2}, ω_{r1} and ω_{r2} are wheel rotational velocities for, respectively, the front left, front right, rear left and rear right wheels. J_{rf1}, J_{rf2}, J_{rr1} and J_{rr2} are inertia of, respectively, the front left, front right, rear left and rear right tyre in vehicle. C_m is the motor traction torque. T_{ij}, r_{ij}, F_{ij} ($i = f, r$ and $j = 1, 2$) are respectively braking, wheel nominal radius and longitudinal tire force. μ_w is the coupling term due to dynamics of the other subsystems and the neglected dynamics $\eta_0(t, q, \dot{q})$ (see [21]).

2.3 State Space model with unknown inputs

Approximations are made when neglecting the coupling terms μ du to connections with the other sub systems. We can verify that these μ are bounded such and as $|\mu_i| < k_i\ \forall\ t$, $i = (1, ..., 4)$ [21][22][23].

Contact forces are the most difficult part to model in vehicle applications. Some assumptions are often used for existing models, like, for example, in the case of magic formula [24]:

- Assume wheel velocities constant, then slip angle and steering are constant,
- Assume that the pneumatic stiffness C_x, C_y, are invariant and constant
- Assume that the normal forces F_z exerted on the wheels are constant
- Assume that the behavior is uniform.

Actual situation are different from steady state conditions considered when modelling tire contact for steady states experiments. This means that at least the road characteristics have some perturbations and variables have small random variation, perturbations or additional noise $e(t)$. In this work, without any knowledge on force characteristics, we consider the case of driving in an homogenous straight road like in a highway. It can be justified if we get acceptable results and performance. We assume to have slow variations in average.

Let us consider that we have very small variations in the mean: $\dot{F} \simeq 0$. So we can use $\dot{x}_3 = e(t)$ where $e(t)$ is a centred random noise with a small mean average.

The system (6) can be written in state space form as follows:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = f(t) + g(t, x_3) - \mu \\ \dot{x}_3 = e(t) y = x_1 \end{cases} \tag{7}$$

with

$$f(t) = \begin{bmatrix} \dfrac{C_m - T_{f1}}{J_{rf1}} & \dfrac{C_m - T_{f2}}{J_{rf2}} & -\dfrac{T_{r1}}{J_{rr1}} & -\dfrac{T_{r2}}{J_{rr2}} \end{bmatrix}^T \tag{8}$$

$$g(t, x_3) = \begin{bmatrix} -r_{f1} & 0 & 0 & 0 \\ 0 & -r_{f2} & 0 & 0 \\ 0 & 0 & -r_{r1} & 0 \\ 0 & 0 & 0 & -r_{r2} \end{bmatrix} \begin{bmatrix} x_{31} \\ x_{32} \\ x_{33} \\ x_{34} \end{bmatrix} = U.x_3$$

$$\mu = \begin{bmatrix} \mu_{f1} & \mu_{f2} & \mu_{r1} & \mu_{r2} \end{bmatrix}^T \tag{9}$$

3 First Order Sliding Mode Observer (*FOSMO*)

3.1 SM Observer

In this section, we develop a robust First Order Sliding Mode Observer. This technique is an attractive approach for robustness [23][25][26][27].
The proposed *FOSMO* has the form

$$\begin{cases} \dot{\hat{x}}_1 = \hat{x}_2 - \Lambda_1 sign(\hat{x}_1 - x_1) \\ \dot{\hat{x}}_2 = \hat{f}(t) + \hat{g}(t, \hat{x}_3) - \Lambda_2 sign(\hat{x}_1 - x_1) \\ \dot{\hat{x}}_3 = -\Lambda_3 sign(\hat{x}_1 - x_1) \end{cases} \tag{10}$$

where $\hat{x}_1$, $\hat{x}_2$ and $\hat{x}_3$ are respectively the state estimations of positions, velocities and input forces, Λ_1, Λ_2 and Λ_3 are the observer gains matrices ($\Lambda_i \in \mathbb{R}^4; i = 1,2,3$), $\hat{f}(t)$ and $\hat{g}(t,\hat{x}_3) = \hat{U}\hat{x}_3$ are respectively the estimation of $f(t)$ and $g(t,x_3)$.

$$\begin{array}{l} \Lambda_1 = diag\{\lambda_{11},\lambda_{12},\lambda_{13},\lambda_{14}\} \\ \Lambda_2 = diag\{\lambda_{21},\lambda_{22},\lambda_{23},\lambda_{24}\} \\ \Lambda_3 = diag\{\lambda_{31},\lambda_{32},\lambda_{33},\lambda_{34}\} \end{array}$$

For the convergence analysis, we take $H(t,\hat{x}_3,x_3) = \hat{f}(t) - f(t) + g(t,\hat{x}_3) - g(t,x_3) + \mu$, $\tilde{x}_1 = \hat{x}_1 - x_1$ and $\tilde{x}_2 = \hat{x}_2 - x_2$. The observation error dynamic is then:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - \Lambda_1 sign(\tilde{x}_1) \\ \dot{\tilde{x}}_2 = H(t,\hat{x}_3,x_3) - \Lambda_2 sign(\tilde{x}_1) \\ \dot{\tilde{x}}_3 = -\Lambda_3 sign(\tilde{x}_1) \end{cases} \tag{11}$$

3.2 Convergence analysis

The Lyapunov candidate function $V_1 = \dfrac{1}{2}\tilde{x}_1^T\tilde{x}_1$, in a first step, shows that $\tilde{x}_1 = 0$ is an attractive surface if we ensure that $\dot{V}_1 < 0$ by means of choices of λ_{1i} $(i = 1,..,4)$

$$\dot{V}_1 = \tilde{x}_1^T(\tilde{x}_2 - \Lambda_1 sign(\tilde{x}_1)) \tag{12}$$

$$\lambda_{1i} >\mid \tilde{x}_{2i} \mid \quad for\ i = 1,..,4 \tag{13}$$

The convergence, in finite time (t_0) for the first system state is obtained: $\hat{x}_1$ goes to x_1, so $\dot{\tilde{x}}_1 = 0\ \forall t > t_0$. We can then deduce, in average, from the first equation of system (11) that $\tilde{x}_2 = \Lambda_1 sign_{eq}(\tilde{x}_1)$, with $sign_{eq}$ the average value of function "*sign*" in the sliding surface. Then we obtain the following average for the over equations:

$$\begin{array}{l} \dot{\tilde{x}}_2 = H(t,\hat{x}_3,x_3) - \Lambda_2\Lambda_1^{-1}\tilde{x}_2 \\ \dot{\tilde{x}}_3 = -\Lambda_3\Lambda_1^{-1}\tilde{x}_2 \end{array} \tag{14}$$

with

$$H(t,\hat{x}_3,x_3) = \tilde{f}(t) + \hat{U}\hat{x}_3 - Ux_3 \pm U\hat{x}_3 + \mu \tag{15}$$

$$= \tilde{f}(t) + \tilde{U}\hat{x}_3 - U\tilde{x}_3 + \mu \tag{16}$$

In the second step, we take as Lyapunov candidate function $V_2 = \frac{1}{2}\tilde{x}_2^T\tilde{x}_2 + \frac{1}{2}\tilde{x}_3^T\tilde{x}_3$, the derivative of this function is:

$$\dot{V}_2 = \tilde{x}_2^T(H - \Lambda_2\Lambda_1^{-1}\tilde{x}_2) - \tilde{x}_3^T\Lambda_3\Lambda_1^{-1}\tilde{x}_2 \tag{17}$$

if $\Lambda_3 = U^T \Lambda_1$, the $\dot{V}_2$ becomes:

$$\dot{V}_2 = \tilde{x}_2^T(\tilde{f}(t) + \tilde{U}\hat{x}_3 + \mu) - \tilde{x}_2^T \Lambda_2 \Lambda_1^{-1} \tilde{x}_2 \tag{18}$$

In the following, $N(t, \hat{x}_3) = \tilde{f}(t) + \tilde{U}\tilde{x}_3 + \mu$ and $\tilde{x}_2$ will be supposed to be bounded for all t as follows:

$$|N(t, \hat{x}_3)| \leq N_{max} \tag{19}$$

$$|\hat{x}_2^T| \leq X_{max} \tag{20}$$

if $\Lambda_2 > N_{max} X_{max}^T \Lambda_1$, then $\dot{V}_2 < 0$ and consequently $\hat{x}_2$ and $\hat{x}_3$ goes respectively to x_2 and x_3.

Under some conditions, the system state estimation error converges, in a finite time, to vicinity of zero, the velocity errors go asymptotically to zero and the force estimation errors are bounded and decreasing. In Conclusion, the conditions to get the convergence of *FOSMO* can be summarized as:

* $\Lambda_1 >| \tilde{x}_2 |$
* $\Lambda_2 > N_{max} X_{max}^T \Lambda_1$
* $\Lambda_3 = U^T \Lambda_1$

4 Second Order Sliding Mode Observer (*SOSMO*)

4.1 Robust Differential Estimator

In order to estimate the state vector and to deduce the unknown input forces vector ($x_3 = F_x$), the following Second Order Sliding Mode Observer *SOSMO* (using the super-twisting algorithm) is proposed: [28][26][29][30][31][22]

$$\dot{\hat{x}}_1 = \hat{x}_2 + z_1 \tag{21}$$

$$\dot{\hat{x}}_2 = \hat{f}(t) + z_2 \tag{22}$$

where $\hat{x}_1$, $\hat{x}_2$ and $\hat{x}_3$ are the state estimations, α and β ($\in \mathbb{R}^4$) are the observer gains and the correction variables z_1 and z_2 are calculated by:

$$\begin{aligned} z_1 &= -\alpha \mid \hat{x}_1 - x_1 \mid^{\frac{1}{2}} sign(\hat{x}_1 - x_1) \\ z_2 &= -\beta sign(\hat{x}_1 - x_1) \end{aligned}$$

4.2 Convergence analysis

At the initial moment we take $\hat{x}_{1(t=0)} = x_{1(t=0)}$ and $\hat{x}_{2(t=0)} = 0$. It is important to note that in a first step, input effects on the dynamic are rejected by the proposed observer like a perturbation. The dynamics estimation errors for SOSMO, with $\tilde{x}_1 = \hat{x}_1 - x_1$ and $\tilde{x}_2 = \hat{x}_2 - x_2$, are calculated as follows:

$$\begin{cases} \dot{\tilde{x}}_1 = \tilde{x}_2 - \alpha\sqrt{\mid \tilde{x}_1 \mid} sign(\tilde{x}_1) \\ \dot{\tilde{x}}_2 = \Phi(t, \hat{x}_3, x_3) - \beta sign(\tilde{x}_1) \end{cases} \tag{23}$$

Where $\Phi(t, \hat{x}_3, x_3) = \hat{f}(t) - f(t) - g(t, x_3) + \mu$. In our case, we suppose that the system states can be assumed bounded, then the existence of a constant bound g^+ is ensured, such that the inequality

$$| \Phi(t, \hat{x}_3, x_3) | < \Phi^+ \tag{24}$$

In order to define the bound Φ^+ let us consider the system physical properties. Consider x_{3max} and f_{max} are defined such that $\forall t \in \mathbb{R}^+$, $\forall \hat{x}_3, | \hat{x}_3 | \leq 2x_{3max}$ and $\forall \tilde{f}(t), | \tilde{f}(t) | \leq f_{max}$. The state boundedness is true, because mechanical systems are passive and BIBS stable, and the control action input x_3 is bounded. The maximal possible acceleration in the system is a priori known and it coincides with the bound Φ^+. We have:

$$\underline{U} I \leq U \leq \overline{U} I \tag{25}$$

where $\underline{U}$ is the minimal eigenvalue and $\overline{U}$ the maximal one. The Φ^+ can be written as

$$\Phi^+ = f_{max} + \overline{U} x_{3max} \tag{26}$$

Let α and β satisfy the following inequalities, where p is some chosen constant, $0 < p < 1$ (see [32])

$$\beta > \Phi^+ \tag{27}$$

$$\alpha > \sqrt{\frac{2}{\beta - f_{max}} \frac{(\beta + f_{max})(1+p)}{(1-p)}} \tag{28}$$

The presented observer ensures that in finite time, we have $\tilde{x}_2 \simeq 0$, then $\dot{\tilde{x}}_2 \simeq 0$ holds after some finite time

$$\dot{\tilde{x}}_2 = \tilde{f}(t) - g(t, x_3) + \mu + z_2 = 0 \tag{29}$$

It was assumed that the term z_2 changes at a high (infinite) frequency. However, in reality, various imperfections make the state oscillate in some vicinity of the intersection and components of z_2 are switched at finite frequency, this oscillations have high and slow frequency components. The high frequency term z_2 is filtered out and the motion in the sliding mode is determined by the slow components. The equivalent control is close to the slow component and may be derived by low pass filtering. The filter time constant should be sufficiently small to preserve the slow components undistorted but large enough to eliminate the high frequency component [25][27]. Let us take a low-pass filtering of z_2, then we obtain in the mean average:

$$\overline{z}_2 = g(t, x_3) - \xi(t) \tag{30}$$

where $\xi(t) = \tilde{f}(t) + \mu$ and z_2 is filtered signal of z_2. Then we obtain

$$x_3 = U^{-1}(\overline{z}_2 + \xi(t)) \tag{31}$$

As $\xi(t)$ is unknown function and is assumed small, then we can written an estimation of longitudinal tire force $\hat{x}_3$ as following

$$\hat{x}_3 = U^{-1}\overline{z}_2 \tag{32}$$

In Conclusion, the conditions to get the convergence of *SOSMO* can be summarized as:

* $\beta > \Phi^+$
* $\alpha > \sqrt{\dfrac{2}{\beta - f_{max}}\dfrac{(\beta + f_{max})(1+p)}{(1-p)}}$
* $\hat{x}_3 = U^{-1}\overline{z}_2$ is an approximate estimation of contact longitudinal forces.

5 Experimental Results

5.1 Description of SCANeR^{TM}studio

SCANeR^{TM} software is compliant with any kind of driving system simulator from desktop system to full scale motion based system. OKTAL[1] is able to provide you with different types of products:

- To end users requiring a turn key solution, OKTAL provides relevant driving or testing simulator.
- To systems integrators or end users requiring only a software solution, OKTAL provides them with SCANeR^{TM} software and associated services

OKTAL has packaged two different softwares:

* **SCANeR^{TM}studio** dedicated to engineering & research.
* **SCANeR^{TM}DT** dedicated to training and safety awareness.

SCANeR^{TM}studio software is a modular configuration, which means that you can use and add a module or bundle depending on your needs. The main characteristics of SCANeR^{TM}studio software are:

- Distributed architecture,
- Ethernet based communication back-bone,
- Multi-platform (Windows^{TM} XP, Vista, Windows 7),
- Centralised user-friendly supervising application.

The SCANeR^{TM}studio features are regrouped according to different components to run experimentations: ► Core, ► Driver, ► Environment, ► Vehicle, ► Simulation. In this section, we give some results in order to test and validate our approach an the proposed observers. We use the simulator of vehicle SCANeR^{TM}-Studio which is in laboratory LSIS. The Figures (3-a) and (3-b) present respectively the various constituents of simulator and the view module of simulator.

[1] www.oktal.fr

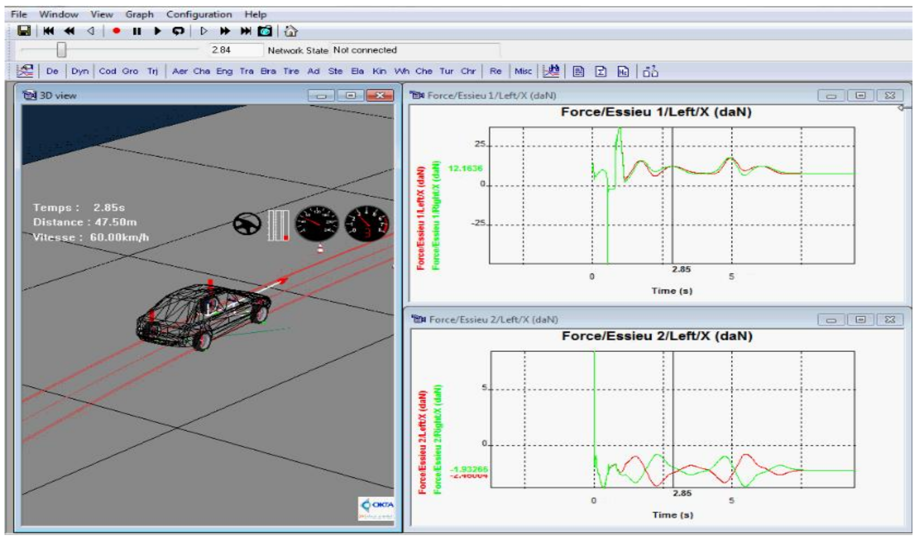

Fig. 2. Analysing Tool example

5.2 Results of *FOSMO*

In this simulation, the vehicle crosses the chicane trajectory (Figure (4)) with two constant speed at $60km/h = 16.66m/s$ and at $80km/h = 22.22m/s$ (Figure (5-c)). The tire characteristic used here is 185/70/R14. The road is considered plane, no slope and no banking road. The forces are generated by use of the Magic Formula tire model. The inputs of vehicle model are the steering angle for the two front wheels, presented by the Figure (5-a), and the angular position of the four wheels. The Figure (5-b) present the longitudinal acceleration used in simulation.

	FOSMO	
	60 km/h	**80 km/h**
λ_{11}	1.2	1.2
λ_{12}	1.2	1.2
λ_{13}	1.2	1.2
λ_{14}	1.2	1.2
λ_{21}	75	150
λ_{22}	75	150
λ_{23}	75	150
λ_{24}	75	150

Tableau 1. *FOSMO* parameters

	SOSMO	
	60 km/h	**80 km/h**
α_{11}	1.5	10
α_{12}	1.5	10
α_{13}	1.5	10
α_{14}	1.5	10
β_{21}	150	150
β_{22}	150	150
β_{23}	150	150
β_{24}	150	150

Tableau 2. *SOSMO* parameters

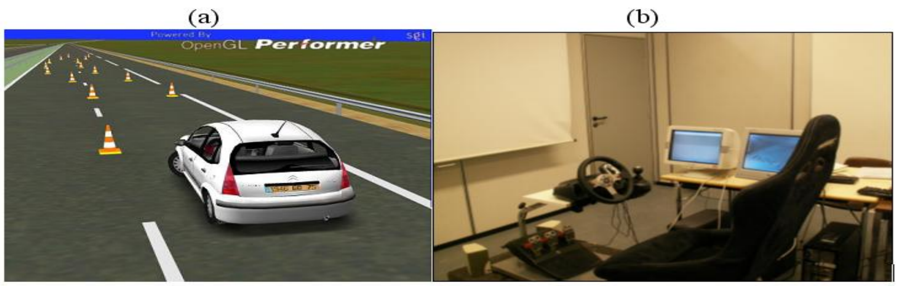

Fig. 3. Car simulator SCANeR^{TM}studio

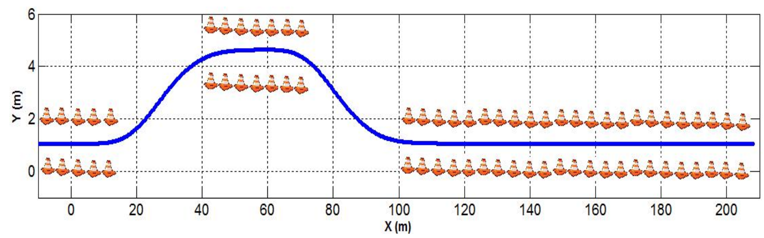

Fig. 4. Trajectory

The *FOSMO* Parameters are presented by the table (1). The Figures (6-a), (6-b), (6-c) and (6-d) presents the real and the estimation angular velocity of the four wheel: Front Left, Front Right, Rear Left and Rear Right wheel with a longitudinal velocity at $60km/h$. We show the asymptotic convergence of the angular velocity. With the same longitudinal velocity $V_x = 60km/h$, the figures (8-a), (8-b), (8-c) and (8-d) show the convergence of the estimated longitudinal force to their actual value in finite time for respectively the front left, front right, rear left and rear right tire of vehicle using the observer *FOSMO*.

The performance of this estimation approach is satisfactory since the estimation error is minimal for angular velocities and longitudinal tire force at $60km/h$. Nevertheless, we remark the appearance of a chattering for $t \in [0, 2s]$ that comes from to the sign functions. For $V_x = 80km/h$, we present respectively in Figure (7) and Figure (9), the real and observed angular velocities for the four wheels in vehicle and the longitudinal tire force F_x. We note also a good convergence despite the chattering phenomena.

5.3 Results of *SOSMO*

in this part, we present the experimental results obtained for the observer *SOSMO*. The table (2) present the parameters used in our simulation. We show, respectively, in Figure (10) and Figure (11) a good convergence of the angular velocities

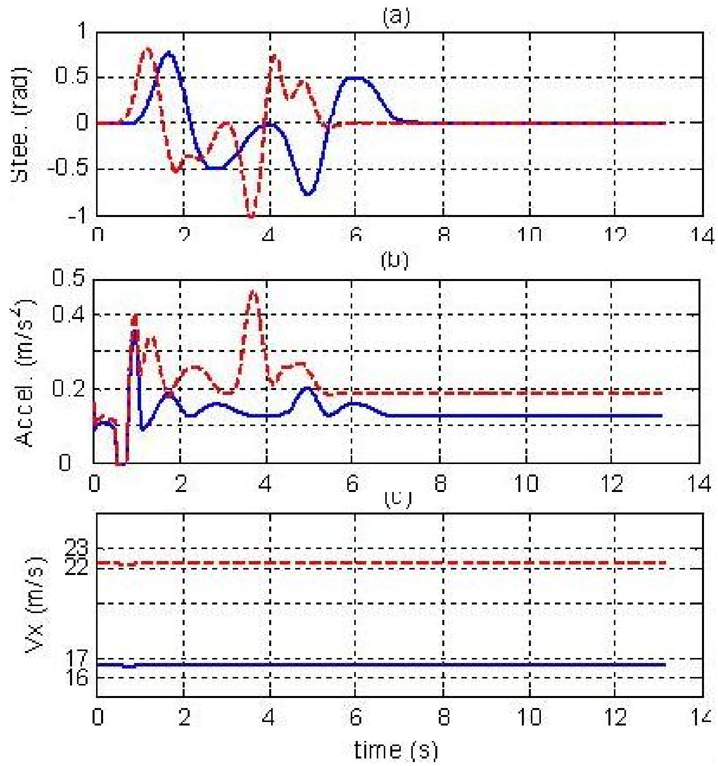

Fig. 5. Vehicle dynamic states ($--$: $80km/h$, $-$: $60km/h$)

for the four wheels of vehicle at longitudinal velocity $60km/h$ and $80km/h$. The Figures (12) and (13) presents respectively the estimation of the longitudinal tire force Fx at $60km/h$ and $80km/h$. We remark a good convergence in finite time and we see that the chattering phenomena is reduced.

6 Conclusions

In this paper, we present a First and Second Order Sliding Mode Observer to estimate the longitudinal tire force in vehicle. This approach allows us to avoid the observability problems dealing with inappropriate use of modelling equations. For vehicle systems, it is very hard to build up a complete and appropriate model for observation of all the system states. The finite time convergence of the two observer, *FOSMO* and *SOSMO*, is proved.
The gains of the proposed observer are chosen very easily ignoring the system parameters. Validation of performance has been proven using a driving simulator SCANeR^{TM}Studio with two longitudinal velocities of vehicle V_x at $60km/h$ and at $80km/h$.

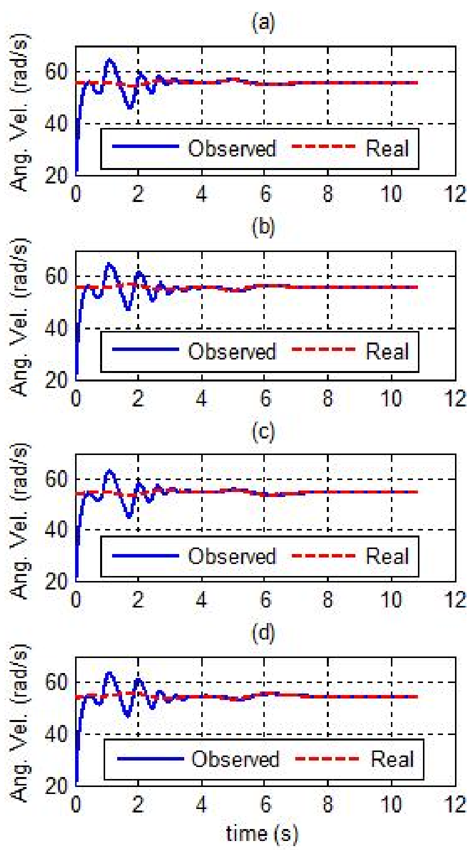

Fig. 6. Real & Observed Angular velocities of wheels with $FOSMO$ at $60km/h$

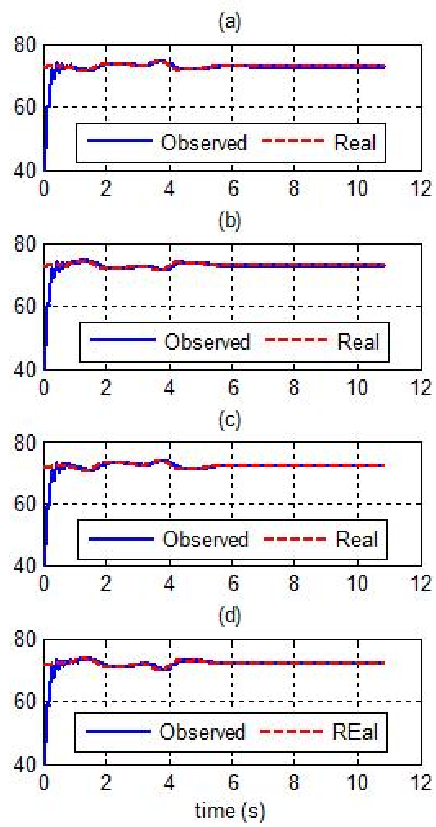

Fig. 7. Real & Observed Angular velocities of wheels with $FOSMO$ at $80km/h$

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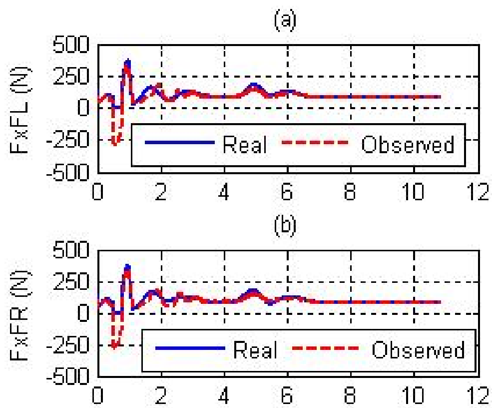

Fig. 8. Real & Observed F_x with $FOSMO$ at $60km/h$

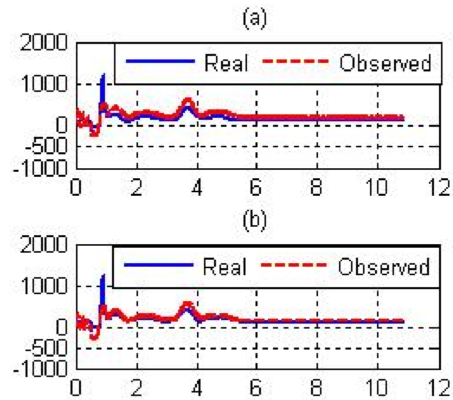

Fig. 9. Real & Observed F_x with $FOSMO$ at $80km/h$

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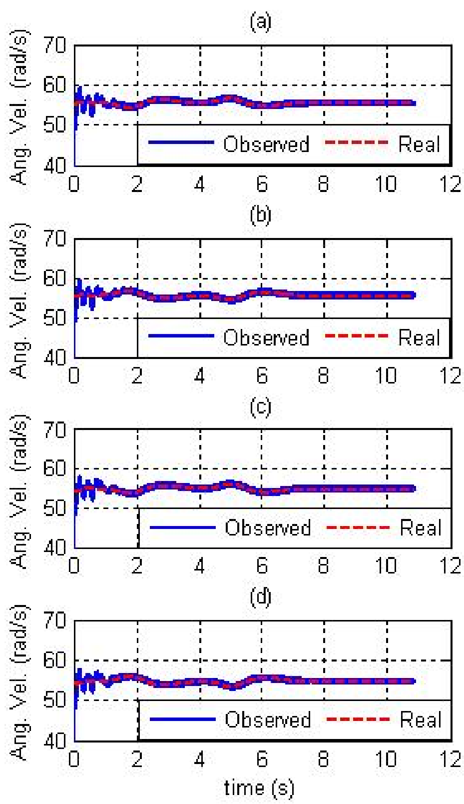

Fig. 10. Real & Observed Angular velocities of wheels with $SOSMO$ at $60km/h$

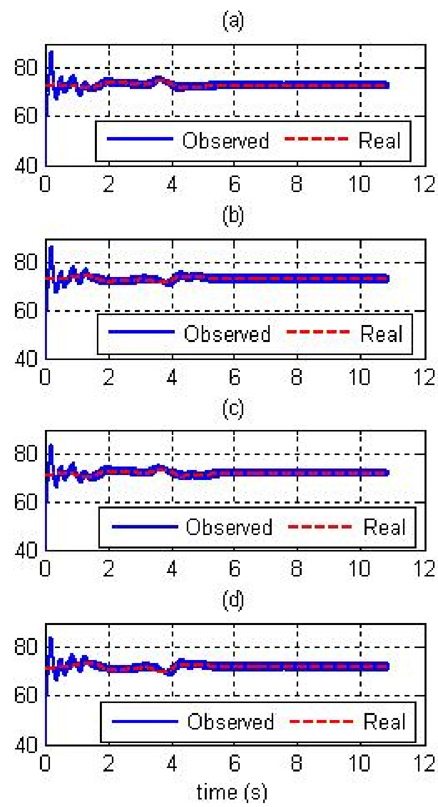

Fig. 11. Real & Observed Angular velocities of wheels with $SOSMO$ at $80km/h$

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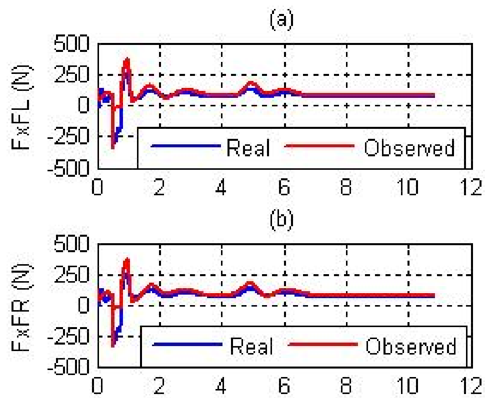

Fig. 12. Real & Observed F_x with $SOSMO$ at $60km/h$

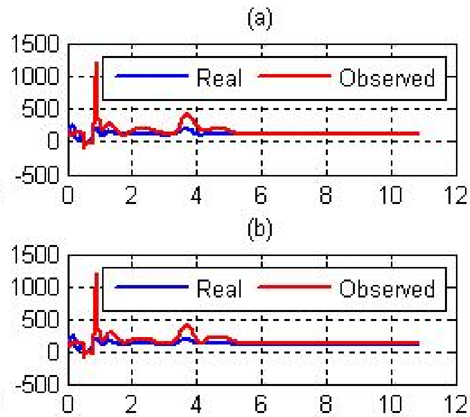

Fig. 13. Real & Observed F_x with $SOSMO$ at $80km/h$

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