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Observers With Unknown Inputs to Estimate Contact Forces and Road Profile

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***Abstract*— This paper presents sliding mode observers designed to estimate tire forces and road profile. Tire forces affect the vehicle dynamic performance and behavior properties. The tire forces and road friction are difficult to measure and their modelling is rather complex. In this work we deal with a simple model of vehicle combined with sliding mode approach to develop robust observers.**

***Index Terms*— Nonlinear observers, Sliding Modes, Vehicle-Road Interaction Models, State Estimation, Tire Forces, Road Profile.**

I. Introduction

Knowledge of tire forces is essential for systems such as antilock braking systems (ABS), traction control systems (TCS) and electronic stability program (ESP). Vehicle dynamics depend largely on the tire forces which are nonlinear functions of wheel slip and slip angles and depend on some factors such as tire wear, pressure, normal load tire road interface properties [1][2][3]. Recently, many analytical and experimental studies have been performed on estimation of the frictions and contact forces between tires and road [4][5][6]. The tire forces affect the vehicle dynamic performance and behavior properties. Thus for vehicles and road safety analysis, it is necessary to take into account the contact force characteristics. However, tire forces and road friction are difficult to measure directly and complex to be precisely represented by some deterministic equations. Vehicle dynamics depends largely on the tire forces represented by the nonlinear functions of wheel slip. The tire models encountered are complex and depend on several factors (as load, tire pressure, environmental characteristics, etc.). This makes on line estimation of forces and parameters difficult for vehicle control applications like detection and diagnosis for driving monitoring and surveillance. In this paper, modelling of the contact forces and interactions between a vehicle and road is revisited in the objective of on line force estimation using robust observers coupled with a robust and adaptive estimation of contact forces. We propose a robust observer to estimate the vehicle state and an adaptive estimator for tire forces identification[7]. The designed observer is based on the sliding mode approach. The main contribution is on-line estimation of inputs (the tire forces and road profile) needed for control. In this work, we deal with a simple vehicle model coupled with an appropriate wheel-road contact model in order to estimate contact forces. Then, we develop a method to observe tire forces and road profile.

This paper is organized as follows. Section 2 deals with the vehicle description and modelling for estimation of contact forces. The design of an observer and an adaptive tire force estimation is presented in section 3. Section 4 is devoted to develop an observer with unknown inputs to estimate the road profile. Some results about the states observations and estimation of the two kinds of unknown inputs are presented in section 5. Finally, some remarks and perspectives are given in a concluding section.

II. VEHICLE MODELLING

In the literature, many studies deal with vehicle modelling [8][9][10]. The objective may be either analysis for better design and features enhancement or increase of safety and maniability. The vehicle is a complex mechanical system that exhibits nonlinear behaviors. Commonly, the proposed and used models are not very complicated and give partial representation of the system dynamics. It would be relatively difficult and intricacy to involve more complete models and to define the size of different parameters. Several models have been considered in literature for analysis of the road - vehicle interaction and its consequence on the behavior (see eg Figures 1 and 2). The motions (longitudinal, lateral, and vertical) depend on interaction between the wheels and the road, the disruptions and the gravity.

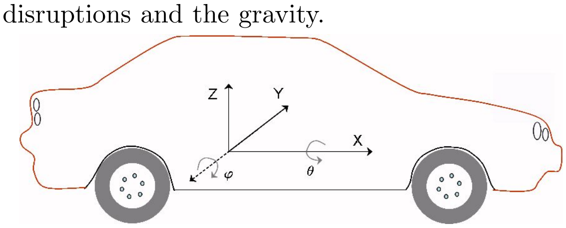

Fig. 1. Half Vehicle Longitudinal and Vertical Model (x, z, φ)

We generally we can distinguish three types of vehicle models:

- Longitudinal (Figures 1)
- Lateral
- longitudinal + lateral (Figure 2).

Let us consider, for our case, the last type of representation (Figure 2). We can define as dynamic equations of the vehicle:

$$m \begin{bmatrix} \dot{v}_x \\ \dot{v}_y \\ \dot{v}_z \end{bmatrix} = \begin{bmatrix} \sum F_x \\ \sum F_y \\ \sum F_z \end{bmatrix} \quad (1)$$

$$J \begin{bmatrix} \ddot{\theta} \\ \ddot{\phi} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} \sum M_x \\ \sum M_y \\ \sum M_z \end{bmatrix} \quad (2)$$

The wheel angular motion is described by:

$$\begin{aligned} J_{fi}\dot{\omega}_{fi} &= T_{fi} - r_{fi}F_{xfi} \\ J_{ri}\dot{\omega}_{ri} &= T_{ri} - r_{ri}F_{xri} \end{aligned} \quad (3)$$

where $v = [v_x, v_y, v_z]^T$: vehicle velocities along x, y, z,

The subscripts f, r stand for front and rear wheels respectively.

ω_{fi} and ω_{ri} : rotation speeds of wheels (front and rear),

r_{fi} and r_{fi} :wheel radius at front and rear respectively,

J_{fi} and J_{ri} : front and rear wheels inertia respectively,

F_{xf} , F_{yf}: longitudinal, lateral forces on front wheels,

F_{xr} , F_{yr} :longitudinal, lateral force on the rear wheels,

T_{fi} and T_{ri} : torques applied to front and rear wheels,

$\sum F_x$, $\sum F_y$, $\sum F_z$, M_x, M_y and M_z : forces and moments with respect to x, y and z. and θ ϕ ψ.

We propose therefore, with these dynamic equations, to use a longitudinal-lateral (tire-road) contact model, in order to take into account the slip effects and ground forces. This will provide the latter variables which inputs of the depicted equations.

A. *Simplified model*

Known as the bicycle model ([10][11]), the structure described by Figure 2, gives a fairly good representation of the vehicle behavior in the (x,y).plane. This representation is obtained by replacing the two front wheels by an equivalent one. A similar approach is applied for the two rear wheels.

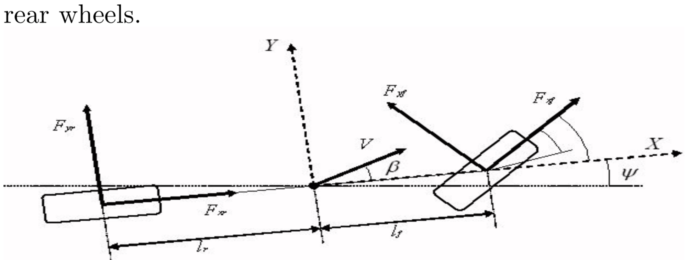

Fig. 2. Half Vehicle Longitudinal and Lateral Model (x, y, ψ)

This representation considers the following assumptions :

1) The epicenter is assumed to be on road level
2) Neglect the roll, pitch, and vertical motion;
3) The road is assumed to be perfectly flat
4) Neglect influence of aerodynamic side forces.

The dynamic model represents the longitudinal, lateral, yaw motions and the rotation of the wheels as shown in figure 2. The resulting equations of the simplified vehicle model are then

$$\begin{aligned} m\dot{V}_x &= F_{xf}(\cos(\delta_F) - F_{yf}\sin(\delta_F) + F_{xr} \\ m\dot{V}_y &= F_{xf}\sin(\delta_F) + F_{yf}\cos(\delta_F) + F_{yr} \\ J_z\ddot{\psi} &= F_{xf}l_f\sin(\delta_F) + F_{yf}l_f\cos(\delta_F) - l_rF_{yr} \end{aligned} \quad (4)$$

δ_F is the front wheel steering angle ($\delta_r = 0$); l_r and l_f distances between the center of gravity of the vehicle and the rear and front axis respectively.

B. *Tire modelling*

The forces are produced by contact between the road and tires. They are transmitted trough the dynamics of wheels and vehicle. They are of major importance for the dynamic behavior of a road vehicle. Hence, accurate tire models are necessary components of models aimed at analyzing or simulating vehicle motion in real driving conditions. The driver can then control the vehicle trough this dynamic. A lot of work has been done in the area of tires model fitting and estimation [12][13]. Many models have been previously used to describe the tire-forces. Some of them are theoretical in the sense that they aim at modelling the physical processes that generates the forces. Other ones are empirically oriented and their aim is to describe observed phenomena in a simple way.

1) Lateral dynamics and transient phenomenon: The dynamic behavior of the transverse motion was the subject of several works [14]. In [[18], Pacejka describes by a first order model, the variations of the lateral strength and the moment of auto-alignment in presence of weak values of the slip angle, while using the notion of the relaxation length.

To illustrate the concept of the relaxation length, let us consider the dynamic variations of strength lateral F_y in the case of weak rates of slip. Suppose that the variation of the vertical strength is weak as $\dot{F}_z = 0$, then the variation F_y is associated to a variation of the lateral speed of point of contact represented mainly by V_{cy}. So to describe the transient, the variation of F_y is represented by a differential first order equation:

$$\sigma_{yi}\dot{F}_{yi} + V_{xi}F_{yi} = C_yV_y \qquad i = f, r \quad (5)$$

where: C_y is the rigidity of the lateral slip, and σ_y represents the length of relaxation. We can extend this equation (5) to large slips to get:

$$\sigma_{yi}\dot{F}_{yi} = -V_x(F_{yi} - F_{yi0}) + C_yV_y \quad i = f, r \quad (6)$$

the unknown parameters F_{yi0} is intersection of the tangent $\frac{\partial F_{yi}}{\partial \lambda_{yi}}$ and the axis of F_{yi}

2) Longitudinal dynamics and transient phenomenon: By analogy, the notion of relaxation length is used to describe the longitudinal dynamics. In [Clover 98], the authors present the variations of the slip rate by a first order differential equation. They use this representation and a longitudinal linear model to study the stability of the automotive dynamics in different rolling situations. In longitudinal case the variation can be represented by a first order model:

$$\begin{array}{l} \text{a) during braking}, \quad i=f,r \\ \sigma_{xi}\dot{F}_{x_i} = -V_x(F_{xi}-F_{xi0}) + C_x(V_x - r_i\omega_i) \\ \text{b) during acceleration}, \quad i=f,r \\ \sigma_{xi}\dot{F}_{x_i} = -r_i\omega_i(F_{xi}-F_{xi0}) + C_x(V_x - r_i\omega_i) \end{array} \quad (7)$$

Finally, during acceleration, the model can be written:

$$\begin{aligned} \dot{F}_{xf} &= -\frac{V_x}{\sigma_{lf}}F_{xf} + \frac{V_x}{\sigma_{lf}}F_{xf0} + \frac{C_x}{\sigma_{lf}}V_x - \frac{C_x}{\sigma_{lf}}r_f\omega_f \\ \dot{F}_{xr} &= -\frac{V_x}{\sigma_{lr}}F_{xr} + \frac{V_x}{\sigma_{lr}}F_{xr0} + \frac{C_x}{\sigma_{lr}}V_x - \frac{C_x}{\sigma_{lr}}r_r\omega_r \\ \dot{F}_{yf} &= \frac{C_y}{\sigma_{tf}}V_y - \frac{V_x}{\sigma_{tf}}F_{yf} + \frac{V_x}{\sigma_{tf}}F_{yfo} \\ \dot{F}_{yr} &= \frac{C_y}{\sigma_{tr}}V_y - \frac{V_x}{\sigma_{tr}}F_{yr} + \frac{V_x}{\sigma_{tr}}F_{yro} \end{aligned} \quad (8)$$

III. Adaptive Estimation of Tire forces

A. Expression of the robust observer

The system equations can be rewritten in the following state space form:

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = \Omega(u)x_4 \\ \dot{x}_3 = Gu - [H \ ; \ 0]\, x_4 \\ \dot{x}_4 = \Psi(X)\theta \end{cases} \quad (9)$$

where $x_1 = (x_{11}, x_{12}, x_{13}) = (x, y, \psi)$
$x_2 = (x_{21}, x_{22}, x_{23}) = (V_x, V_y, \dot{\psi})$
$x_3 = (x_{31}, x_{32}) = (\omega_f, \omega_r)$
$x_4 = (x_{41}, x_{42}, x_{43}, x_{44}) = (F_{xf}, F_{xr}, F_{yf}, F_{yr})$
$x_{4x} = (x_{41}, x_{42})$, $x_{4y} = (x_{43}, x_{44})$

$$\Omega(u) = \begin{bmatrix} \frac{1}{m}\cos(\delta_F) & \frac{1}{m} & \frac{1}{m}\sin(\delta_F) & 0 \\ \frac{1}{m}\sin(\delta_F) & 0 & \frac{1}{m}\cos(\delta_F) & \frac{1}{m} \\ \frac{1}{J_z}l_f\sin(\delta_F) & 0 & \frac{1}{J_z}l_f\cos(\delta_F) & -\frac{1}{J_z}l_r \end{bmatrix}$$

$$G = \begin{bmatrix} 0 & 0 & 0 \\ 0 & \frac{1}{J_f} & 0 \\ 0 & 0 & \frac{1}{J_r} \end{bmatrix}; \ H = \begin{bmatrix} \frac{r_f}{J_f} & 0 & 0 & 0 \\ 0 & \frac{r_f}{J_f} & 0 & 0 \end{bmatrix}$$

$$\Psi(X) = \begin{bmatrix} \Psi_1(X) & 0 \\ 0 & \Psi_2(X) \end{bmatrix}$$

$$\begin{array}{l} \Psi_1(X) = \\ \\ \Psi_2(X) = \end{array} \begin{bmatrix} C_x x_{11} - x_{11}x_{31} - C_x r_f x_{21};\ 0; x_{11}; 0 \\ 0; C_x x_{11} - x_{11}x_{32} - C_x r_r x_{22}; 0; x_{11}; \\ C_y l_f x_{13} + C_y x_{12} - x_{33}x_{11}; 0; x1_{11}; 0 \\ 0; C_y x_{12} - C_y l_r x_{13} - x_{34}x_{11}; 0; x_{11} \end{bmatrix}$$

To estimate both state an unknown parameters we propose the following sliding mode based observer [15]:

$$\begin{cases} \dot{\hat{x}}_2 = \Omega(u)\widehat{x}_4 + \Lambda_2 sign(\widetilde{x}_2) \\ \dot{\hat{x}}_3 = Gu - H\widehat{x}_{4x} + \Lambda_3 sign(\widetilde{x}_3) \\ \dot{\hat{x}}_4 = \Psi(\hat{X})\widehat{\theta} + \Lambda_4 sign(\widetilde{x}_2) + \Lambda_5 sign(\widetilde{x}_3) \\ \dot{\hat{\theta}} = \eta \end{cases} \quad (10)$$

where $\widehat{x}_i$represents the observed state vector, $\widetilde{x}_2 = x_2 - \widehat{x}_2$, $\widetilde{x}_3 = x_3 - \widehat{x}_3$ are the state estimation errors. The observer gains Λ_i and the unknown η will be defined thereafter..We can write

$$\Delta\Psi_1 = \Psi_1(X) - \Psi_1(\hat{X}) = \begin{bmatrix} -x_{21}\tilde{x}_{41} & 0 & 0 & 0 \\ 0 & -x_{21}\tilde{x}_{42} & 0 & 0 \end{bmatrix}$$

$$\Delta\Psi_2 = \Psi_2(X) - \Psi_2(\hat{X}) = \begin{bmatrix} -\tilde{x}_{43}x_{21} & 0 & 0 & 0 \\ 0 & -\tilde{x}_{44}x_{21} & 0 & 0 \end{bmatrix}$$

we have

$$\left\|\Psi(X) - \Psi(\hat{X})\right\| \le a\left\|\widetilde{x}_{4x}\right\| + b\left\|\widetilde{x}_{4y}\right\|$$

We can write

$$\Psi(\hat{X})\widehat{\theta} - \Psi(X)\theta = -\Psi(\hat{X})\widetilde{\theta} + \Delta\Psi\theta$$

$$\varepsilon = \Delta\Psi\theta = (\Psi(X) - \Psi(\hat{X})\theta = \begin{bmatrix} \Delta\Psi_1.\Theta_1 \\ \Delta\Psi_2.\Theta_2 \end{bmatrix}$$

$$\varepsilon = -x_{21}\begin{bmatrix} \theta_1 & 0 & 0 & 0 \\ 0 & \theta_2 & 0 & 0 \\ 0 & 0 & \theta_5 & 0 \\ 0 & 0 & 0 & \theta_6 \end{bmatrix}.\tilde{x}_4 = -x_{21}.F.\tilde{x}_4$$

$$\|\varepsilon\| \le (c.\max(\theta))\left\|\widetilde{x}_4\right\|$$

$$\begin{cases} \dot{\tilde{x}}_2 = \Omega(u_1)\widetilde{x}_4 - \Lambda_2 sign(\widetilde{x}_2) \\ \dot{\tilde{x}}_3 = -H\ \widetilde{x}_{4x} - \Lambda_3 sign(\widetilde{x}_3) \\ \dot{\tilde{x}}_4 = -x_{21}.F.\tilde{x}_4 - \Psi(\hat{X})\widetilde{\theta} - \Lambda_4 sign(\widetilde{x}_2) - \Lambda_5 sign(\widetilde{x}_3) \\ \dot{\tilde{\theta}} = -\eta \end{cases}$$

B. Convergence analysis

Define the state estimation errors $\widetilde{x}_2$, $\widetilde{x}_3$, $\widetilde{x}_1 = x_1 - \widehat{x}_1$, and $\widetilde{\theta} = \theta - \widehat{\theta}$, the parameters estimation errors can be written as:

$$\begin{cases} \dot{\tilde{x}}_2 = \Omega(u)\widetilde{x}_4 - \Lambda_2 sign(\widetilde{x}_2) \\ \dot{\tilde{x}}_3 = H\widetilde{x}_{4x} - \Lambda_3 sign(\widetilde{x}_3) \\ \dot{\tilde{x}}_4 = \Psi(X)\theta - \Psi(\hat{X})\widehat{\theta} - \Lambda_4 sign(\widetilde{x}_2) - \Lambda_5 sign(\widetilde{x}_3) \\ \dot{\tilde{\theta}} = -\eta \end{cases} \quad (11)$$

In order to study the observer stability, we proceed, step by step. To proceed, let us consider the following Lyapunov function:

$$V_2 = \frac{1}{2}\widetilde{x}_2^T\widetilde{x}_2 \quad (12)$$

The time derivative of this function is given by

$$\dot{V}_2 = \widetilde{x}_2^T(\Omega(u_1)\widetilde{x}_4 - \Lambda_2 sign(\widetilde{x}_2)) \quad (13)$$

By chosing $\Lambda_2 > \|\Omega(u_1)\widetilde{x}_4\|$, then $\dot{V}_2 < 0$ therefore, from sliding mode theory, the surface defined by $\widetilde{x}_2 = 0$ is attractive, leading $\widehat{x}_2$ to converge to x_2 in finite time t_0. Moreover,we have $\dot{\widetilde{x}}_2 = 0 \ \forall t \ge t_0$ and then for $\forall t \ge t_0$:

$$sign_{equ}(\widetilde{x}_2) = \Lambda_2^{-1}\left(\Omega(u_1)\widetilde{x}_4\right) \quad (14)$$

where $sign_{equ}$ represents an equivalent form of the $sign$ function on the sliding surface. Now let us consider a (second) Lyapunov V_3 and its time derivative $\dot{V}_3$:

$$\begin{aligned} V_3 &= \frac{1}{2}\widetilde{x}_3^T\widetilde{x}_3 & (15) \\ \dot{V}_3 &= \widetilde{x}_3^T(-H\widetilde{x}_{4x} - \Lambda_3 sign(\widetilde{x}_3)) & (16) \end{aligned}$$

Also by considering $\Lambda_3 > \|H\widetilde{x}_{4x}\|$, then $\dot{V}_3 < 0$ therefore, the surface defined by $\widetilde{x}_3 = 0$ is attractive and we obtain $sign_{equ}(\widetilde{x}_3) = \Lambda_3^{-1}\left(H\widetilde{x}_{4x}\right)$. Then the system becomes (for $\forall t \geq t_0$):

$$\begin{aligned} \dot{\widetilde{x}}_2 &= 0 \text{ and } \dot{\widetilde{x}}_3 = 0 && (17) \\ \dot{\widetilde{x}}_4 &= -A.\widetilde{x}_4 - \Psi(\hat{X})\widetilde{\theta} \quad \text{and} \quad \dot{\widetilde{\theta}} = -\eta && (18) \\ A &= \Lambda_4\Lambda_2^{-1}\Omega(u) + \Lambda_5\Lambda_3^{-1}\left[H; 0_2\right] + x_{21}.F && (19) \end{aligned}$$

Now consider the function of Lyapunov :

$$V_4 = \frac{1}{2}\widetilde{x}_4^T G\widetilde{x}_4 + \frac{1}{2}\widetilde{\theta}^T P\widetilde{\theta} \tag{20}$$

The time derivative of this function is given by

$$\begin{aligned} \dot{V}_4 &= \widetilde{x}_4^T G\dot{\widetilde{x}}_4 + \dot{\widetilde{\theta}}^T P^{-1}\dot{\widetilde{\theta}} && (21) \\ \dot{V}_4 &= -\widetilde{x}_4^T GA.\widetilde{x}_4 - \widetilde{\theta}^T\left(P^{-1}\eta + \Psi(\hat{X})^T G^T\widetilde{x}_4\right) && (22) \end{aligned}$$

An appropriate choice of the adaptation law would be $\eta = P\Psi(\hat{X})^T G^T\widetilde{x}_4$. Knowing that one will have (on the sliding surface) after convergence of x_2 and x_3 (in average): $sign_{equ}(\widetilde{x}_2) = \Lambda_2^{-1}\left(\Omega(u_1)\widetilde{x}_4\right)$ and $sign_{equ}(\widetilde{x}_3) = \Lambda_3^{-1}H\widetilde{x}_{4x}$, finally one can approach of this case (while using $sign(\widetilde{x}_2)$ and $sign(\widetilde{x}_3)$). One must choose G^T to insure that $A > 0$ by the choice of $\Lambda_2, \Lambda_3, \Lambda_4$ and Λ_5 to assure a good convergence (see[4])

IV. Estimation of the road profile

In the previous section we estimate the input forces, here we develop a method based on sliding mode to observe the road profile.

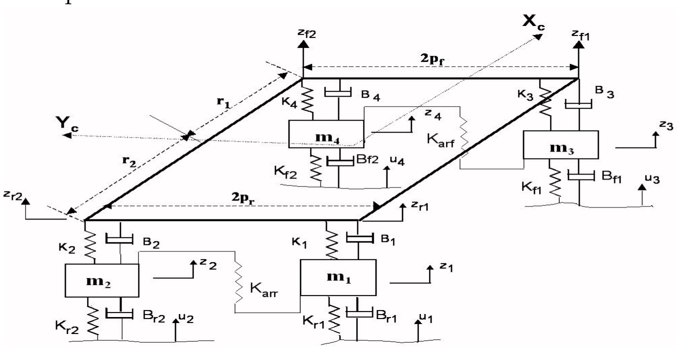

Fig. 3. Vehicle Model for estimation of road profile

A. Observer design

The model considered for this objective is described in figure (5). When considering the vertical displacement along z axis, the model can be written as:

$$M\ddot{q} + C\dot{q} + Kq = \zeta \tag{23}$$

where $(\dot{q}, \ddot{q})$ represent the velocities and accelerations vector respectively. G is related to the gravity effects. M is the inertia matrix, C is related to the damping effects, K is the springs stiffness vector, and $q \in \Re^8$ is the coordinates vector defined by:

$$q = [z_1, z_2, z_3, z_4, z, \theta, \phi]^T \tag{24}$$

The variable $\zeta \in \mathbb{R}^{4\times 4}$ is defined by:

$$\zeta = A_{11}U + B_{11}\dot{U} \tag{25}$$

where $U = (u_1, u_2, u_3, u_4)^T$ is the vector of unknown inputs which characterize the road profile. The matrices A_{11} and $B_{11} \in \mathbb{R}^{4\times 4}$ are:

$$A_{11} = \begin{bmatrix} k_{r1} & 0 & 0 & 0 \\ 0 & k_{r2} & 0 & 0 \\ 0 & 0 & k_{f1} & 0 \\ 0 & 0 & 0 & k_{f2} \end{bmatrix} \tag{26}$$

$$B_{11} = \begin{bmatrix} B_{r1} & 0 & 0 & 0 \\ 0 & B_{r2} & 0 & 0 \\ 0 & 0 & B_{f1} & 0 \\ 0 & 0 & 0 & B_{f2} \end{bmatrix} \tag{27}$$

$(M_i, C_{ij}\ , i, j = 1, 2)$ are defined in $\mathbb{R}^{4\times 4}$, where C_{11} is a positive diagonal matrix. The matrices M, C and K are defined by:

$$M = \begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix},\ C = \begin{bmatrix} C_{11} & C_{12} \\ C_{21} & C_{22} \end{bmatrix} \tag{28}$$

Let us define a state vector: $x = (x_1, x_2)^T$ where $x_1 = q = (x_{11}^T, x_{12}^T)^T$, $x_2 = \dot{q} = (x_{21}^T, x_{22}^T)^T$, $x_{11} = [z_1\ z_2\ z_3\ z_4]^T$, $x_{12} = [z\ \theta\ \phi]^T$, $x_{21} = [\dot{z}_1\ \dot{z}_2\ \dot{z}_3\ \dot{z}_4]^T$ and $x_{22} = [\dot{z}\ \dot{\theta}\ \dot{\phi}]^T$.

We put then the model (23) in the state form as follows:

$$\left\{ \begin{array}{l} \dot{x}_1 = x_2 \quad ; \qquad x_1 = q \\ \dot{x}_{21} = -M_1^{-1}(C_{11}x_{21} + C_{12}x_{22} + K_{11}x_{11} + K_{12}x_{12} + \zeta) \\ \dot{x}_{22} = -M_2^{-1}(C_{21}\ x_{21} + C_{22}\ x_{22} + K_{21}\ x_{11} + K_{22}\ x_{12}) \\ y = [x_{12}^T, x_{22}^T, x_{31}^T]^T \quad ; \quad x_{31} = \dot{x}_{21} \end{array} \right. . \tag{29}$$

The positives matrices K_{11} and C_{11} are defined in $\mathbb{R}^{4\times 4}$, whereas the matrices K_{12} and C_{12} are in $\mathbb{R}^{4\times 3}$. The positives matrices K_{22} and C_{22} are in $\mathbb{R}^{3\times 3}$, whereas the matrices K_{21} and C_{21} are defined in $\mathbb{R}^{3\times 4}$. The system being posed, we can define an observer and study its convergence. In order to estimate the state vector x and to deduce the unknown input vector ζ, we propose the following sliding mode observer:

$$\left\{ \begin{array}{l} \dot{\hat{x}}_{11} = \hat{x}_{12} + H_{11}sign(\tilde{x}_{22}) \\ \dot{\hat{x}}_{12} = \hat{x}_{22} + H_{12}sign(\tilde{x}_{22}) \\ \dot{\hat{x}}_{21} = -M_1^{-1}(C_{11}\ \hat{x}_{21} + C_{12}\ \hat{x}_{22} + K_{11}\ \hat{x}_{11} + K_{12}\ x_{12}) \\ \qquad + M_1^{-1}\hat{\zeta} + H_{21}sign(\tilde{x}_{31}) \\ \dot{\hat{x}}_{22} = -M_2^{-1}(C_{21}\ \hat{x}_{21} + C_{22}\ \hat{x}_{22} + K_{21}\ \hat{x}_{11} + K_{22}\ x_{12}) \\ \qquad + H_{22}sign(\tilde{x}_{22}) \end{array} \right. \tag{30}$$

H_{11} , H_{21} $\in \Re^{4\times 4}$ and H_{12} , H_{22} $\in \Re^{3\times 3}$ represent positive diagonal gain matrices.

B. Convergence analysis

Let us define the state estimation errors $\tilde{x}_i = x_i - \hat{x}_i$ $(i = 1..4)$, then we can write the dynamics estimation errors as follows:

$$\begin{cases} \dot{\tilde{x}}_{11} = \tilde{x}_{12} - H_{11} sign(\tilde{x}_{22}) \\ \dot{\tilde{x}}_{12} = \tilde{x}_{22} - H_{12} sign(\tilde{x}_{22}) \\ \dot{\tilde{x}}_{21} = -M_1^{-1}(C_{11}\ \tilde{x}_{21} + C_{12}\ \tilde{x}_{22} + K_{11}\ \tilde{x}_{11}) \\ \qquad -M_1^{-1}\tilde{\zeta} - H_{21} sign(\tilde{x}_{31}) \\ \dot{\tilde{x}}_{22} = -M_2^{-1}(C_{21}\ \tilde{x}_{21} + C_{22}\ \tilde{x}_{22} + K_{21}\ \tilde{x}_{11}) \\ \qquad -H_{22} sign(\tilde{x}_{22}) \end{cases} \quad (31)$$

In order to study the observer stability and to find the gain matrices, let us consider the following Lyapunov function:

$$V_1 = \frac{1}{2}\tilde{x}_{22}^T \tilde{x}_{22} \quad (32)$$

The time derivative of this function is given by:

$$\begin{aligned} \dot{V}_1 &= -\tilde{x}_{22}^T M_2^{-1}(C_{21}\ \tilde{x}_{21} + C_{22}\ \tilde{x}_{22} + K_{21}\ \tilde{x}_{11}) \quad (33) \\ &\quad -\tilde{x}_{22}^T H_{22} sign(\tilde{x}_{22}) \end{aligned}$$

Since the states are bounded and while choosing the matrix H_{22} components (H_{i2}, $i = 1...3$) such that $H_{i2} > |C_{21}\tilde{x}_{21} + K_{21}\tilde{x}_{11}|$, the equation (33) becomes:

$$\dot{V}_1 = -\tilde{x}_{22}^T M_2^{-1} C_{22}\ \tilde{x}_{22} - \tilde{x}_{22}^T H_{22} sign(\tilde{x}_{22}) < 0 \quad (34)$$

Therefore, the surface $\tilde{x}_{22} = 0$ is attractive and we have the convergence of $\hat{x}_{22}$ towards x_{22} in finite time t_0.

Then according to (31), we have $\dot{\tilde{x}}_{12} = 0$ and consequently $\dot{\tilde{x}}_{11} = 0$. Then we deduce the unknown vector $\tilde{\zeta}$ such as:

$$\tilde{\zeta} = \zeta - \hat{\zeta} = C_{11}\ \tilde{x}_{21} + K_{11}\ \tilde{x}_{11} + M_1 H_{21} sign(\tilde{x}_{31}) \quad (35)$$

Finally, we get the variable ζ:

$$\zeta = \hat{\zeta} + M_1 H_{21} sign(\tilde{x}_{31}) \quad (36)$$

In order to estimate the elements of the unknown vector and according to (25), we can solve the following equation:

$$\zeta = A_{11} U + B_{11}\frac{dU}{dt} \quad (37)$$

When we consider the initial conditions $U(t = 0) = 0$, we obtain from (37), the unknown input vector so that:

$$u_i = \frac{\zeta_i}{a_{ii}}(1 - \exp(-\frac{a_{ii}}{b_{ii}}t)),\ \ i = 1..4 \quad (38)$$

where a_{ii} and b_{ii} are the elements of the matrices A_{11} and B_{11} respectively.

V. Simulation results

In this section, we give some results to validate our approach. In simulated models, forces are generated by the Magic formula tire model [[18]]. The steering angle applied is shown in figure (4). The figures (4) and (5) show the convergence of the estimated state vectors to the actual ones in finite time. In figure (6) we show the asymptotic convergence of the tire force to actual values. Then performance of sliding mode observer and adaptive estimation are satisfactory. The simulation results show that the adaptive observer is robust with respect to parameters uncertainties and the changes on the road conditions.

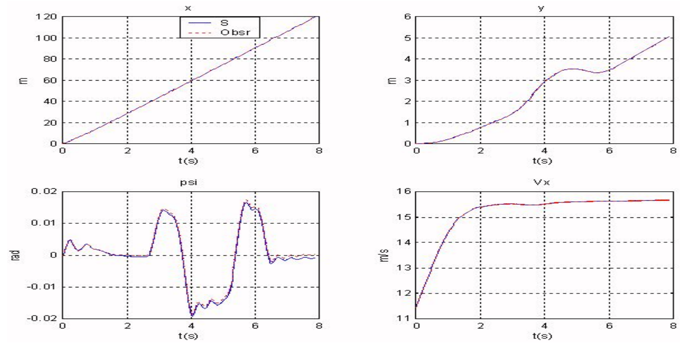

Fig. 4. Estimated and Measured States

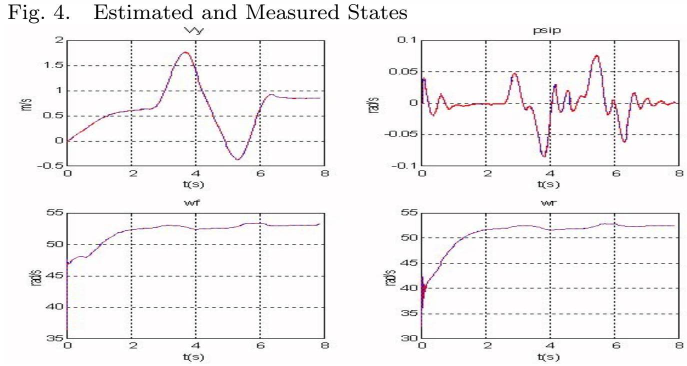

Fig. 5. Estimated and Measured States

After these simulations emphasizing features of the first method, let us consider some experimental results to validate road-profile estimation. The estimated road profile is compared to the profile measured by a longitudinal profile analyzer (LPA) developed at the LCPC Laboratory. It is equipped with laser sensor and accelerometer to measure the elevation of the road profile (see Figure (7)). Through Figure (8), we present the behavior of the front right wheel estimator and the equivalent velocities. In the first two plots on top, the vertical displacement of the wheels are presented. The bottom of this figure, represent the velocities. We can see that the estimated vertical velocities of the wheels are accurate compared to the measured signals. However, some chattering occurs in the estimation of positions. Figure (9) presents a comparison between the measured states, namely, vertical velocity of the body, roll velocity and the pitch velocity, with the estimated one. We notice that the estimated velocities converges well towards observed one. The Figure (10), presents both the measured road profile and the estimated one. We can then observe, that the estimated values are quite close to the true ones (LPA measures).

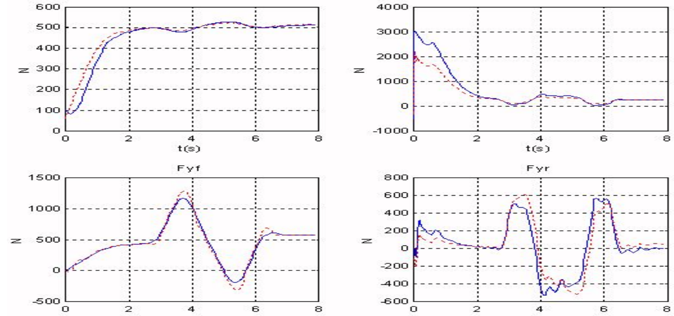

Fig. 6. Estimated and Measured Forces

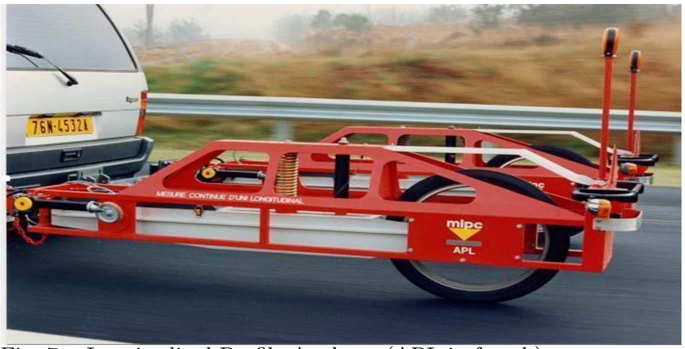

Fig. 7. Longitudinal Profile Analyser (APL in french)

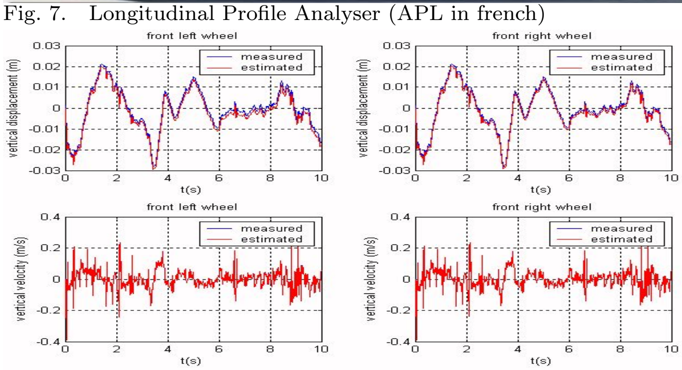

Fig. 8. Estimated and measured displacement of the wheels

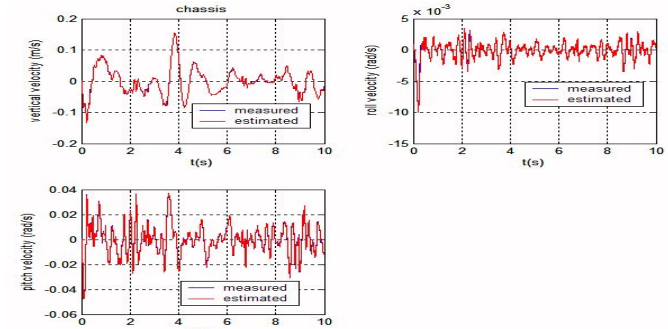

Fig. 9. Estimated and measured velocities of the body

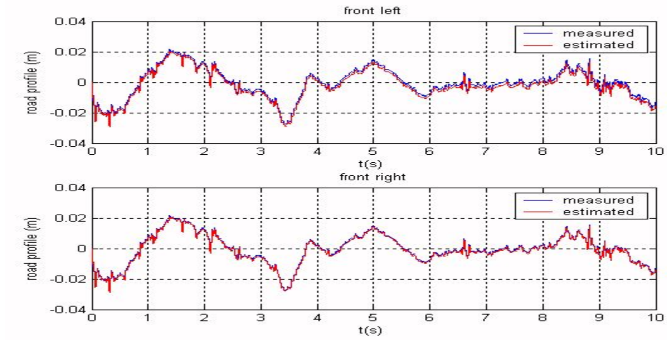

Fig. 10. Estimated road profile compared to the LPA measures

VI. Conclusion

In this paper, we have developed a new estimation method for vehicle dynamics based sliding mode observer. Simulation results illustrate the ability of this approach to give well estimation of both vehicle states, road profile and tire forces. The robustness of the sliding mode observer versus uncertainties on the model parameters has also been shown in simulation. The future work will try to extend these observers and combine estimations.

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