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# LPV-MPC Fault Tolerant Control of Automotive Suspension Dampers <sup>\*</sup>

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**Abstract:** In this short work, the problem of designing a Fault Tolerant dynamic output-feedback controller for Semi-Active Suspension Systems is considered. The suspension system is considered as subject to Loss of Effectiveness (time-varying) faults on each of the four actuators (suspension's dampers). An active reconfiguration fault tolerant scheme is proposed, considering a Linear Parameter Varying (LPV) Model Predictive Control approach. The proposed solution aims to maintain the plant's driving performances (considering handling and comfort indexes) whenever there are sudden actuator faults. These faults are identified through a parallel LPV-based FDD scheme, which is also explained. The performance of the proposed control structure is demonstrated through simulation. Results show the overall good operation of this control scheme, which is compared to other standart control approaches.

*Keywords:* FTC; LPV Control; MPC; Vehicle Semi-Active Suspensions.

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## 1. INTRODUCTION

For several years, the automotive engineering sector has rapidly come to know the use of passive safety features, such as modern seat belts and *state-of-the-art* airbags, evermore present on vehicles. On the other hand, active safety and comfort features are still arising. There is an incipient trade-off when dealing with comfort and road handling performances, for these characteristics are naturally conflicting, as states Hrovat (1997). In order to enhance a vehicle's driving performance aiming road handling **and** ride comfort, a trade-off can be achieved if the vehicle's suspension system is smoothly controlled.

On this matter, *Semi-Active* suspension systems have to be given attention: these systems are efficient and altogether less expensive and energy-consuming than purely active suspensions. This type of suspension is present on new *top-cars* and a good deal of academic and industrial research is focused on this topic, as seen in Lu and DePoyster (2002), Savaresi et al. (2010) and others. Some of the most recent and modern control techniques have been applied for this kind of problem. In Poussot-Vassal et al. (2012) and Tseng and Hrovat (2015), extensive reviews of semi-active suspension control are presented. This study considers, thus, this type of suspension system, where the controlled input is the suspension's damper's damping effort.

Most of the practical control systems are subject to possible faults, failures, component malfunctions and others. These might imply on significant performance degradation or even loss of control and instability. Accordingly, in

recent years attention has been considerably been given to fault tolerant control (FTC) schemes. A FTC system has the goal to allow a system to recover performances when faults occur (or, at least, guarantee continuous stability). FTC systems are categorized into two types: passive and active approaches, as seen in (firstly) Blanke et al. (1997) and Lunze and Richter (2008). Passive approaches usually reside on more conservative control schemes whereas active approaches reside on continuous reconfiguration of the controller, whenever faults are detected. The accurate behaviour of active FTC schemes, then, depends on a solid Fault Detection and Diagnosis (FDD) system. The modulated design (FDI and FTC designed separately) presents its benefits, being more flexible for practical applications and easier to test and implement. This is the approach used herein. FTC applied to suspension control has been studied in rather few works: Tudon-Martinez et al. (2013) presents a fault tolerant semi-active suspension control, considering LPV accommodation; Moradi and Fekih (2014) presents a sliding-mode approach for the same goals. Varrier et al. (2013) and Qiu et al. (2011) are also interesting works. Clearly, this research topic is still open.

The main challenge faced by semi-active suspension control problems is how to handle the **dissipativity constraints** of these dampers. Some of the most recent and modern control techniques have been applied for this kind of problem. Nevertheless, the most natural approach towards optimal control of processes subject to constraints is Model Predictive Control (MPC), as enlightened by Camacho and Bordons (2013). MPC allows to explicitly consider the effect of input and state constraints in the control design process.

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The control of semi-active suspension systems resides in manipulating the damping coefficient of a controlled damper, the actuator from the system's point-of-view. Semi-active dampers have dissipativity constraints that can be treated as an actuator saturation problem. And, as of this, the *MPC* framework presents itself as a plausible and elegant control solution.

Some works have employed a *MPC* approach for semi-active suspension systems, although most of these studies only consider reduced-order vehicle models. However, these models are not sufficient to describe the dynamics of a full vehicle with four semi-active dampers. The idea of solving the control problem at each corner of the car (four separate controllers) might seem persuading and simple enough, but the effects of coupling and load transfer distribution between corners may not be handled, which should lead to degraded performance, as discussed in Nguyen et al. (2016). Nonetheless, some of these studied should be marked: In Poussot-Vassal et al. (2010), a methodology is proposed for optimal semi-active suspension control, based on *MPC*, considering a *quarter-car* vehicle model and previously-measured road disturbances; In Canale et al. (2006), a *fast MPC* scheme is designed for a *half-car* vehicle, where the controller is tuned based on a *quarter-car* suspension model and does not take into account the effect of future disturbances; In Beal and Gerdes (2013), an *MPC* is formulated aiming safe handling performances and validated with experimental results, considering a linear bicycle model and an affine force-input model. Throughout literature, only a few studies have considered multivariable *MPC* semi-active control techniques considering the full car dynamics. This is dealt with herein.

Synthesis and investigation techniques for Linear Parameter Varying (*LPV*) Systems have gathered evermore attention from the Control Systems community, see Mohammadpour and Scherer (2012) and Senane et al. (2013). This kind of system can be understood as a representation methodology to be well suited for the control of dynamical systems with parameter variations. These (*LPV*) structures can be represented as an extension of *LTI* systems, assuming the classical state-space representation matrices are dependent on known **bounded** scheduling parameter  $\rho$ .

This work shall be preoccupied with the use of the *LPV* methodology for two different matters: *i*) to autonomously adjust a faulty vehicle model, that changes due to the presence of faults; *ii*) to represent an extended-state observer, for the case of fault estimation. In terms of the latter, a *LPV*-based fault estimation is able to autonomously adjust and schedule observer (detection filter) gains.

### 1.1 Problem Statement

Considering the given contextualization, the problem dealt within this work is the following: how to design an efficient Fault Tolerant Control scheme, considering actuator (damper) faults, for the control of a full vehicle model with four semi-active suspension systems, maintaining (sufficient) comfort and handling performances whenever a faulty situation occurs? This shall be treated by a twofold: Firstly, a *LPV*-based *FDD* structure shall be presented, in order to detect *Loss of Effectiveness* faults on each of the

suspensions' dampers; Then, a Fault-Tolerant *LPV* Model Predictive Controller is designed, in order to guarantee closed-loop stability and performances, based on (*LPV*) model reconfiguration in the case of fault events. This complete problem is synthesized by the block-diagram in Figure 1.

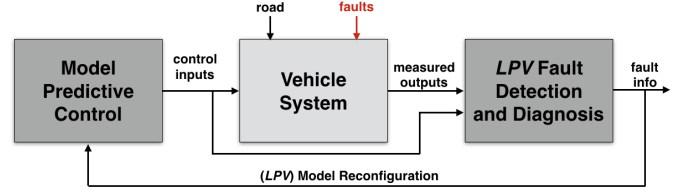


Fig. 1. Outline of Studied Problem

The rest of this work is outlined as follows: The used vehicle models are presented in Section 2; In Section 3, the proposed *LPV* Fault Detection and Diagnosis structure is described and simulation results are presented; In Section 4, performance specifications are detailed and the proposed Fault Tolerant *LPV* Model Predictive controller is designed; Simulation results and a thorough discussion are seen in Section 5 and conclusions are drawn in Section 6.

## 2. NOTATIONS AND PRELIMINARIES

This Section briefly reviews the used notation and recalls some preliminaries. The presented vehicle, tire, spring and damper models are well known in literature and readers are invited to refer to Nguyen (2017) and Poussot-Vassal et al. (2011) for more details.

An automotive suspension system comprises, basically, two components: a spring and a damping (shock absorbing) structure, as represented in Figure 2. These components have to work together to maintain the tire's contact to the ground. The goal of the damping structure is to reduce the effect of travelling upon a rough road by absorbing shock and helping with driving performance, ensuring a smoother and safer drive.

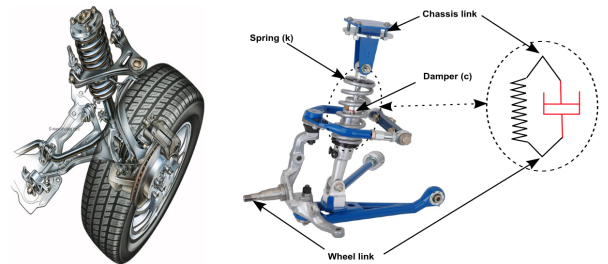


Fig. 2. Outline of Vehicle Suspension Systems

In this work, the tire forces ( $F_{tz_{ij}}$ ) are considered as proportional to the wheel deflection, as given by (1), where  $k_{t_{ij}}$  represent the stiffness coefficients of the tires and  $z_{r_{ij}}$  are the road disturbances that the vehicle is subject to. Each vertical suspension force (at each corners) is represented by  $F_{s_{ij}}$  and, in this study, will be modeled by a spring and a damper with passive and semi-active parts, as given by (2). Remark: this study uses the notation

$z_{def_{ij}} = z_{s_{ij}} - z_{us_{ij}}$  as the suspension deflection. The subscripts  $i - j$  stand, respectively, for front/rear and left/right corners.

$$F_{tz_{ij}} = k_{t_{ij}}(z_{us_{ij}} - z_{r_{ij}}) \quad (1)$$

$$F_{s_{ij}} = k_{ij}(z_{s_{ij}} - z_{us_{ij}}) + c_{0_{ij}}(\dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}) + u_{ij} \quad (2)$$

Throughout literature there are some established dynamical models of vehicles and automotive suspension systems. In this work, two models are used:

1) A reduced-order Quarter-of-Vehicle model (*QoV*) is used to analyse the behaviour of a single vehicle corner. A *state-space* representation can be found if considering system states as (3), disturbance as (4) and the measured outputs as (5).

$$x^{ij}(t) = [z_{s_{ij}}(t) \ \dot{z}_{s_{ij}}(t) \ z_{us_{ij}}(t) \ \dot{z}_{us_{ij}}(t)]^T \quad (3)$$

$$w^{ij}(t) = z_{r_{ij}}(t) \quad (4)$$

$$y^{ij}(t) = [z_{def_{ij}}(t) \ z_{us_{ij}}(t)]^T \quad (5)$$

Thus, the following *LTI* system arises:

$$\begin{aligned} \dot{x}^{ij}(t) &= A^{ij}x^{ij}(t) + B_1^{ij}w^{ij}(t) + B_2^{ij}u_{ij}(t) \\ y^{ij}(t) &= C^{ij}x^{ij}(t) + D_1^{ij}w^{ij}(t) + D_2^{ij}u_{ij}(t) \end{aligned} \quad (6)$$

where the matrices  $A^{ij}$  to  $D_2^{ij}$  are all constant. Note that this representation repeats itself for every corner.

2) A Full Vertical vehicle model (*FVV*) is used for analysis and control goals. It represents a classic 7 degrees of freedom model. This model comprises the chassis dynamics (vertical displacement ( $z_s$ ), roll angle ( $\theta$ ) and pitch angle ( $\phi$ )) and the vertical displacements of the wheels ( $z_{us_{ij}}$ ) at the front/rear - left/right corners ( $i = (f, r)$  and  $j = (l, r)$ ). This (*FVV*) system model can also be represented by a *state-space* representation - this is:

$$\begin{aligned} \dot{x}(t) &= Ax(t) + B_1w(t) + B_2u(t) \\ y(t) &= Cx(t) + D_1w(t) + D_2u(t) \end{aligned} \quad (7)$$

where the system states are given by (8), the controlled inputs are given by  $u = \text{col}\{u_{ij}\}$ , the disturbances are given by (9a) and the measured outputs are seen in equation (9b). Remark:  $A$ ,  $B_1$ ,  $B_2$ ,  $C$ ,  $D_1$  and  $D_2$  are constant matrices. In (8), the time variable  $t$  is suppressed.

$$x = [z_s \ \theta \ \phi \ z_{us_{fl}} \ z_{us_{fr}} \ z_{us_{rl}} \ z_{us_{rr}} \ \dots \ z_s \ \dot{\theta} \ \dot{\phi} \ \dot{z}_{us_{fl}} \ \dot{z}_{us_{fr}} \ \dot{z}_{us_{rl}} \ \dot{z}_{us_{rr}}]^T \quad (8)$$

$$\text{a) } w(t) = \text{col}\{w^{ij}(t)\} \quad \text{b) } y(t) = \text{col}\{y^{ij}(t)\} \quad (9)$$

### 3. FAULT DETECTION AND DIAGNOSIS SCHEME

Now that the used vehicle models have been presented, this Section aims to detail the used *FDD* system. As explained beforehand, this work considers actuator faults upon each of the *ER* dampers of the full suspension system. Considering semi-active suspension systems, a fault can occur due to oil leakage, physical deformation or even the presence of air on the *ER* fluid.

Faults on each of the actuators, in a practical sense, result on the **loss of effectiveness** of these components. For

this these faults can be represented by a multiplicative factor  $\alpha_{ij}$  upon each semi-active damper force  $u_{ij}$ . This multiplicative fault representation has been introduced in Hernández-Alcántara et al. (2016), and introduces a solid framework for the modelling and identification of damper faults. In Figure 3, this fault paradigm is presented.

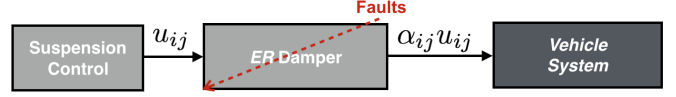


Fig. 3. Damper Loss of Effectiveness Fault Problem

*Remark 1.* In a faultless situation, it is implied that  $\alpha_{ij} = 1$  and, in the worst case,  $\alpha_{ij} = 0$ . For this,  $\alpha_{ij} \in [0, 1]$ . It is also worth noting that even if  $\alpha$  is assumed to be constant, the corresponding additive fault magnitude on  $n$ th semi-active damper is given by  $f_{ij}(t) = (1 - \alpha_{ij})u_{ij}(t)$  (which is time-varying). Thanks to this fault representation framework, it is assumed that each  $\alpha_{ij}$  is slow-varying and, thus,  $\dot{\alpha}_{ij} \approx 0$ .

Synthetically: The problem herein is, thus, to track these fault factors  $\alpha_{ij}$  only through the available measurements  $y^{ij}(t)$ , see equation (5). The approach used in this work to track these factors  $\alpha_{ij}$  consists, basically, on the used of an *LPV* extended observer. This is rigorously detailed with a solid mathematical formalisms, simulation examples and experimental validation results in Morato et al. (2017). Reader is invited to visit this work for further details; this section is compressed.

A reduced-order *FDD* structure can be designed for each of the vehicle's suspensions systems, considering the use of the *QoV* model. For this goal, the following augmented *space-state* representation is written, taking  $x_a^{ij}(t) = [(x^{ij})^T(t) \ \alpha_{ij} \ w^{ij}(t)]^T$ :

$$\begin{aligned} \underbrace{\begin{bmatrix} x_a^{ij}(t) \\ \dot{x}^{ij}(t) \\ \dot{\alpha}_{ij} \\ \dot{w}^{ij}(t) \\ y^{ij}(t) \end{bmatrix}}_{C_a^{ij}} &= \underbrace{\begin{bmatrix} A^{ij} & B_2^{ij}u_{ij}(t) & B_1^{ij} \\ 0 & 0 & 0 \\ 0 & 0 & A_{mw}^{ij} \\ C^{ij} & D_{2_{ij}}^{ij}u_{ij}(t) & D_1^{ij} \end{bmatrix}}_{A_a^{ij}} x_a^{ij}(t) \end{aligned}$$

under the assumption that model of the road profile is known:  $\dot{w}^{ij}(t) = A_{mw}w^{ij}(t)$ .

As the damper semi-active force signal  $u_{ij}(t)$  is *perfectly know* (use of compression force sensors), and **bounded** (due to saturation constraints of each damper) inside the convex set  $\mathcal{U}_{\text{sat}}$  (delimited by the minimal and maximal values of  $u_{ij}$ ), then,  $u_{ij}$  is assumed as a scheduling parameter  $\rho^{ij}(t)$ , as it satisfies  $0 < \rho_{\text{min}}^{ij} \leq \rho^{ij} \leq \rho_{\text{max}}^{ij}$ . Thus, system (10) is *LPV*, polytopic and affine on  $\rho^{ij}$ .

Finally, the *FDD* design is based on the synthesis of an asymptotical state observer, with the form (10), to estimate the value of each fault factor  $\alpha^{ij}$ , by tracking  $x_a^{ij}$ . As usual, the necessity of the synthesis to work is that the dynamics of the estimation error are stable.

$$\begin{aligned} \frac{d\hat{x}_a^{ij}}{dt}(t) &= (A_a^{ij} - L^{ij}C_a^{ij})\hat{x}_a^{ij}(t) - Ly^{ij}(t) \quad (10) \\ \hat{\alpha}_{ij}(t) &= [0 \ \mathbb{I} \ 0] \hat{x}_a^{ij}(t) \end{aligned}$$

To compute the observer matrix gain  $L^{ij}$  of the proposed extended observer (10) and guarantee the stability of the estimation error, this work follows a  $H_2$  (noise filtering) criterion, see Khosrowjerdi et al. (2004), which means that the measurement noise effect on estimated fault factors will be diminished.

To finalize this section of this work, some simulation results are presented to prove the proposed  $FDD$  can be accurately used. The indexes  $i - j$  are omitted for simpler notation purposes. For these goals, a small sinusoidal road profile  $w(t)$  is used in order to represent a series of bumps for a vehicle running on a dry road at constant speed. Figure 4 shows this road profile and the suspension force  $u(t)$ . The suspension damper is initially fault-less ( $\alpha = 1$ ) but, then, a sequence of steps is taken as successive loss of effectiveness faults. This can represent, for instance, a successive oil leakage scenario. The estimation  $\hat{\alpha}$  is seen in Figure 5, in comparison with the actual value of  $\alpha$ . This is a satisfactory simulation result, for the accurateness of the used  $FDD$  approach is very strong. This results points toward strong active fault tolerant control results, as states Blanke et al. (2001).

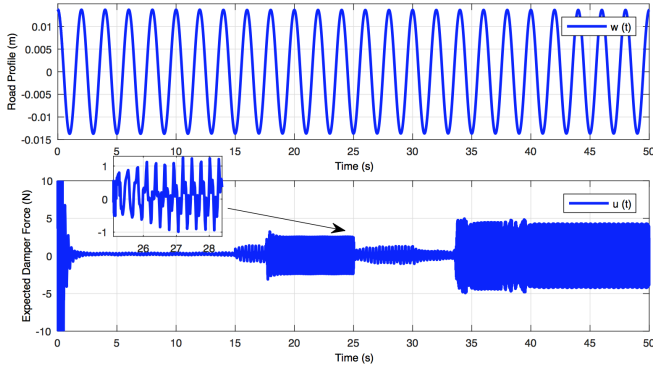


Fig. 4. Simulation Scenario

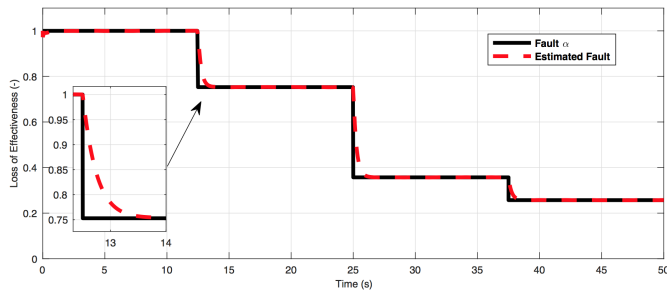


Fig. 5. LPV  $FDD$  Fault Estimation

#### 4. PROPOSED LPV MODEL PREDICTIVE CONTROL SOLUTION AS A FAULT TOLERANT CONTROLLER

This Section presents the main interest of this study: the design of an efficient Fault Tolerant Control scheme, considering damper (actuator) faults in a full suspension

system. Herein, for obvious reasons, the used (control) model is the faulty  $FVV$  model, considering the depicted states (8), measured outputs (9b), disturbances (9a) and control inputs  $\text{col}\{u_{ij}\}$ . This faulty model is given by (7), with matrices  $B_2$  and  $D_2$  changed to  $B_2^{\text{faulty}} = B_2 \times \text{diag}\{\alpha_{ij}\}$  and  $D_2^{\text{faulty}} = D_2 \times \text{diag}\{\alpha_{ij}\}$ .

Firstly, let it be stated that the main goal of a vehicle suspension control is to isolate the body from the road disturbances, without deteriorating road handling. These two objectives can be referred to as *comfort performance* and *handling performance*, respectively, and can be described through the vehicle's  $COG$  acceleration (given by  $\ddot{z}_s$ ) and roll angle (given by  $\theta$ ), as describes Lu and DePoyster (2002). For control design purposes, let two performance indexes be considered, with respect to each control objective:  $J_{\text{comfort}} = \int_0^\tau \ddot{z}_s^2(t)dt$  and  $J_{\text{handling}} = \int_0^\tau \theta^2(t)dt$ , where  $\tau$  represents a given time interval. It is well-known that (physically) these two objectives are conflicting and, for this reason, the chosen control method must take into account a suitable trade-off between these two indexes.

This problem can be solved by a well-posed constrained optimization problem, formulated inside the Model Predictive Control framework. The  $MPC$  control approach to the semi-active suspension problem consists on solving the minimization of the following cost function at every discrete-time step  $k$  in a computational time smaller than the sampling period  $T_s$ :

$$\begin{aligned} J(U, x[k], w[k], N_p, \xi) &= \sum_{j=1}^{N_p} \left[ \xi \left( \frac{\ddot{z}_s[k+j|k]}{\ddot{z}_s^{\text{max}}} \right)^2 \right. \\ &\quad \left. + (1 - \xi) \left( \frac{\theta[k+j|k]}{\theta_{\text{max}}} \right)^2 \right] + \sum_{j=0}^{N_p-1} u^T[k+j|k] Q_u u[k+j|k] \end{aligned} \quad (11)$$

where  $N_p$  is the given prediction horizon,  $u[k+j|k]$ ,  $\ddot{z}_s[k+j|k]$  and  $\theta[k+j|k]$  denote, respectively, the control efforts, the chassis acceleration and roll angle predicted for instant  $k+j$  at instant  $k$ , using the faulty  $FVV$  (prediction) model and considering the initial states  $x[k]$  and disturbance information  $w[k]$ , and

$$U = [u[k|k] \ u[k+1|k] \ \dots \ u[k+N_p-1|k]]^T \quad (12)$$

represents the vector of control efforts inside the horizon (to be optimized).  $Q_u$  is a weighting matrix and  $\xi$  a weighting coefficient that sets the trade-off between handling ( $J_{\text{handling}}$ ) and comfort ( $J_{\text{comfort}}$ ) performances.

This  $MPC$  control problem depends on a  $LPV$  representation of the studied faulty suspension system. This is: as each  $\alpha_{ij}[k]$  is bounded (inside the  $[0, 1]$  set) and estimated by the proposed  $FDD$  approach, these variables can be considered (together) as a scheduling vector  $\rho$ . This is:

$$\rho[k] = [\alpha_{fl}[k] \ \alpha_{fr}[k] \ \alpha_{rl}[k] \ \alpha_{rr}[k]]^T \quad (13)$$

Then, the (discrete-time) system representation shall change from what was presented in equation (7) to:

$$\begin{aligned} x[k+1] &= A_d x[k] + B_{1d} w[k] + B_{2d} \text{diag}\{\rho[k]\} u[k] \\ y[k] &= C_d x[k] + D_{1d} w[k] + D_{2d} \text{diag}\{\rho[k]\} u[k] \end{aligned} \quad (14)$$

where the matrices  $A_d$  to  $D_{2d}$  are the discrete-time equivalent matrices to  $A$  to  $D_2$ .

Then, the scheduling vector  $\rho$  can be considered (for simplicity) to remain **constant** at  $\rho_0$  during the prediction horizon, from the *MPC*'s point-of-view. Thus, finally, the *LPV MPC* problem can be defined as:

$$\begin{aligned} \min_U \quad & J(U, x[k], w_0, N_p, \rho_0) \\ \text{s.t.} \quad & x[k+1] = A_d x[k] + B_{1d} w_0 + B_{2d} \text{diag}\{\rho_0\} u[k] \end{aligned} \quad (15)$$

dampers dissipativity constraints

As usual *MPC* control approaches, the problem (15) is solved at every iteration  $k$  and the control effort to be applied to the real system corresponds to the first entry of the minimized control effort vector (12), solution to (15). An observer has to be designed in order to compute the system states  $x$  from available measurements  $y$ , using the (*FVV*) model. This is easily done in Unger et al. (2013) and in other applications as well.

*Remark 2.* Notice that the velocity of the *FDD* scheme might influence the closed-loop performances. For the controller design, this is ignored, supposing that each fault is perfectly detected. This is an adequate assumption as the convergence speed of the proposed *FDD* scheme is very small (around 1 s), refer to Figure 5.

## 5. RESULTS AND DISCUSSION

Finally, in this Section, simulation results are presented to assess the overall behaviour of the proposed *LPV-MPC* Fault Tolerant Control scheme. For the following results, *LPVMPC* stands for the proposed *LPV-MPC* Fault Tolerant Controller considering the detection and diagnosis of faults provided by the proposed *LPV FDD* scheme; *SMPC*, on the other hand, stands for a simpler *MPC* controller solved with the same weighting functions and controlled inputs, but that does not take into account the effect of the faults (use of fault-free *FVV* model). Reader should also know that a high-frequency measurement noise was added to each measured output ( $y$ ), in order to mimic realistic conditions. The presented results consider the synthesis of the *MPC* controller and *LPV-FDD* observer with the aid of tools *Matlab*, (Mathworks, 2017) and *Yalmip* toolbox, (Lofberg, 2004).  $\xi$  is taken as 1, in order to only prioritize comfort performances of the controlled vehicle, where the effect of faults are more visible.  $N_p$  is taken as a 10-step-ahead horizon. These results consider the dynamics of a reduced-order vehicle, please refer to Vivas-Lopez et al. (2014).

First of all, the chosen simulation scenario, seen Figure 6, is depicted: a vehicle is running at 120 km/h in a straight line on a dry road, when a first 5 cm bump occurs simultaneously on all wheels, to excite the bounce motion and chassis vibration, a second 5 cm bump occurs afterwards, but only on the left wheels, to cause a roll motion and, finally, a third bump occurs at frontal wheels, causing a pitch motion. In terms of faults, these are depicted in Figure 7, where reader can see the loss of effectiveness faults that occur on every corner of the vehicle, in different instants, and their respective estimation by the *FDD* scheme.

The final control results considering the proposed controller as an *FTC* scheme is depicted by Figure 8, where the dynamics of the Chassis' Acceleration are seen. In this Figure, the plot is zoomed at important fault instants

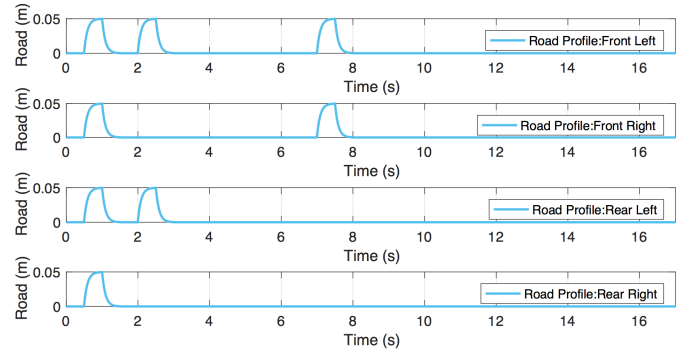


Fig. 6. Simulation Scenario: Road Profile

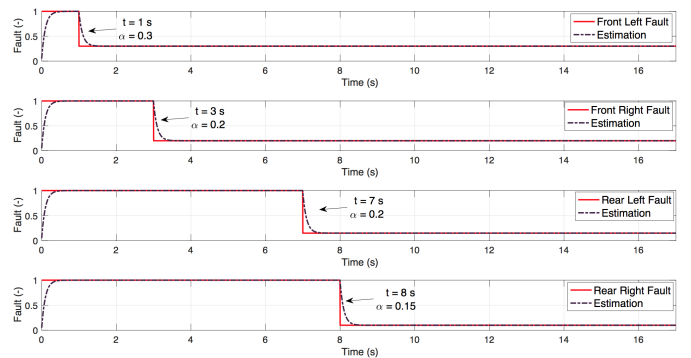


Fig. 7. Simulation Scenario: Faults

wherein it is clear that the *FTC* (*LPVMPC*) can cope well with the presence of damper faults, diminishing its effects on the controlled output  $\ddot{z}_s(t)$ . In comparison to a simpler *MPC* (*SMPC*), the fault-tolerant approach further minimizes the chassis' acceleration, which implies on a more comfortable ride. The effect of faults is more degrading when there is no fault detection or model reconfiguration, as expected, and this causes a delay in terms of rejecting its effects (*SMPC* plot). In terms of *RMS* values of  $\ddot{z}_s(t)$ , considering each approach: for *SMPC*,  $2.004747 \text{ m/s}_{\text{RMS}}^2$ ; and for *LPVMPC*,  $2.002224 \text{ m/s}_{\text{RMS}}^2$ .

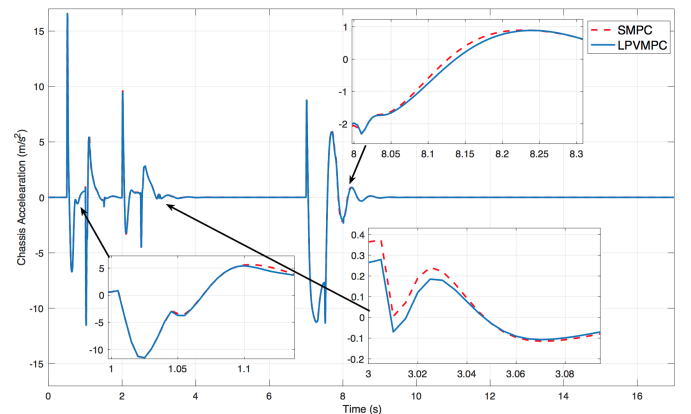


Fig. 8. Chassis' Acceleration,  $\ddot{z}_s(t)$

## 6. CONCLUSIONS

This short work presented the issue of controlling a full vehicle *semi-active* suspension system, during faulty situations on the dampers. A *LPV*-based *FDD* structure is

designed to provide accurate informations on the *loss of effectiveness* of each damper. Then, a *LPV Model Predictive Controller* is designed as a *Fault Tolerant Control* scheme to cope with these faults. The presented results enlighten that the proposed *LPV-MPC* paradigm is an interesting track to be followed towards *Fault Tolerant Control* of *Semi-Active suspensions*. Results prove that the proposed scheme can accurately re-adjust the control law so that faults are overlapped. Still, for further works, authors plan to analyse the effect of badly estimated faults in terms of robustness of the proposed scheme.

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