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LPV-MPC Fault Tolerant Control of Automotive Suspension Dampers ^{*}

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Abstract: The design of a Fault Tolerant dynamic output-feedback controller for Semi-Active Suspension Systems is considered in this work. The suspension system is assumed to undergo *loss of effectiveness* (time-varying) faults on each of the four actuators (suspension's dampers). An active fault tolerant reconfiguration scheme is proposed, considering a Linear Parameter Varying (LPV) Model Predictive Control approach. The proposed solution aims to maintain the vehicle's driving performances (handling and comfort indexes) whenever there are sudden actuator faults. These faults are identified through a parallel Fault Detection and Diagnosis scheme, which is also explained. The performance of the proposed control structure is demonstrated through simulation. Results show the good operation of this control scheme which is compared to other standard control approaches, considering a reduced-size car.

Keywords: Fault Tolerant Control; LPV; Model Predictive Control; Semi-Active Suspensions.

1. INTRODUCTION

For several years, the automotive engineering sector has come to know the use of passive safety features, such as modern seat belts and airbags. Active safety and comfort features are also growing, such as controlled suspension systems. On this matter, *Semi-Active* suspension systems have to be examined: these systems are efficient, while being less expensive and energy-consuming than purely active ones. This type of suspension can be found on new *top-cars* and in a good deal of academic and industrial research, as (Lu and DePoyster, 2002), (Savaresi et al., 2010) and others. These *Semi-Active* dampers can influence the vehicle's driving performance, being able to enhance road handling **and** ride comfort if smoothly controlled. Nonetheless, there is an incipient trade-off when dealing with comfort and handling performances, because these characteristics are naturally conflicting. Most of the practical control systems are subject to possible faults, failures and component malfunctions, just as in vehicle suspensions, where, for example, the damper fluid might leak and the damping is less than the expected. These events imply performance degradation or even loss of control (instability). Accordingly, in recent years, attention has been considerably given to Fault Tolerant Control (FTC) schemes.

FTC aims to allow a system to recover performances if faults occur (or, at least, guarantee some continuous stability). These systems can be either passive or active.

Passive approaches usually stand for more conservative control schemes, as the use of predictive controller seen in (Xu et al., 2017), where the effect of faults is overlapped (passively) by the robustness of the controller. Active approaches, on the other hand, reside in online reconfiguration of the controller, whenever faults are detected, as in (Nazari et al., 2017) where an actuator *FTC* is designed for systems with some polytopic uncertainties. The accurate behaviour of Active *FTC* systems depends on a solid Fault Detection and Diagnosis (*FDD*) system. Literature shows that the modulated design (*FDD* and *FTC* designed separately) presents its benefits, being more flexible for practical applications and easier to test and implement.

The use of *FTC* applied to *Semi-Active* 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.

The main challenge faced by *Semi-Active* suspension control problems is how to handle the dissipativity constraints of the dampers while following some driving performance objective, as enhancing handling and comfort. Some of the most recent and modern control techniques have been applied towards this control problem: in (Poussot-Vassal et al., 2012) and (Tseng and Hrovat, 2015), extensive reviews of semi-active suspension control schemes are presented. Nevertheless, it should be noticed that the most natural approach towards optimal control of processes subject to constraints is Model Predictive Control (*MPC*) (Camacho and Bordons, 2013). The control of *Semi-Active* suspensions consists in changing *online* the damping prop-

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erty of the controlled dampers (the actuator from the system’s point-of-view). The dissipativity constraints of these dampers can be tackled as an actuator saturation problem. Thus, the *MPC* framework presents itself as a plausible and elegant control solution as it allows to explicitly consider the effect of input and state constraints in the control design process.

Some works use *MPC* approaches for *Semi-Active* suspension systems, although most of these studies only consider *quarter-car* vehicle models. However, these models are not sufficient to describe the dynamics of a full vehicle with four dampers. While the idea of solving the control problem at each corner of the car (four separate controllers) might seem convincing and simple enough, the effects of coupling and load transfer distribution between corners may not be handled, which could lead to degraded performance, as discussed in (Nguyen et al., 2016). Nonetheless, some of these studies should be mentioned; 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 few studies have considered multivariable *MPC Semi-Active* control techniques applied to the full car dynamics.

Considering the given contextualization, the global problem dealt within this work is the following: how to design an efficient Fault Tolerant Control scheme for a vehicle with four *Semi-Active* suspensions, considering faults on the dampers, while maintaining (sufficient) comfort and handling performances whenever a faulty situation occurs?

To tackle this issue, this work proposes to use an *MPC* controller, as seen in literature, as a Fault Tolerant scheme, with a Linear Parameter-Varying (*LPV*) model that is able to describe the vehicle in both faulty and faultless situations. Also, to do so, a separate *FDD* system is designed to collect information about whether the damper is faulty, following an extended-observer design methodology, as seen in (Nguyen et al., 2015b). This work can be compared to (Tudon-Martinez et al., 2013), while a new methodology is used, since (*LPV MPC-FTC*) hasn’t yet been seen applied to vehicle suspensions in the literature. The efforts herein were done to demonstrate that handling and comfort performances of a vehicle can be enhanced when using controlled *Semi-Active* suspensions, even if faulty situations occur. Overall good results are obtained and illustrated with the aid of high-fidelity simulations and comparisons to simpler control schemes.

The paper is organized as follows: firstly, the model that describes the controlled vehicle’s dynamics is presented in Section 2; then, the *FDD* system used to collect information about faults is presented in Section 3; the car’s driving performance specifications are detailed in Section 4, wherein the proposed Fault Tolerant Model Predictive controller is minutely designed; finally, simulation results and their discussion are given in Section 5 and the work ends with conclusions.

2. NOTATIONS AND PRELIMINARIES

Firstly, the used notation is briefly reviewed and some preliminaries are recalled. The presented vehicle, tire, spring and damper models are well known in literature and readers are invited to refer to (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 1. 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.

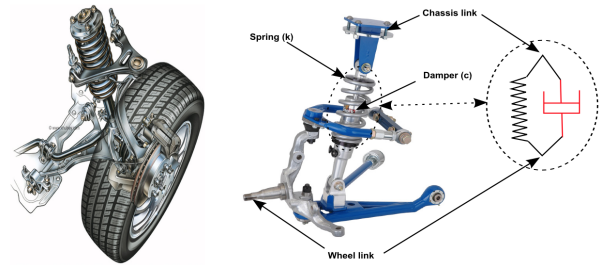


Fig. 1. Outline of Vehicle Suspension Systems

In this work, the vertical tire forces ($F_{tz_{ij}}$) are considered as proportional to the wheel deflection, as given by (1), where $k_{t_{ij}}$ represents the stiffness coefficients of the tires and $z_{r_{ij}}$ are the road disturbances acting on the vehicle. Each vertical suspension force (at each corner), represented by $F_{s_{ij}}$, is modeled by a spring and a damper with passive and semi-active parts, as given by (2), where u_{ij} , the control input, should satisfy some dissipativity constraints¹. Note that $z_{def_{ij}} = z_{s_{ij}} - z_{us_{ij}}$ stands for the suspension deflection. The subscripts (i, j) stand, respectively, for front/rear and left/right corners.

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

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

Throughout literature, there are some well-established dynamical models of vehicles and automotive suspension systems. In this work, 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 of the chassis (z_s), roll angle (θ) and pitch angle (ϕ)) 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 be also given by the following *state-space* representation:

$$\begin{aligned} \dot{x}(t) &= Ax(t) + B_1w(t) + B_2u(t) \\ y(t) &= Cx(t) + D_1w(t) + D_2u(t) \end{aligned} \quad (3)$$

where the system states are given by (4), the controlled inputs by $u = \text{col}\{u_{ij}\}$, and the disturbances and the measured outputs by (5). Note that A , B_1 , B_2 , C , D_1

¹ Note the semi-active damper is represented by an additive term $c_0\dot{z}_{def_{ij}} + u_{ij}$, where u is controlled.

and D_2 are constant matrices. In (4), the time variable t is omitted for simplicity.

$$x = \begin{bmatrix} z_s & \theta & \dot{\phi} & z_{us_{fl}} & z_{us_{fr}} & z_{us_{rl}} & z_{us_{rr}} & \dots \\ \dot{z}_s & \dot{\theta} & \dot{\phi} & \dot{z}_{us_{fl}} & \dot{z}_{us_{fr}} & \dot{z}_{us_{rl}} & \dot{z}_{us_{rr}} \end{bmatrix}^T \quad (4)$$

$$w(t) = \text{col}\{w^{ij}(t)\}, y(t) = \text{col}\{z_{def_{ij}}(t), z_{us_{ij}}(t)\} \quad (5)$$

3. FAULT DETECTION AND DIAGNOSIS SCHEME

This Section details the *FDD* system used in this work. The possible faults that occur on the *Semi-Active* damper may happen due to internal oil leakage, physical deformation or even to the presence of air in the damping fluid. This leads, in practice, to the *loss of effectiveness* of these components. Thus, these faults can be represented by a multiplicative factor α_{ij} upon each *Semi-Active* damper force u_{ij} . This representation, presented in (Hernández-Alcántara et al., 2016), provides a solid framework to deal with damper faults, as summarized in Figure 2.

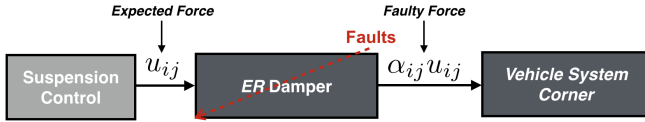


Fig. 2. Damper Loss of Effectiveness Fault Problem

Remark 1. In a faultless situation, $\alpha_{ij} = 1$ and $\alpha_{ij} = 0$ when the actuator is in failure. So, $\alpha_{ij} \in [0, 1]$. Note that even if α is assumed to be constant, the corresponding additive fault magnitude upon the $i-j$ semi-active damper is given by $f_{ij}(t) = (1 - \alpha_{ij})u_{ij}(t)$ (which is time-varying). Thanks to this fault representation, it is assumed that each α_{ij} is slowly varying and, thus, $\dot{\alpha}_{ij} \approx 0$. This is coherent with the considered type of faults, linked to the damper state of health.

The *FDD* problem is, thus, to estimate these fault factors α_{ij} by solely using the available measurements $y^{ij}(t)$, see equation (5), and the expected force signal $u^{ij}(t)$. The approach used in this work consists, basically, in the use of an *LPV* extended observer. This has been done as in (Yamamoto et al., 2015), wherein mathematical formalism and simulation examples are seen.

A reduced-order *FDD* structure is designed for each of the vehicle's suspension systems, considering the use of a *quarter-car* model. An augmented *space-state* representation is written, denoting $x_a^{ij}(t) = [(x^{ij})^T(t), \alpha_{ij}, w^{ij}(t)]^T$:

$$\begin{cases} \dot{x}_a^{ij}(t) = A_a^{ij} x_a^{ij}(t) \\ y^{ij}(t) = C_a^{ij} x_a^{ij}(t) \end{cases}$$

under the assumption that a road profile model is known: $\dot{w}^{ij}(t) = A_{mw} w^{ij}(t)$. Note that assuming to know the road type/model (and not the road profile signal) is not restrictive. Modern cars present cameras and other features that serve to this purpose. This information can come from an adaptive estimator, as done by Tudón-Martínez et al. (2015), or from frequency-wise approaches, as proposed by Unger et al. (2013). A_{mw} can be understood as the *ISO* road surface categories.

Finally, the used *FDD* is based on the synthesis of an asymptotical state observer (6), which estimates the value of each fault factor α^{ij} , by asymptotically tracking x_a^{ij} . Of course, the dynamics of the estimation error must be stable.

$$\frac{d\hat{x}_a^{ij}}{dt}(t) = (A_a^{ij} - L^{ij}C_a^{ij})\hat{x}_a^{ij}(t) - L^{ij}y^{ij}(t) \quad (6)$$

$$\hat{\alpha}_{ij}(t) = [0 \ \mathbb{I} \ 0] \hat{x}_a^{ij}(t)$$

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

A simulation result is presented below to rapidly demonstrate that the used *FDD* is sufficiently accurate. A small sinusoidal $w(t)$ is used to represent a series of bumps as the road profile. Figure 3 shows the computed damper force $u_{fl}(t)$, given by some suspension control algorithm (*sky-hook*, *ground-hook*, *MPC*, etc). The (front-left corner) damper is initially faultless but, suddenly, a sequence of steps simulates *loss of effectiveness* faults (oil leakages). The estimation $\hat{\alpha}$ is given in Figure 4, and compared to the actual value of α . Clearly, the chosen *FDD* approach is very accurate.

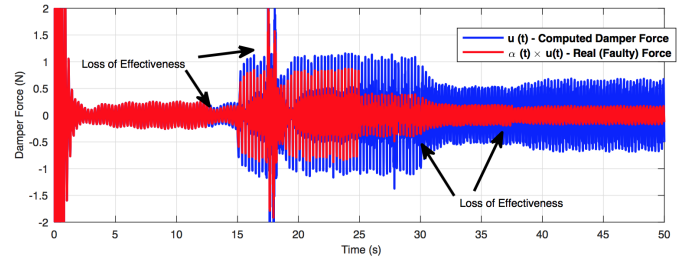


Fig. 3. Damper Force

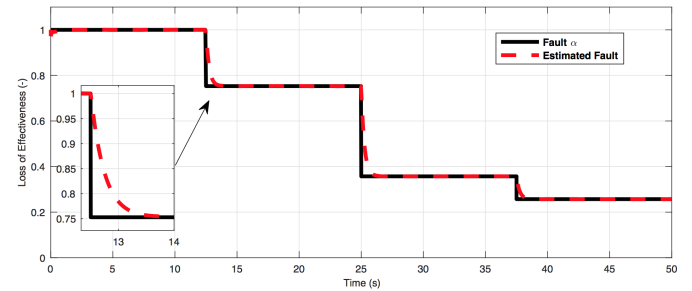


Fig. 4. *LPV FDD* Fault Estimation

4. *LPV MPC* SOLUTION AS A FAULT TOLERANT CONTROLLER

This Section is the main part of this study with the design of an efficient Fault Tolerant Control scheme, when considering damper faults in a full *Semi-Active* suspension system.

Designing an *FTC* scheme requires a model that represents the system when a fault occurs. Considering the *loss of effectiveness* faults on the suspension dampers, the faulty-*FVV* model is given by:

$$\begin{aligned} \dot{x}(t) &= Ax(t) + B_1w(t) + B_2^{\text{faulty}}u(t) \\ y(t) &= Cx(t) + D_1w(t) + D_2^{\text{faulty}}u(t) \end{aligned} \quad (7)$$

where the states are given by (4), disturbances and measured outputs by (5) and control inputs by $u(t)$. In this model, the faulty matrices are:

$$B_2^{\text{faulty}} = B_2 \times \text{diag}\{\alpha_{ij}\}, \quad D_2^{\text{faulty}} = D_2 \times \text{diag}\{\alpha_{ij}\} \quad (8)$$

Recall 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* and *handling performance*, respectively, and can be described through the vehicle's *COG* acceleration (\ddot{z}_s) and roll angle (θ) (Lu and DePoyster, 2002). For control design purposes, let two performance indexes be considered, with respect to each control objective:

$$J_{\text{comfort}} = \int_0^\tau \ddot{z}_s^2(t)dt, \quad J_{\text{handling}} = \int_0^\tau \theta^2(t)dt \quad (9)$$

where τ represents a given time interval. It is well-known that (physically) these two objectives are conflicting and, for this reason, the control solution should take into account a trade-off between these indexes.

This problem can be solved by a well-posed constrained optimization problem, formulated within the Model Predictive Control framework. The *MPC* control approach to the semi-active suspension problem consists in solving the minimization of the following cost function at every step k , in real-time:

$$\begin{aligned} J(U_k, x[k], w[k], N_p, \xi) &= \sum_{j=1}^{N_p} [\xi \left(\frac{\ddot{z}_s[k+j|k]}{\ddot{z}_s^{\max}} \right)^2] \\ &+ (1 - \xi) \left(\frac{\theta[k+j|k]}{\theta^{\max}} \right)^2 + \sum_{j=0}^{N_p-1} u^T[k+j|k] Q_u u[k+j|k] \end{aligned} \quad (10)$$

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 where $U_k = [u[k|k] \ u[k+1|k] \ \dots \ u[k+N_p-1|k]]^T$ is the vector of control efforts inside the prediction horizon (to be optimized). Q_u is a weighting matrix and ξ a weighting coefficient that sets the trade-off between handling (J_{handling}) and comfort (J_{comfort}) performances.

This *MPC* control problem depends on a *LPV* representation of the studied faulty suspension system. Since each $\alpha_{ij}[k]$ is bounded (inside $[0, 1]$) and estimated by the used *FDD* system, these variables are considered as the scheduling vector ρ :

$$\rho[k] = [\alpha_{fl}[k] \ \alpha_{fr}[k] \ \alpha_{rl}[k] \ \alpha_{rr}[k]]^T \quad (11)$$

Then, the (discrete-time) system representation will change from the (continuous-time) one presented in Eq. (7) to:

$$\begin{aligned} x[k+1] &= A_d x[k] + B_{1d} w[k] + B_{2d} \text{diag}\{\rho[k]\} u[k] \\ y[k] &= C_d x[k] + D_{1d} w[k] + D_{2d} \text{diag}\{\rho[k]\} u[k] \end{aligned} \quad (12)$$

where the matrices A_d to D_{2d} are the discrete-time equivalent matrices obtained from A to D_2 . This work considers a sampling period $T_s = 5$ ms, as in Nguyen et al. (2016), given that the dynamics of a vehicle system are fast, specially considering $\ddot{z}_s(t)$ and $\theta(t)$ ²

Assumption 1. As the chosen sampling period is very small, the scheduling vector ρ can be considered **constant** at $\check{\rho}$ (for simplicity) during the prediction horizon ($N_p \times T_s$), from the *MPC*'s point-of-view. Of course, for this to be valid, N_p has to be sufficiently small.

Assumption 2. A road disturbance model (A_{mw}) is known. This information on the type of road profile can be provided by some estimator, as proposed in (Tudón-Martínez et al., 2015). Then, the controller has access to \hat{w} , simulated with this known model.

The Final proposed (*LPV*) *MPC-FTC* solution can be summarized by the block-diagram given in Figure 5 and is defined, mathematically, as:

$$\begin{aligned} \min_{U_k} & \quad J(U_k, x[k], N_p, \check{\rho}) \\ \text{s.t.} & \quad \begin{cases} x[k+1] = A_d x[k] + B_{1d} w[k] + B_{2d} \text{diag}\{\check{\rho}\} u[k] \\ u_{ij}[k] \in \mathcal{D}_{ij}(\check{z}_{def_{ij}}[k], \rho_{ij}[k]) \end{cases} \end{aligned} \quad (13)$$

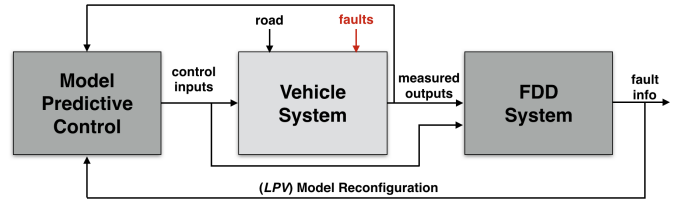


Fig. 5. Outline of Proposed *MPC-FTC* Solution

This minimization problem is solved at every iteration k and the control effort applied at instant k to the real system is the first entry of the control effort vector U_k , solution of (13). Also, an observer is designed in order to compute the system states x from available measurements y . This was easily done, as in (Unger et al., 2013) and in others applications.

Remark 2. The dissipativity constraints of each semi-active damper is given by domain \mathcal{D}_{ij} . In the case of faults, the available damping force is smaller than when healthy, which increases the damping motion $\dot{z}_{def_{ij}}$ - meaning that \mathcal{D}_{ij} shrinks according to ρ_{ij} , as suggests (Nguyen et al., 2015a).

Remark 3. Notice that the dynamics of the *FDD* scheme might influence the closed-loop performances. For the controller design process, this is ignored, supposing that each fault is perfectly detected ($\hat{\alpha}_{ij} = \alpha_{ij}$). This is a realistic assumption as the convergence time of the proposed *FDD* scheme is very small, see Figure 4.

5. RESULTS AND DISCUSSION

Simulation results are presented next to assess the behaviour of the *LPV-MPC* Fault Tolerant Control scheme. Two scenarios are tested: one to evaluate the effect of different values of N_p , given the possibility of violation of Assumption 1; the other to show the efficiency of the

² Note that this sampling rate is adequate for actual *top*-cars.

proposed scheme, when compared to simpler *MPC*s, in terms of fault tolerance. The results are obtained with the aid of softwares packages *Matlab* and *Yalmip*. Herein, the dynamics of a reduced-order vehicle are considered³. Note, also, that a high-frequency measurement noise is added to each measured output (y), in order to mimic realistic conditions. The chosen road profile ($w(t)$) represents the car is running at 120 km/h in a straight line on a dry road, when it encounters a sequence of three bumps that excite, sequentially, bounce, roll and pitch motion.

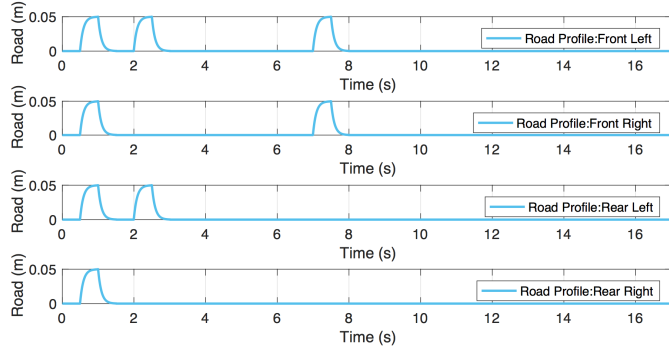


Fig. 6. Simulation Scenario: Road Profile

5.1 Scenario 1

For the first scenario, ξ in (10) is taken as 0.5, in order to set a good trade-off between *handling* and *comfort performances*. N_p , on the contrary, is tested with different values, in order to conclude if Assumption 1 is violated and whether the controller has Fault-Tolerance (*FT*) considering damper *loss of effectiveness* faults. For this test, the simulated faults on the four *Semi-Active* dampers are given by Figure 7. Clearly, multiple faults occur, with varying $\alpha_{ij} \in (0, 1]$.

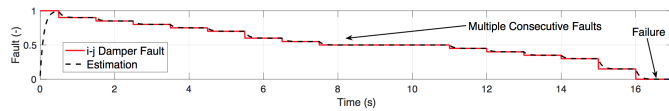


Fig. 7. Scenario 1: Same Faults for all dampers

Table 1 synthesizes the obtained results, showing the influence of N_p on the average values for J , RMS of $\ddot{z}_s(t)$ and of $\theta(t)$. The conclusion about *FT* is given with respect to whether the controller presents better performances when it considers the information on $\rho[k]$. Clearly, the best results are achieved with $N_p = 10$ (50ms), where *FT* is guaranteed and the minimal values were found (best performance). Remark, still, that, for larger N_p , the average computational time t_c is greater than T_s , which would be unable a practical implementation.

5.2 Scenario 2

For the second scenario, N_p is fixed as a 10-steps-ahead horizon, while ξ is set to 1, so only the Chassis Acceleration behaviour is analysed (*comfort performances*), wherein the effect of damper faults is more visible. The same road profile is used (Figure 6), but the simulated faults (and their estimations by the *FDD* scheme) are now depicted

³ See <http://www.gipsa-lab.fr/projet/inove/>.

Table 1.

\times	t_c	mean J	$RMS \ddot{z}_s(t)$	$RMS \theta(t)$	<i>FT</i> ?
$N_p = 2$	3.1 ms	0.8761	2.1302	0.0259	Yes
$N_p = 5$	3.2 ms	0.9169	2.0665	0.0258	Yes
$N_p = 10$	3.9 ms	0.9149	2.0529	0.0258	Yes
$N_p = 15$	4.9 ms	0.9193	2.0597	0.0258	No
$N_p = 25$	5 ms	0.9183	2.0658	0.0259	No
$N_p = 50$	7.5 ms	0.9207	2.0656	0.0259	No
$N_p = 100$	12 ms	0.9207	2.0655	0.0259	No

by Figure 8. The *loss of effectiveness* occurs at the four corners at different instants (1, 3, 7 and 8s) with different values for α_{ij} . In the following Figures, *LPVMPC* stands for the proposed *LPV-MPC* Fault Tolerant Controller, scheduled by the vector of estimated faults $\hat{\rho}[k]$. For comparison goals, *SMPC* stands for a simpler *MPC* controller solved with the same weights and inputs, but not considering the effect of the faults (use of fault-free *FVV* model (3); ρ is constant at $1_{1 \times 4}$). The achieved control results are depicted in Figure 9, which gives the dynamics of the Chassis Acceleration. The plot is zoomed at important fault instants to show that the *FTC* (*LPVMPC*) adapts well to the presence of damper faults, diminishing their effects on the controlled output $\ddot{z}_s(t)$.

Compared to the *SMPC*, the fault-tolerant approach further minimizes the chassis acceleration, which leads to a more comfortable ride. The improvements are not huge because this is a reduced (small) vehicle, and, thus, small changes in \ddot{z}_s do influence the passenger's comfort. Using a large vehicle model, the order of magnitude of \ddot{z}_s would also enlarge. The effect of faults is more degrading when there is no fault detection or model reconfiguration, as expected, and this results in a much slower response to reject their effects (*SMPC* plot). Also, it is important to remark that the dissipativity constraints of the dampers are respected. In Figure 10, the (front-left) damper force and its feasible region are seen; results are similar for the other corners. \mathcal{D}_{fl} (\mathcal{D}) has shrunk according to α_{fl} and only the *LPVMPC* considers this fact, while the *SMPC* remains infeasible. Also, note that the complexity of both *MPC*s is similar, as both resort to optimization problems to be solved within T_s by simple microcontrollers.

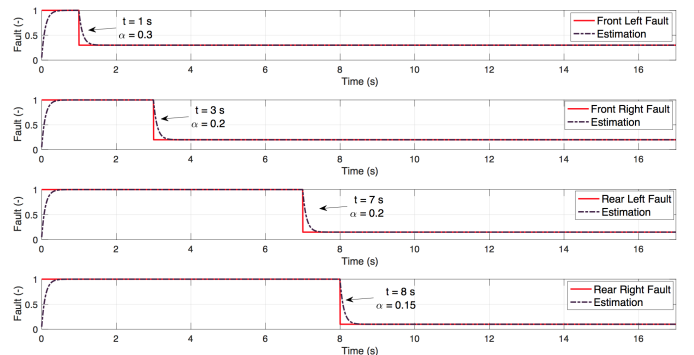


Fig. 8. Scenario 2: Faults for Each Damper

6. CONCLUSIONS

This work presented the issue of controlling a full vehicle *Semi-Active* suspension system, subject to faulty situations on the dampers. A *FDD* system is used to provide

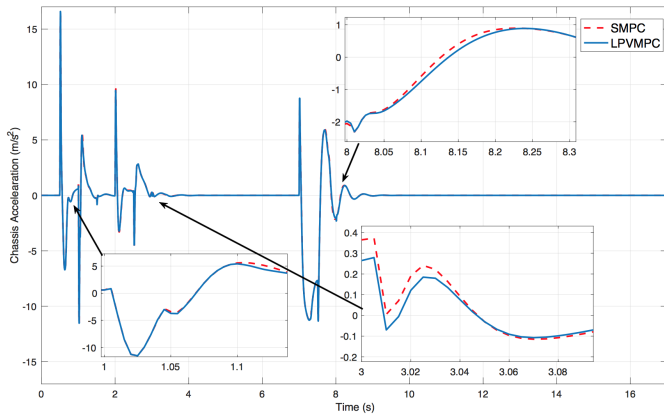


Fig. 9. Chassis' Acceleration, $\ddot{z}_s(t)$

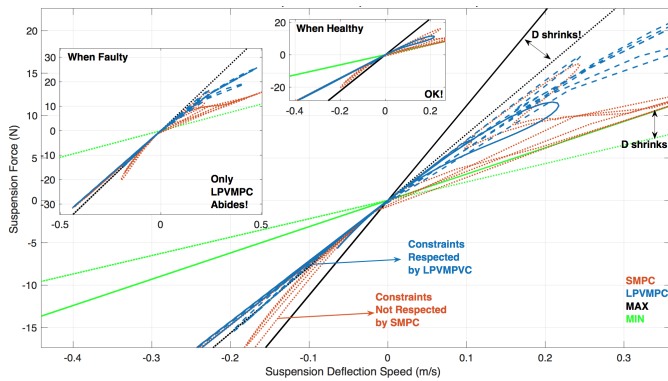


Fig. 10. (Front-Left) Damper Dissipativity Constraints

accurate informations on the damper faults. Then, a *LPV* Model Predictive Controller is designed as a Fault Tolerant Control scheme to cope with these faults. The results enlighten the interest of the proposed *LPV-MPC* paradigm to the development of *FTC* of *Semi-Active* suspensions. Results show that the proposed scheme can accurately re-adjust the control law so that faults are mitigated. For further works, the analysis of badly estimated faults in terms of robustness of the proposed scheme will be made.

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