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**LPV/ H_∞ Fault Tolerant control for automotive semi-active
suspensions using roll and pitch monitoring information**

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ABSTRACT

This paper is concerned with an innovative LPV/ H_∞ fault tolerant control (Linear Varying Parameters) strategy that aims to manage the suspensions dampers failures or malfunctions. Indeed, the vehicle vertical dynamics are mainly affected by the suspension system conditions. The failure of one of the suspensions dampers can cause huge load transfers in the dynamical behaviour of the car. The increasing roll (left/right bounce) and pitch (front/ rear bounce) can lead to the vehicle instability and to the loss of manoeuvrability during dangerous driving situations.

The main idea of this paper is to use a smart monitoring strategy based on the supervision of the roll and pitch dynamics, considering a healthy vehicle behaviour as a reference, to detect the dampers failures. Indeed, taking into account the magnitude and the sign of each one of the roll and pitch dynamics, the localisation of the faulty damper may be possible.

Then, this information is used to generate the varying parameters that allow designing the LPV/ H_∞ fault tolerant robust controllers. The LPV framework will allow reconfiguring the multivariable suspension robust controllers and allocating the accurate suspensions efforts in the four corners of the car, to compensate the loss or lack of the force provided by the faulty damper. This control allocation through the LPV control structure is very interesting since it changes smoothly and progressively, depending on the state of the faulty dampers and it will ensure the vehicle stability by using an online suspension control reconfiguration. Simulations are achieved on a nonlinear full vehicle model validated by experimental procedure on a real vehicle (Renault Mégane Coupé) in MIPS (Mulhouse control system laboratory, France).

1.1 Introduction

Vehicle vertical dynamics are affected by many interrelated sub-systems of the car aim at improving passengers comfort and especially vehicle safety and road

holding. Among all sub-systems affecting the vertical vehicle dynamics, suspension systems play a key role for vehicle handling in critical situation since they ensure the

link between the wheels and the chassis, see [1]–[3]. Several types of suspension systems have been developed and commercialized. In the last decade, semi-active suspensions have received a lot of attention by both academic and

industrial communities, see [4]–[7], since they provide the best compromise between cost (energy, volume, and number of sensors) and performance (road holding, comfort and vehicle behaviour).

Indeed, the increasing number of actuators and sensors gives more possibility of the vehicle dynamics control, but also may cause some issues in the case of equipment failures. The risks of accident is then very high. A lot of studies have proposed several strategies for the vehicle dynamics control (see Kiencke and Nielsen (2000), Milliken and Milliken (1995), Gillespie (1992)).

FTC (Fault Tolerant Control) main objective is to keep the normal operation system when some malfunctions and/or failures appear (see Blanke et al. (1997)). Then,

this kind of control aims at ensuring the closed-loop system stability, and some level of performance, which could be degraded. The most intuitive method is the physical

redundancy with duplication of actuator and sensor components, but since the information redundancy is quiet expensive, some innovative works have focused on estimation and control algorithms to handle the system malfunctions.

In this work, a specific type of semiactive suspension is under interest, namely, the Magneto Rheological Dampers (MRDampers, see [8]–[10]). While some of the authors works have been concerned with global chassis control using active or semi-active suspension [11], [12], the fault tolerant control problem of such systems has been considered only in [13] where a pre-defined distribution of the suspension forces (computed from the steady state behaviour) is used o compensate a damper oil

leakage. This study focuses on the fault tolerant control reconfiguration of MR semi-active dampers. Indeed, few works have been concerned with the control reconfiguration in the presence of suspension system malfunctions or failures.

Semi-active dampers allow the development of novel techniques to model, monitor and control an intelligent suspension system with fault-tolerance

features to improve the process reliability. A Fault-Tolerant Controller (*FTC*) in a suspension system is designed to maintain the desired comfort and road holding performances as much as possible when a fault occurs.

While detecting a damper malfunction, the proposed strategy aims at keeping the vehicle stability and performance through an adequate distribution of the 3 remaining healthy actuators.

The characteristics of magneto rheological dampers allow to compensate the lack of the vertical force in the faulty suspension corner by reconfiguring the global suspensions control.

In this study, a new multivariable LPV/Hinf Global Chassis allocation control strategy is proposed. To solve that problem a new LPV/H1 fault tolerant control is introduced to manage the deterioration of the vertical dynamics by using a varying parameter that coordinate the use of the healthy dampers. The main idea involves 2 steps. First, a monitoring system is introduced to evaluate the state of health of the suspension system. Here, the lateral and longitudinal load transfer induced by a damper malfunction is considered caused mainly by the pitch and the roll dynamics.

To achieve these objectives, the authors have chosen to fix the structure of the LPV/H1 controller by making the LMI's orthogonal with parameters dependency, as follow:

$$\begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho)C_c^0(\rho)}_{C_c(\rho)} x_c(t) \quad (1)$$

The suspension forces distribution is obtained through the matrix $U(\rho)$:

$$U(\rho) = \begin{pmatrix} \rho & 0 & 0 & 0 \\ 0 & \rho & 0 & 0 \\ 0 & 0 & \rho & 0 \\ 0 & 0 & 0 & \rho \end{pmatrix} \quad (2)$$

where ρ_i are the varying parameters given by the considered suspensions monitoring strategies.

Here, two varying parameters are taken into account based on the roll and pitch dynamics (ρ_r and ρ_p , respectively) that affect greatly the vehicle behaviour.

The paper is organised as follows: Section 2 briefly presents the vehicle and MR damper models used for synthesis and validation purposes. Section 3 is devoted to the main contribution of the paper, i.e a LPV/Hinf fault tolerant control of vehicle dynamics. The performance analysis is done in Section 4 with time domain simulations performed on a complex nonlinear full vehicle model. Conclusions and future works are given in the last section.

Paper notations and vehicle parameters:

Parameter/Variable	Description	Value
$a_{1,2}$	MR damping coefficients	37.8 s/m & 22.1 m ⁻¹
$b_{1,2}$	Passive damping coefficients	2,830 Ns/m & -7,897 N/m
f_c	MR dynamic yield force	600.9 N/A
I_{xx}, I_{yy}	Roll and pitch inertia	844 & 4434 Kg · m ²
$k_{s1,2}, k_{s3,4}$	Front and rear spring stiffness	43,500 & 39,300 N/m
$k_{t1,2,3,4}$	Tire stiffness	230,000 N/m
l_f	Distance COG* - front track	1.56 m
l_r	Distance COG* - rear track	2.11 m
m_s	Total sprung mass	2,032 Kg
$m_{us1,2}$	Front unsprung masses	81.5 Kg
$m_{us3,4}$	Rear unsprung masses	140 Kg
t_f	Distance COG* -front left tire	0.85 m
t_r	Distance COG* -rear left tire	0.85 m
F_{pi}	Passive damping forces	$i = 1, \dots, 4$
F_{sai}	Semi-active damping forces	$i = 1, \dots, 4$
F_{si}	Suspension forces	$i = 1, \dots, 4$
F_{ti}	Tire forces	$i = 1, \dots, 4$
z_{ri}	Ground vertical positions	
z_{usi}	Unsprung mass positions	
\dot{z}_{usi}	Unsprung mass velocities	
\ddot{z}_{usi}	Unsprung mass accelerations	
z_{si}	Sprung mass positions	
\dot{z}_{si}	Sprung mass velocities	
\ddot{z}_{sc}	COG* acceleration of m_s	
$\ddot{\theta}$	Roll acceleration of m_s	
$\ddot{\phi}$	Pitch acceleration of m_s	

* COG Center Of Gravity

2. FULL VEHICLE MODELING

A seven degrees of freedom model of the vertical dynamics of a full vehicle, including a semi-active suspension system, is considered in this study :

$$\begin{aligned}
 m_s \ddot{z}_{s_c} &= -(F_{s_1} + F_{s_2} + F_{s_3} + F_{s_4}) \\
 I_{xx} \ddot{\theta} &= (F_{s_1} - F_{s_2}) t_f + (F_{s_3} - F_{s_4}) t_r \\
 I_{yy} \ddot{\phi} &= (F_{s_4} + F_{s_3}) l_r - (F_{s_2} + F_{s_1}) l_f \\
 m_{us_i} \ddot{z}_{us_i} &= F_{s_i} - F_{t_i}
 \end{aligned}$$

where the vertical suspension force at each corner (F_{s_i} , with $i = 1, \dots, 4$) is composed by the spring (stiffness k_{s_i}) and the semi-active damping force (F_{sai}), as:

$$F_{s_i} = k_{s_i} (z_{s_i} - z_{us_i}) + F_{sai}$$

while the tire vertical force, with stiffness k_{t_i} , is given by:

$$F_{t_i} = k_{t_i} (z_{us_i} - z_{r_i})$$

The semi-active force is modeled following ,as :

$$\begin{aligned}
 F_{sai} &= \underbrace{b_1 (\dot{z}_{s_i} - \dot{z}_{us_i}) + b_2 (z_{s_i} - z_{us_i})}_{F_{p_i}} + F_{I_i}(I) \\
 F_{I_i}(I) &= I \cdot f_c \cdot \tanh [a_1 (\dot{z}_{s_i} - \dot{z}_{us_i}) + a_2 (z_{s_i} - z_{us_i})]
 \end{aligned}$$

where I is the electric current and F_{I_i} is the controlled semiactive force to improve the comfort and/or maintain the road holding. Note that for $I = 0$, F_{sai} reduces to the passive damping force F_{p_i} of the suspension system.

3. LPV/Hinf fault tolerant control of vehicle dynamics

In this section, the proposed strategy is developed in two steps : the first one is the fault detection based on the roll and pitch dynamics of the car, by using the thresholds provided by the vehicle constructor. The second one, is using the information from the fault detection strategy to achieve the desired performance objectives.

3.1 Fault detection and suspension behaviour monitoring :

First, the sprung mass positions and velocities at each corner can be linearly derived from the vehicle dynamics, as follows :

$$\begin{array}{ll}
 z_{s1} \approx z_{sc} + l_f \phi - t_f \theta & z_{s3} \approx z_{sc} - l_r \phi - t_r \theta \\
 z_{s2} \approx z_{sc} + l_f \phi + t_f \theta & z_{s4} \approx z_{sc} - l_r \phi + t_r \theta \\
 \dot{z}_{s1} \approx \dot{z}_{sc} + \dot{\phi} l_f - \dot{\theta} t_f & \dot{z}_{s3} \approx \dot{z}_{sc} - \dot{\phi} l_r - \dot{\theta} t_r \\
 \dot{z}_{s2} \approx \dot{z}_{sc} + \dot{\phi} l_f + \dot{\theta} t_f & \dot{z}_{s4} \approx \dot{z}_{sc} - \dot{\phi} l_r + \dot{\theta} t_r
 \end{array}$$

The variables and parameters of this full vehicle model are described in previously introduced table. It can obviously seen that the roll and pitch dynamics affect the vertical behaviour of the vehicle.

In this study, we are considering that : lateral load transfers from the right to the left (positive sign) and the longitudinal load transfers from the front to the rear (positive sign).

Based on the construction description on the considered car Renault Megan Coupé, the following thresholds are established for an acceptable range of variation :

The bounds of the roll motion of the car is $[-15, 15]^\circ$.

The bounds of the pitch motion of the car is $[-22, 22]^\circ$.

Then, when the values of these dynamics are beyond this range, no fault is detected, but when the roll and pitch dynamics take values outside this set of variation the faulty damper is detected using the following table :

	Fsrr	Fsrl	Fsfr	Fsfl
Roll	+	-	+	-
Pitch	-	-	+	+

By comparing the values of the roll and the pitch at each moment to this table, we can detect and then be will be compensated thanks to allocation control strategy.

The varying parameters generation used then is the following :

For the roll :

$$\left\{ \begin{array}{l} F_{z_l} = m_s \times g/2 + m_s \times h \times a_y/l_f \\ F_{z_r} = m_s \times g/2 - m_s \times h \times a_y/l_r \\ \rho_{\text{roll}} = \frac{|(\delta_{fl}F_{z_{fl}} + \delta_{rl}F_{z_{rl}}) - (\delta_{fr}F_{z_{fr}} + \delta_{rr}F_{z_{rr}})|}{|(F_{z_{fl}} + F_{z_{rl}} + F_{z_{fr}} + F_{z_{rr}})|}; \end{array} \right.$$

For the pitch :

$$\left\{ \begin{array}{l} F_{z_f} = m_s \times g/2 + m_s \times h \times a_y/\mathbf{tf} \\ F_{z_r} = m_s \times g/2 - m_s \times h \times a_y/\mathbf{tr} \\ \rho_{\text{pitch}} = \frac{|(\delta_{fl}F_{z_{fl}} + \delta_{rl}F_{z_{rl}}) - (\delta_{fr}F_{z_{fr}} + \delta_{rr}F_{z_{rr}})|}{|(F_{z_{fl}} + F_{z_{rl}} + F_{z_{fr}} + F_{z_{rr}})|}; \end{array} \right.$$

4. Simulation results

Time domain simulations are performed on the full nonlinear vehicle model given in Section 2. For sake of completeness, the results of the proposed LPV/H ∞ fault tolerant control are denoted "LPV strategy" in red and compared to the "vehicle with the damper failure" in blue.

To test the efficiency of the proposed LPV/H ∞ FTC of vehicle roll dynamics under semi-active damper malfunction, the following scenario is used:

The vehicle runs at 100km/h in straight line on wet road ($\mu= 0.5$, where μ is a coefficient representing the adherence to the road). In this scenario, The front right damper of the vehicle is considered faulty (a failure of 70% on the nominal behaviour of the healthy dampers). The vehicle meets 5cm bump on the left wheels (from $t = 0.5$ s to $t = 1$ s), then another one on the right wheels (from $t = 3$ s to $t = 4$ s).

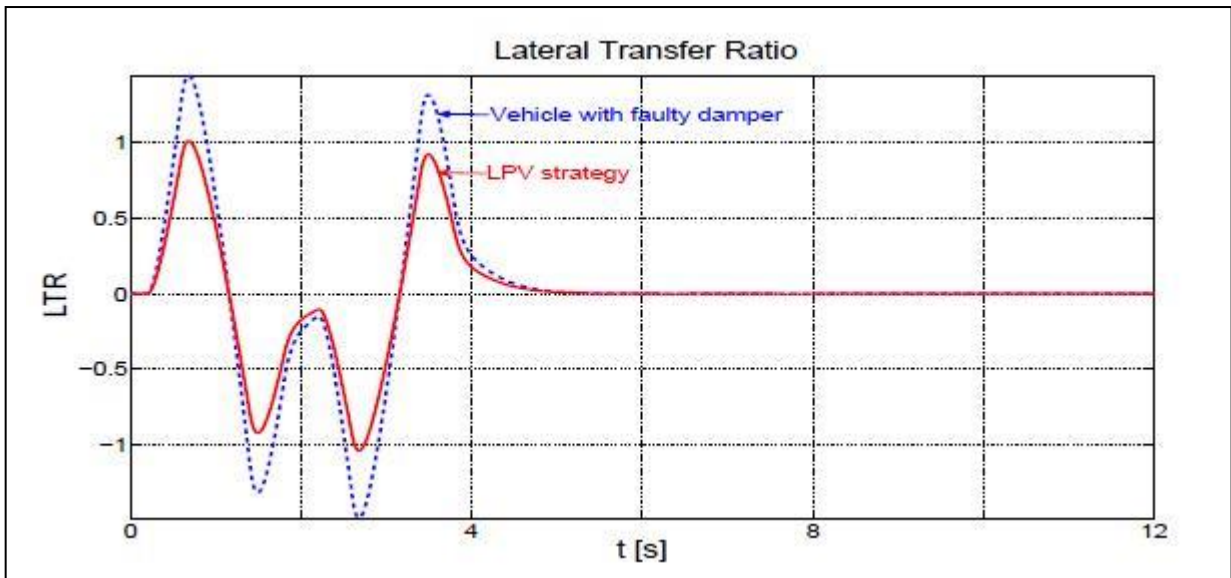


Fig.1. Lateral load transfer

Fig. 3 shows the lateral load transfer generated by the driving scenario; based on it, the scheduling parameters ρ_{roll} , ρ_{pitch} are calculated.

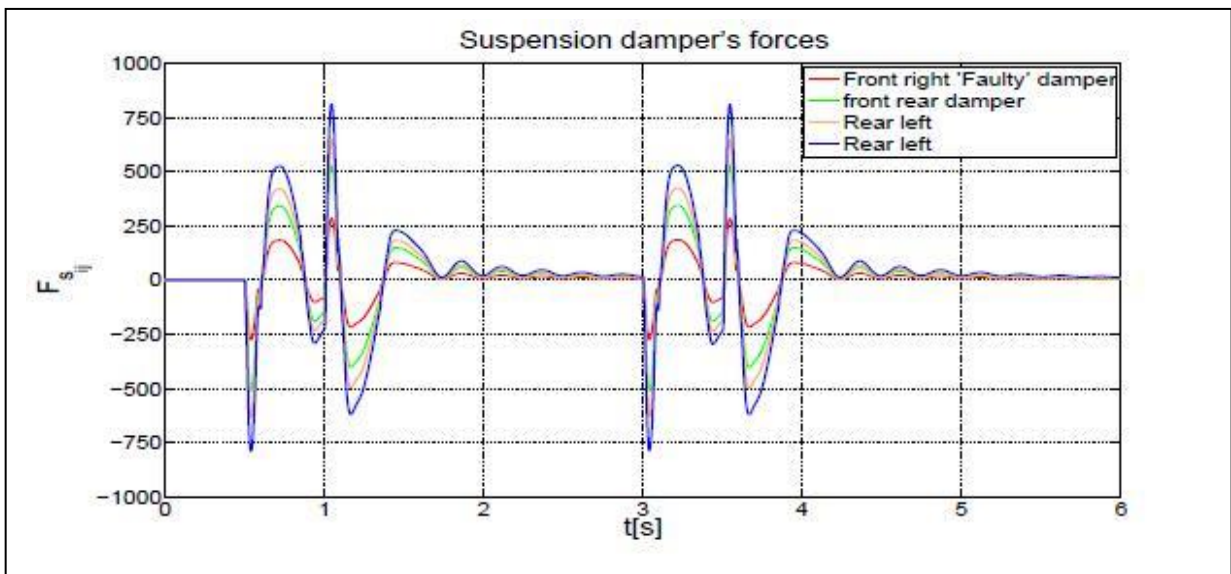


Fig.2. Suspension damper's forces: the faulty and healthy dampers efforts

In Fig.2 , the 4 semi-active dampers efforts provided by the designed fault tolerant LPV/H1 controller are given. It is clear that the failure occurs on the front rear damper which can not provide more then 30% of the nominal force of the healthy MR dampers. Also, it can be seen that the dampers forces distribution is scheduled, following the varying parameter ρ_l (generated by monitoring the lateral transfer ratio). The suspensions forces provided on the right side of the vehicle are larger than those on the left side, due

to the big load supported by their dampers. Moreover, the force provided by the front right damper is greater than the one provided by the rear right one, because it compensates the load due to the front left damper.

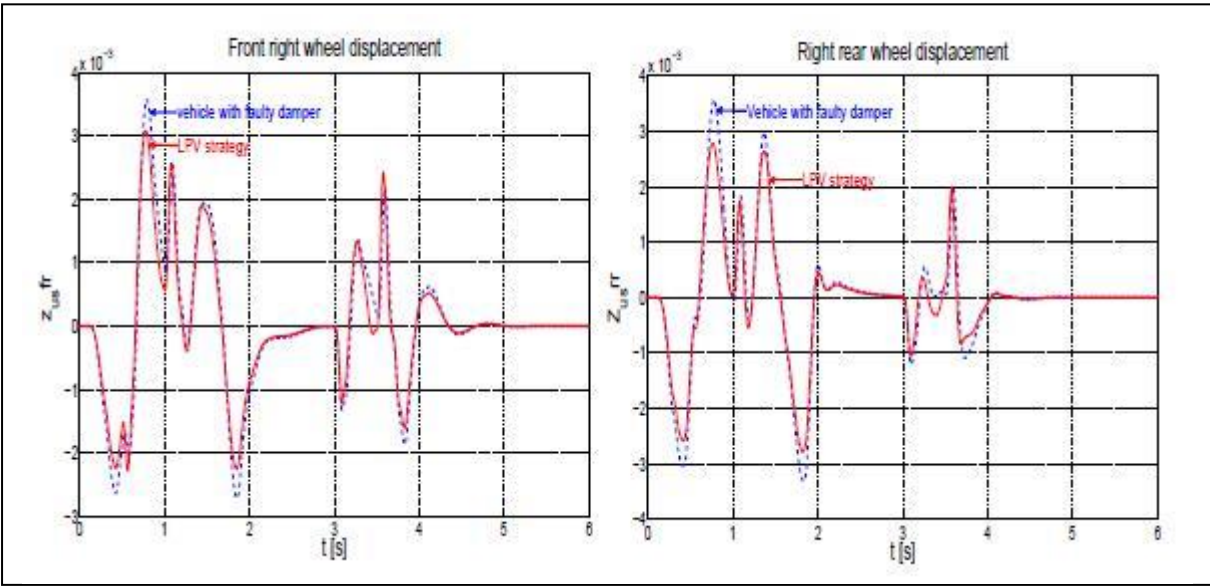


Fig.3. Wheel displacement in front/rear right corners.

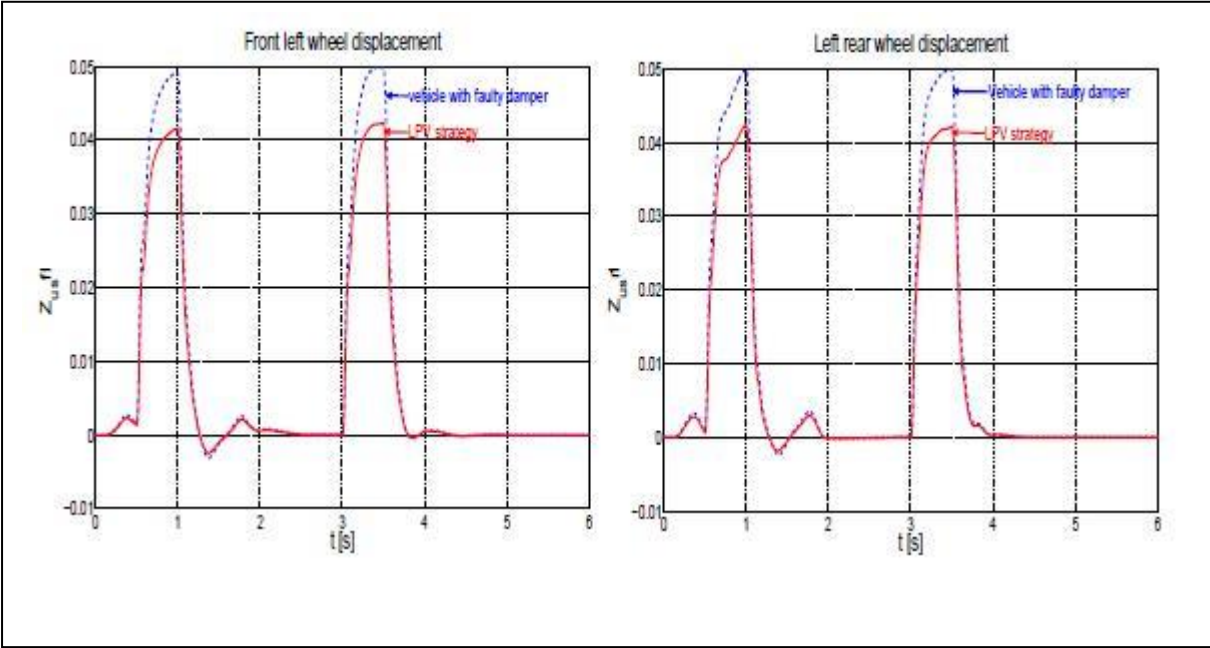


Fig.4. . Wheel displacement in front/rear left corners

In Fig. 3, 4, the four wheels bounce of the vehicle are shown. It can be seen also that the improvements brought by the designed controller on the left side are better than on the right side, due to the larger damping forces supplied on this side to handle the load transfer.

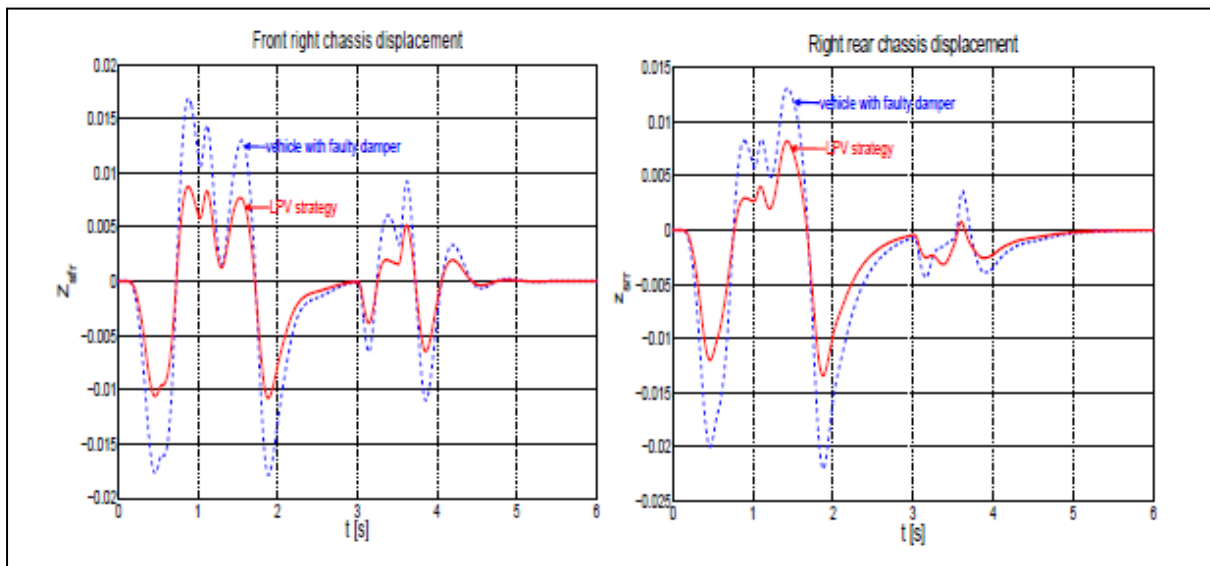


Fig.5. chassis displacement in front/rear right corners.

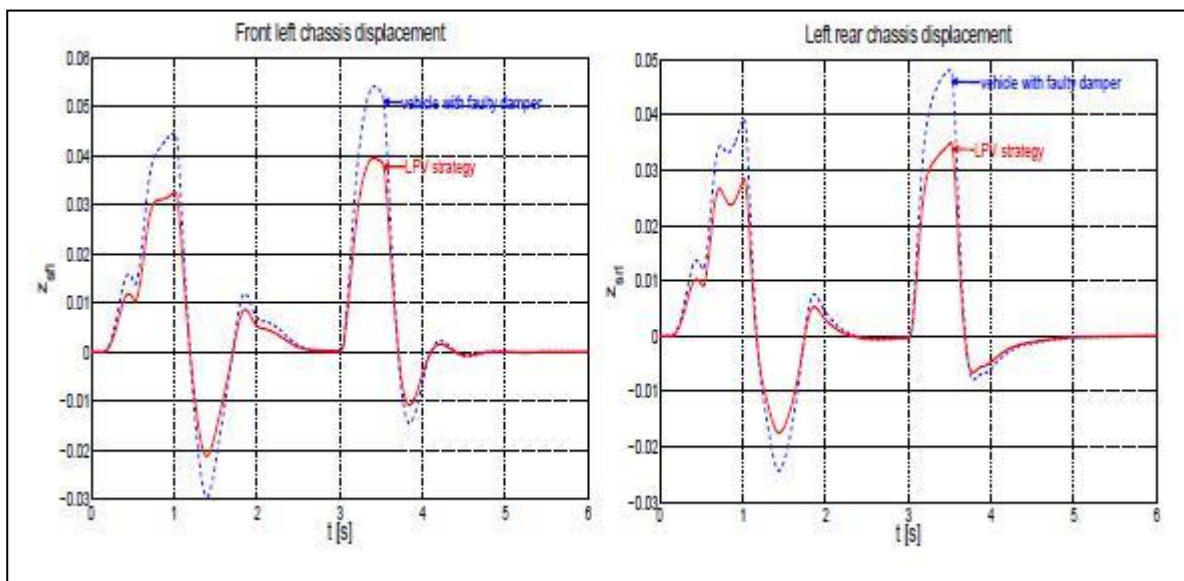


Fig.5. chassis displacement in front/rear left corners.

In Fig.5, 6, show the chassis displacement in each corner of the car. These plots allow to study the various comfort performances on each corner of the vehicle

It is clearly noticed that the performance objectives are differently reached, depending on the suspension forces distribution and reconfiguration given by the proposed LPV/ H^∞ fault tolerant control. This allows to handle the damper's failure effect on the vehicle dynamics

in several driving situations.

5. Conclusion

This paper has presented a new LPV/ H^∞ control strategy based on the load transfer monitoring. The monitoring strategy allows to provide the considered varying parameters that schedules the behaviors of the developed robust control depending on the driving situation. Simulations of a validated full vehicle model and considering a faulty actuators show the strength and efficiency of the proposed strategy trough the LPV framework.

Further works will be the implementation of the strategy on the **INOVE test bed** developed in Gipsa-lab, Grenoble, France, for vertical dynamics study.

