A LPV/Hinf fault tolerant control of vehicle roll dynamics under semi-active damper malfunction

Soheib Fergani, Olivier Sename, Luc Dugard

A LPV/$\mathcal{H}_\infty$ fault tolerant control of vehicle roll dynamics under semi-active damper malfunction

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***Abstract*— This paper proposes a LPV/$\mathcal{H}_\infty$ fault tolerant control strategy for roll dynamics handling under semi-active damper's malfunction. Indeed, in case of damper's malfunction, a lateral load transfer is generated, that amplifies the risks of vehicle roll over.**
In this study, the suspension systems efficiency is monitored through the lateral (or longitudinal) load transfer induced by a damper's malfunction.

The information given by the monitoring system is used in a partly fixed LPV/$\mathcal{H}_\infty$ controller structure that allows to manage the distribution of the four dampers forces in order to handle the over load caused by one damper's malfunction. The proposed LPV/$\mathcal{H}_\infty$ controller then uses the 3 remaining healthy semi-active dampers in a real time reconfiguration.

Moreover, the performances of the car vertical dynamics (roll, bounce, pitch) are adapted to the varying parameter given by the monitoring of the suspension system efficiency, which allows to modify online the damping properties (soft/hard) to limit the induced load transfer.

Simulations are performed on a complex nonlinear full vehicle model, equipped by 4 magneto-rheological semi-active dampers. This vehicle undergoes critical driving situations, and only one damper is considered faulty at ones. The simulation results show the reliability and the robustness of the proposed solution.

***Keywords*: LPV/$\mathcal{H}_\infty$ control, semi-active suspension, fault tolerant control, .**

I. 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). In this work, a specific type of semi-active suspension is under interest, namely, the Magneto Rheological Dampers (MRDampers, see [8]–[10]).

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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.
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.

To solve that problem a new LPV/$\mathcal{H}_\infty$ 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 load transfer induced b a damper malfunction is considered, but different methods could be integrated in the proposed control strategy (observers, parity space, ...). Then the global suspension control is scheduled according to the monitor parameter to adapt on-line the damper control distribution, and the performances of the suspension systems as well (in term of comfort and road holding).

To achieve these objectives, the authors have chosen to fix the structure of the LPV/$\mathcal{H}_\infty$ controller by making the LMI's orthogonal with parameters dependency, as follow:

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

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

$$U(\rho) = \begin{pmatrix} \rho_1 & 0 & 0 & 0 \\ 0 & \rho_2 & 0 & 0 \\ 0 & 0 & \rho_3 & 0 \\ 0 & 0 & 0 & \rho_4 \end{pmatrix} \tag{2}$$

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

Remark 1: This kind of structure has been used by the authors for vehicle dynamics control with braking, steering and suspension actuators [14], [12].

Here, this approach is extended to account for suspension actuator's malfunction. Since roll dynamics affect very much the vehicle behaviour, the authors have chosen to schedule the suspension control using the lateral load transfer as a varying parameter (ρ_l). The controller output matrix shows the dependency on this varying parameter and ensures the suspension efforts reconfiguration, as follows:

$$U(\rho_l) = \begin{pmatrix} 1-\rho_l & 0 & 0 & 0 \\ 0 & \rho_l & 0 & 0 \\ 0 & 0 & 1-\rho_l & 0 \\ 0 & 0 & 0 & \rho_l \end{pmatrix} \quad (3)$$

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/$\mathcal{H}_\infty$ fault tolerant control of vehicle roll 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:
Throughout the paper, the following notations will be adopted: indices $i = \{f, r\}$ and $j = \{l, r\}$ are used to identify vehicle front, rear and left, right positions respectively. Then, index $\{s, t\}$ holds for forces provided by suspensions and tires respectively. $\{x, y, z\}$ holds for forces and dynamics in the longitudinal, lateral and vertical axes respectively. Then let $v = \sqrt{v_x^2 + v_y^2}$ denote the vehicle speed, $R_{ij} = R - (z_{us_{ij}} - z_{r_{ij}})$ the effective tire radius, $m = m_s + m_{us_{fl}} + m_{us_{fr}} + m_{us_{rl}} + m_{us_{rr}}$ the total vehicle mass. The model parameters are those of a Renault Mégane Coupé, obtained during a collaborative study with the MIPS laboratory in Mulhouse, through identification with real data, see [14].

II. Full vehicle modeling

A. Full vehicle model

The model (4) used in this work is a nonlinear full vehicle model. Details of this model and the corresponding parameters can be found in [14]. It involves several car chassis dynamics: vertical (z_s), longitudinal (v_x), lateral (v_y), roll (θ), pitch (ϕ) and yaw (ψ). It also models the vertical and rotational motions of the wheels ($z_{us_{ij}}$ and ω_{ij} respectively), the slip ratios ($\lambda_{ij} = \frac{v_{ij} - R_{ij}\omega_{ij}\cos\beta_{ij}}{\max(v_{ij}, R_{ij}\omega_{ij}\cos\beta_{ij})}$) and the center of gravity side slip angle (β_{cog}) dynamics as a function of the tires and suspensions forces. The main dynamical equations are given in equation (4), where $F_{tx_i} = F_{tx_{il}} + F_{tx_{ir}}$, $F_{ty_i} = F_{ty_{il}} + F_{ty_{ir}}$, $F_{tz_i} = F_{tz_{il}} + F_{tz_{ir}}$ are the tire forces (based on Pacejka tire non linear model) and $F_{sz_i} = F_{sz_{il}} + F_{sz_{ir}}$, $(i = \{f, r\})$.

B. Vertical modeling

The model used for the controller synthesis is the linear vertical 7-DOF model. It includes several vertical dynamics as the chassis acceleration $\ddot{z}_s$, the four wheels accelerations $\ddot{z}_{us_{ij}}$, the roll bounce acceleration $\ddot{\theta}$ and the pitch acceleration $\ddot{\phi}$.

C. Semi-active Magneto-rheological damper

In this study, the proposed strategy is applied to a vehicle equipped by four semi-active MR dampers. There are various approaches to model semi-active dampers. In the parametric model of [15], the hysteresis loop force-velocity is well modeled by an hyperbolic tangent function.

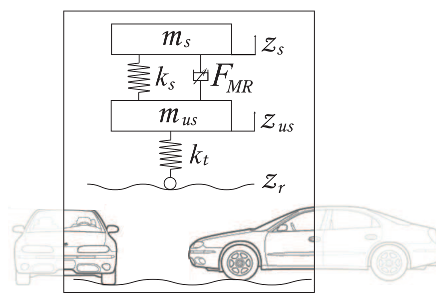

Fig. 1. *QoV* model for a semi-active suspension in a vehicle.

The *MR* damping force is given by:

$$F_{MR} = I f_c \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) + b_1 \dot{z}_{def} + b_2 z_{def} \quad (5)$$

where the electric current is bounded between $0 \leq I_{min} \leq I \leq I_{max} \leq 2.5$. I_{min} and I_{max} depend on the *MR* damper specifications. Experimental data obtained from a commercial *MR* damper are used to model the nonlinearities of this actuator by using (5). The parameters of the *MR* damper model used in this analysis are: $fc = 600.9$, $a_1 = 37.8$, $a_2 = 22.1$, $b_1 = 2830.8$ and $b_2 = -7897.2$.

The *QoV* system dynamics, given in a state-space repre-

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$$\left\{\begin{array}{rcl}\dot{v}_x & = & -\big(F_{tx_f}\cos(\delta)+F_{tx_r}+F_{ty_f}\sin(\delta)\big)/m-\dot{\psi}v_y\\ \dot{v}_y & = & \big(-F_{tx_f}\sin(\delta)+F_{ty_r}+F_{ty_f}\cos(\delta)\big)/m+\dot{\psi}v_x\\ \ddot{z}_s & = & -\big(F_{sz_f}+F_{sz_r}+F_{dz}\big)/m_s\\ \ddot{z}_{us_{ij}} & = & \big(F_{sz_{ij}}-F_{tz_{ij}}\big)/m_{us_{ij}}\\ \ddot{\theta} & = & \big((F_{sz_{rl}}-F_{sz_{rr}})t_r+(F_{sz_{fl}}-F_{sz_{fr}})t_f+mh\dot{v}_y\big)/I_x\\ \ddot{\phi} & = & \big(F_{sz_f}l_f-F_{sz_r}l_r-mh\dot{v}_x\big)/I_y\\ \ddot{\psi} & = & \big(l_f(-F_{tx_f}\sin(\delta)+F_{ty_f}\cos(\delta))-l_rF_{ty_r}+(F_{tx_{fr}}-F_{tx_{fl}})t_f\cos(\delta)-(F_{tx_{rr}}-F_{tx_{rl}})t_r+M_{dz}\big)/I_z\\ \dot{\omega}_{ij} & = & (R_{ij}F_{tx_{ij}}-T^f_{b_{ij}})/I_w\\ \dot{\beta}_{cog} & = & (F_{ty_f}+F_{ty_r})/(mv_x)+\dot{\psi}\end{array}\right. \tag{4}$$

sentation, is written as:

$$\underbrace{\begin{bmatrix}\dot{z}_s\\ \ddot{z}_s\\ \dot{z}_{us}\\ \ddot{z}_{us}\end{bmatrix}}_{\dot{x}} = \underbrace{\begin{bmatrix}0 & 1 & 0 & 0\\ -\frac{k_s+b_2}{m_s} & -\frac{b_1}{m_s} & \frac{k_s+b_2}{m_s} & \frac{b_1}{m_s}\\ 0 & 0 & 0 & 1\\ \frac{k_s+b_2}{m_{us}} & \frac{b_1}{m_{us}} & -\frac{k_s+k_t+b_2}{m_{us}} & -\frac{b_1}{m_{us}}\end{bmatrix}}_{A}\underbrace{\begin{bmatrix}z_s\\ \dot{z}_s\\ z_{us}\\ \dot{z}_{us}\end{bmatrix}}_{x} + \underbrace{\begin{bmatrix}0 & 0\\ \frac{-\cdot\rho f_c}{m_s} & 0\\ 0 & 0\\ \frac{\cdot\rho f_c}{m_{us}} & \frac{k_t}{m_{us}}\end{bmatrix}}_{B}\underbrace{\begin{bmatrix}I\\ z_r\end{bmatrix}}_{u}$$

$$\underbrace{\begin{bmatrix}y_1\\ y_2\end{bmatrix}}_{y} = \underbrace{\begin{bmatrix}1 & 0 & -1 & 0\\ 0 & 1 & 0 & -1\end{bmatrix}}_{C}\begin{bmatrix}z_s\\ \dot{z}_s\\ z_{us}\\ \dot{z}_{us}\end{bmatrix} \tag{6}$$

where, $\rho = \tanh\left[a_1\dot{z}_{def} + a_2 z_{def}\right] \in [0,1]$ is a varying parameter, the accelerometers of the sprung ($\ddot{z}_s$) and unsprung mass ($\ddot{z}_{us}$). These measurements are related to the comfort and road holding performances, that depend on the semi-active damper properties and obviously on the road irregularities.

III. Design of the LPV/$\mathcal{H}_\infty$ Fault Tolerant Control of Vehicle Roll Dynamics under Semi-Active Damper Malfunction

In this paper, a new LPV/$\mathcal{H}_\infty$ fault tolerant control strategy is based on the monitoring of the semi-active dampers. When a fault is detected on one of the four semi-active dampers (i.e a lack in the vertical forces), the roll dynamics are amplified, causing vehicle instability and increasing car roll-over risks. To manage this instability, the proposed LPV/$\mathcal{H}_\infty$ suspension control is scheduled thanks to ρ_l the load transfer generated by the roll bounce of the vehicle ρ_l (by comparing the righ/letf forces) and tunes the 3 remaining healthy dampers to achieve fault compensation without reaching saturation. In addition, the performance objectives are set thanks to this varying parameter ρ_l which is included in the considered weighting functions on chassis displacement W_{z_s} and the roll dynamics of the car W_θ.

Scheduling parameters:

This strategy given in Fig. 2 includes 3 varying parameters.

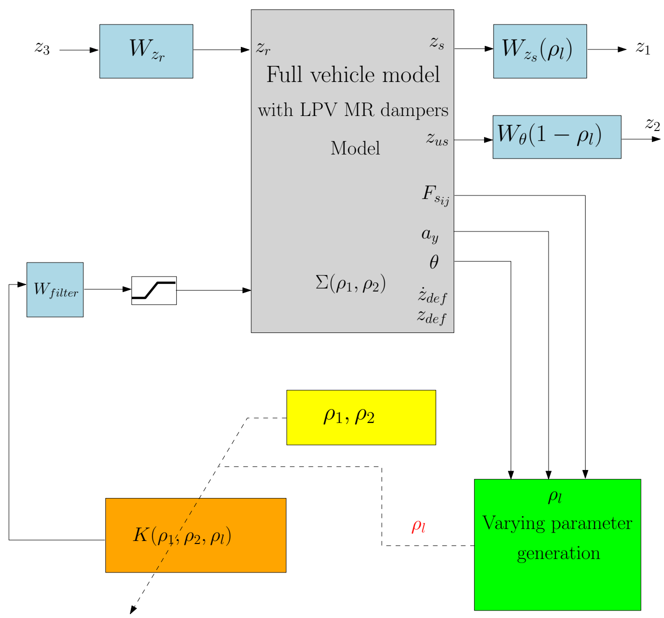

Fig. 2. Global chassis control implementation scheme.

one is used for the control reconfiguration and adaptation to critical driving situations with damper malfunction, the two others parameters are needed to account for the dissipativity and saturation of the semi-active MR , as in [16].

A. *LPV* QoV *model* ρ_1 *and* ρ_2*:*

Indeed, the semi-active force is modeled following [8], as:

$$F_{sa} = \underbrace{b_1\left(\dot{z}_{s_i}-\dot{z}_{us_i}\right)+b_2\left(z_{s_i}-z_{us_i}\right)}_{passive} + \underbrace{I\cdot f_c\cdot\rho}_{semi-active} \tag{7}$$

where I is the electric current to control the semi-active force based on the desired performances and $\rho = \tanh\left[a_1\dot{z}_{def}+a_2 z_{def}\right] \in [0,1]$ represents the nonlinearities of the shock absorber. In the control synthesis for FTC, the varying parameters ρ_1 and ρ_2 allow to ensure that the suspension control meets the semi-activeness and the saturation damper's constraints, respectively. Then, in this paper the suspension in each corner is modeled as:

$$\left\{\begin{array}{l}\dot{x}_{lpv} = A_{lpv}\left(\rho_1,\rho_2\right)x_{lpv}+B_1u_c+B_2w\\ y_{lpv} = C_1x_{lpv}\end{array}\right. \tag{8}$$

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where

$$x_{lpv} = \begin{pmatrix} x_s \\ x_f \end{pmatrix}^T,$$
$$A_{lpv}(\rho_1,\rho_2) = \begin{pmatrix} A_s+\rho_2 B_{s2}C_{s2} & \rho_1 B_s C_f \\ 0 & A_f \end{pmatrix},$$
$$B_1 = \begin{pmatrix} 0 \\ B_f \end{pmatrix}, B_2 = \begin{pmatrix} B_{s1} \\ 0 \end{pmatrix}, C_1 = \begin{pmatrix} C_s \\ 0 \end{pmatrix}^T$$

$$\rho_1 = \tanh(C_{s2}x_s)\tanh(\frac{C_f x_f}{F_1})\frac{F_1}{C_f x_f},$$
$$\rho_2 = \frac{\tanh(C_{s2}x_s)}{C_{s2}x_s}$$

x_s, A_s, B_s, B_{s1}, B_{s2}, C_s and C_{s2} are the state and matrices of a state-space representation of the *QoV* model by including the *MR* damper model in (6) and considering z_{def} and $\dot{z}_{def}$ as output; x_f, A_f, B_f, C_f are the state and matrices of a representation of the low-pass filter $W_{filter} = \omega_f/(s+\omega_f)$ which is added to the system to make the control input matrices parameter independent.

B. LPV controller structure scheduled by ρ_l:

The third scheduling parameter, ρ_l, acts in the presence of damper malfunction, which can be seen directly on the lateral load transfer of the vehicle. This parameter, defined as follows, allows the right/left suspension control reconfiguration:

$$\left\{ \begin{array}{rcl} F_{z_l} & = & m_s \times g/2 + m_s \times h \times a_y/l_f \\ F_{z_r} & = & m_s \times g/2 - m_s \times h \times a_y/l_r \\ \rho_l & = & |(\delta_{fl}F_{z_{fl}} + \delta_{rl}F_{z_{rl}}) - (\delta_{fr}F_{z_{fr}} + \delta_{rr}F_{z_{rr}})| \\ & & /|(F_{z_{fl}} + F_{z_{rl}} + F_{z_{fr}} + F_{z_{rr}})|; \end{array} \right. \quad (9)$$

with δ_{ij}: the suspension systems efficiency given by the considered monitoring system, $F_{z_{ij}}$: the vertical forces, a_y lateral acceleration, $\rho_l \in [0 \quad 1]$: the monitoring parameter. The innovative solution which aims at stabilizing the vehicle in the presence of damper failure is the following: the controller has a partly fixed structure obtained by by making the LMIs structure orthogonal with a parameter dependency on the control output matrix, as follow:

$$\begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho_l)C_c^0(\rho_l)}_{C_c(\rho_l)} x_c(t) \quad (10)$$

The suspension forces distribution is obtained with the matrix $U(\rho_l)$:

$$U(\rho_l) = \begin{pmatrix} 1-\rho_l & 0 & 0 & 0 \\ 0 & \rho_l & 0 & 0 \\ 0 & 0 & 1-\rho_l & 0 \\ 0 & 0 & 0 & \rho_l \end{pmatrix} \quad (11)$$

The parameter ρ_l defined in (9) generates the adequate suspension forces distribution, depending on the load transfer (left $\leftrightarrows$ right) caused by the critical situation.

This suspension tuning is achieved as follows: When one of the suspension dampers is faulty, a load transfer is then generated and influences the vehicle stability and handling. When a malfunction is detected on one of the left front suspension systems, $\rho_l \to 1$, penalizing the provided output suspension force on the faulty corner, changing the level of saturation depending on the detected fault. Also, an overload appears on the right side. To managed that, the lacking suspension effort is compensated by the 3 healthy dampers to stabilise the vehicle. Indeed, left suspensions are set to "hard" to handle the overload caused by the loss of one of the right side dampers. On the other side, suspensions are relaxed and tuned to "soft" for the remaining healthy actuators (since the overload is on the other side) and a level of saturation is applied to the faulty one depending of the degree of deterioration detected. This distribution is handled thanks to the specific structure of the suspension controller, given as follows :

$$K_s(\rho) := \left\{ \begin{array}{l} \dot{x}_c(t) = A_c(\rho_1,\rho_2,\rho_l)x_c(t) + B_c(\rho_1,\rho_2,\rho_l)y(t) \\ \begin{pmatrix} u_{fl}^{\mathcal{H}_\infty}(t) \\ u_{fr}^{\mathcal{H}_\infty}(t) \\ u_{rl}^{\mathcal{H}_\infty}(t) \\ u_{rr}^{\mathcal{H}_\infty}(t) \end{pmatrix} = \underbrace{U(\rho_l)C_c^0(\rho_1,\rho_2)}_{C_c(\rho_1,\rho_2)} x_c(t) \end{array} \right. \quad (12)$$

where $x_c(t)$ is the controller state, $A_c(\rho_1,\rho_2,\rho_l)$, $B_c(\rho_1,\rho_2,\rho_l)$ and $C_c(\rho_1,\rho_2,\rho_l)$ controller scheduled by ρ_l while ρ_1 and ρ_2 ensure the semi-activeness of the dampers. $u^{\mathcal{H}_\infty}(t) = [u_{fl}^{\mathcal{H}_\infty}(t)u_{fr}^{\mathcal{H}_\infty}(t)u_{rl}^{\mathcal{H}_\infty}(t)u_{rr}^{\mathcal{H}_\infty}(t)]$ the input control of the suspension actuators and $y(t) = z_{def}(t)$.

C. The suspension control problem formulation

In this study, a 7 DOF vehicle model is considered, (see (II-B)and augmented with LPV damper model (7) for each corner of the vehicle.

The suspension control with performance adaptation (see [17]) is presented. The following H_∞ control scheme is considered, including parameter varying weighting functions. where $W_{z_s} = \rho_l \frac{s^2+2\xi_{11}\Omega_{11}s+\Omega_{11}^2}{s^2+2\xi_{12}\Omega_{12}s+\Omega_{12}^2}$ is shaped in order to reduce the bounce amplification of the suspended mass (z_s) between $[0, 12]$Hz.

$W_\theta = (1-\rho_l)\frac{s^2+2\xi_{21}\Omega_{21}s+\Omega_{21}^2}{s^2+2\xi_{22}\Omega_{22}s+\Omega_{22}^2}$ attenuates the roll bounce amplification in low frequencies.

$W_u = 3.10^{-2}$ shapes the control signal.

Remark 3.1: The parameters of these weighting functions are obtained using genetic algorithm optimization as in [16].

According to Fig. 2, the following parameter dependent suspension generalized plant ($\Sigma_{gv}(\rho_1,\rho_2,\rho_l)$) is obtained:

$$\Sigma_{gv}(\rho_1,\rho_2,\rho_l) := \left\{ \begin{array}{l} \dot{\xi} = A(\rho_1,\rho_2,\rho_l)\xi + B_1\tilde{w} + B_2 u \\ \tilde{z} = C_1(\rho_1,\rho_2,\rho_l)\xi + D_{11}\tilde{w} + D_{12}u \\ y = C_2\xi + D_{21}\tilde{w} + D_{22}u \end{array} \right. \quad (13)$$

where $\xi = [\chi_{vert} \quad \chi_w]^T$; $\tilde{z} = [z_1 \quad z_2 \quad z_3]^T$; $\tilde{w} = [z_{rij} \quad F_{dx,y,z} \quad M_{dx,y}]^T$; $y = z_{def_{ij}}$; $u = u_{ij}^{\mathcal{H}_\infty}$; and χ_w are the vertical weighting functions states.

One of the main interesting contributions is the use of the parameter ρ_l that schedules the distribution of the left &

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right suspensions on the four corners of the vehicle and tune the suspension dampers smoothly. This is done thanks to the LPV frame work, from "soft" to "hard" to improve the car performances according to the driving situation.

In this synthesis, the authors wish to stress that a very interesting innovation is the use of a partly fixed structure controller with a parameter dependency (ρ_l) on the control output matrix, combined with the scheduling of the weighting functions by the use of the varying parameter ρ_l, on the chassis displacement (z_s, considered as a comfort indicator) and the roll motion (θ, a road holding indicator). This allows to tune various actuators controllers, depending on the driving situation, by a hierarchical activation to optimize their use (coordinate framework with smooth transition between different performance objectives even if they are contradictory).

The LPV system (13) includes 3 scheduling parameters and can be described as a polytopic system, i.e, a convex combination of the systems defined at each vertex of a polytope defined by the bounds of the varying parameter. The synthesis of the controller is made within the framework of the $\mathcal{H}_\infty$ control of polytopic suspensions, (for more details, see [18]).

Remark 2: All controllers presented along the paper are synthesized in the LPV/H_∞ framework. This design is achieved, thanks to the LMI-based H_∞ resolution.

IV. Simulation Results

Time domain simulations are performed on the full nonlinear vehicle model given in Section II-A. For sake of completeness, the results of the proposed LPV/$\mathcal{H}_\infty$ 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/$\mathcal{H}_\infty$ FTC of vehicle roll dynamics under semi-active damper malfunction, the following scenario is used:

1) The vehicle runs at 80km/h in straight line on wet road ($\mu = 0.5$, where μ is a coefficient representing the adherence to the road).
2) The front right damper of the vehicle is considered faulty (a failure of 70% on the nominal behaviour of the healthy dampers).
3) A 5cm bump on the left wheels (from $t = 0.5$s to $t = 1$s),
4) A Another bump on the right wheels (from $t = 3$s to $t = 4$s),

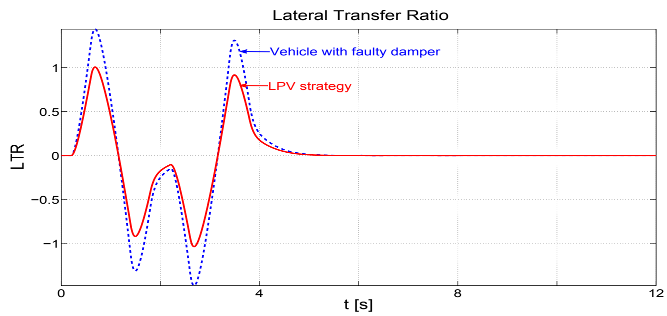

Fig. 3. Lateral load transfer

Fig. 3 shows the lateral load transfer generated by the driving scenario; based on it, the scheduling parameter ρ_l is calculated.

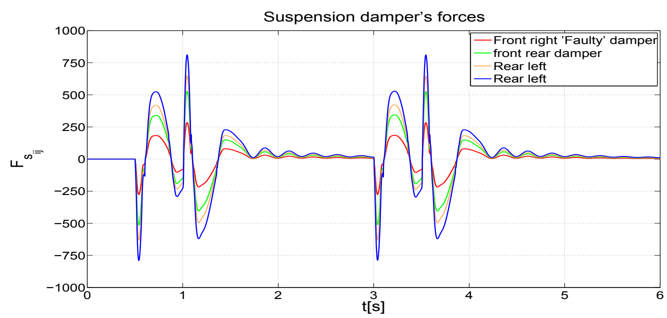

Fig. 4. Suspension damper's forces: the faulty and healthy dampers efforts

In Fig. 4, the 4 semi-active dampers efforts provided by the designed fault tolerant LPV/$\mathcal{H}_\infty$ 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 right 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.

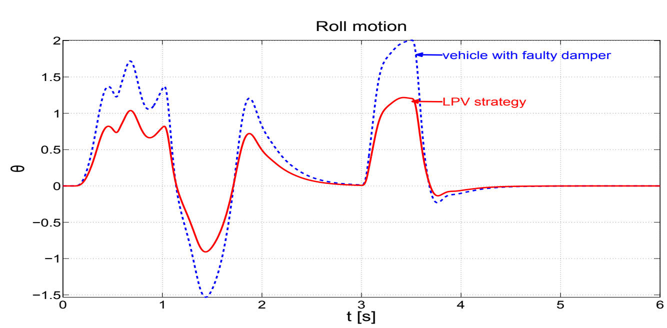

Fig. 5. Roll motion of the vehicle θ

Fig. 5 represents one of the main results of the paper. The roll dynamics are clearly attenuated by the proposed

LPV/$\mathcal{H}_\infty$ FTC strategy. This allows to maintain a good road holding and stability of the vehicle.

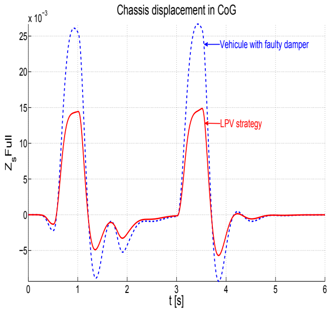

Fig. 6. Chassis displacement in CoG z_s.

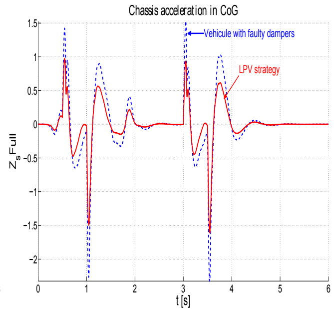

Fig. 7. Chassis acceleration in CoG $\ddot{z}_s$.

From Fig. 6 and Fig. 7, it can be noticed that the developed strategy in addition on enhancing vehicle roadholding, it improves passengers comfort by reducing chassis acceleration $\ddot{z}_s$ and displacement z_s while driving.

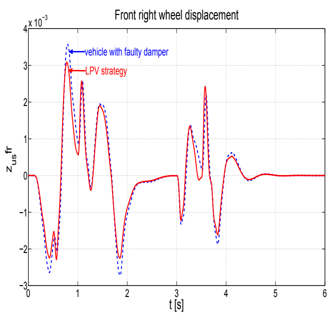

Fig. 8. Wheel displacement in front right $z_{us_{fr}}$.

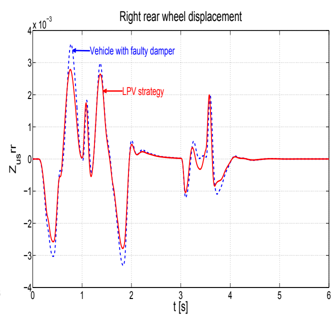

Fig. 9. Wheel displacement in rear right $z_{us_{rr}}$.

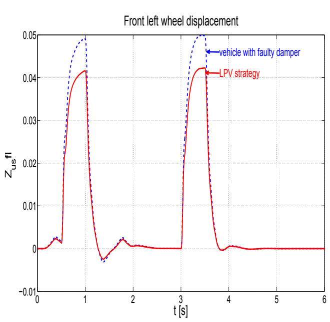

Fig. 10. Wheel displacement in front left $z_{us_{fl}}$.

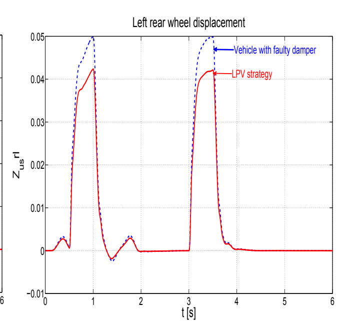

Fig. 11. Wheel displacement in rear left $z_{us_{rl}}$.

In Fig. 8, 9, 10, 11, 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.

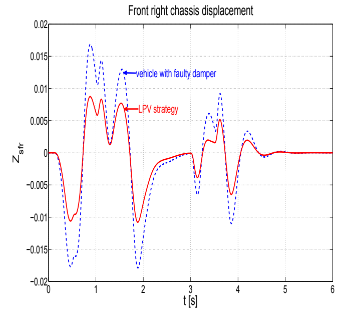

Fig. 12. Chassis displacement in front right $z_{s_{fr}}$.

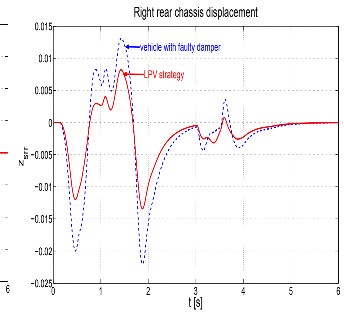

Fig. 13. Chassis displacement in rear right $z_{s_{rr}}$.

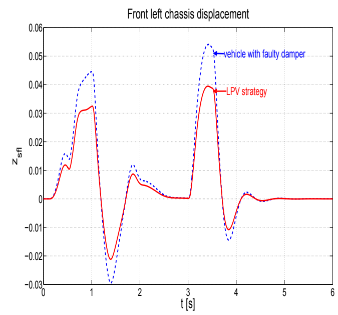

Fig. 14. Chassis displacement in front left $z_{s_{fl}}$.

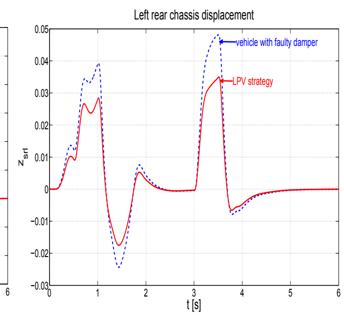

Fig. 15. Chassis displacement in rear left $z_{s_{rl}}$.

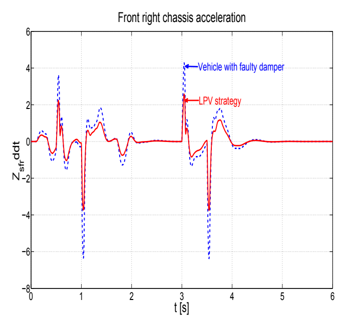

Fig. 16. Chassis acceleration in front right $\ddot{z}_{s_{fr}}$.

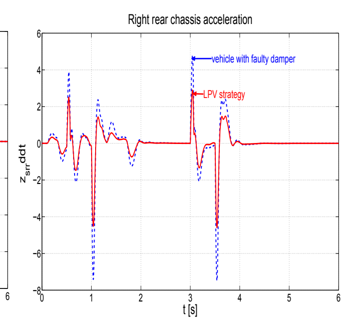

Fig. 17. Chassis acceleration in rear right $\ddot{z}_{s_{rr}}$.

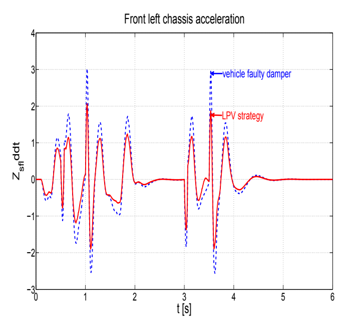

Fig. 18. Chassis acceleration in front left $\ddot{z}_{s_{fl}}$.

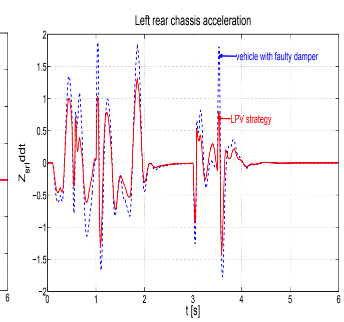

Fig. 19. Chassis displacement in rear left $\ddot{z}_{s_{rl}}$.

Figures from Fig. 12 to Fig. 19 show various comfort performances on each corner of the vehicle (chassis displacement, acceleration, resp). It is clearly noticed that the

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performance objectives are differently reached, depending on the suspension forces distribution and reconfiguration given by the proposed LPV/$\mathcal{H}_\infty$ fault tolerant control. This allows to handle the damper's failure effect on the vehicle dynamics in several driving situations.

V. Conclusion

This paper has presented a new LPV/H_∞ fault tolerant control strategy which handles vehicle roll dynamics under damper malfunction. It proposes a new structure of the controller, by making the corresponding LMIs orthogonal with a parameter dependency on the controller matrix output. The varying parameter used in the developed strategy is obtained by monitoring the lateral transfer ratio caused by the roll bounce of the vehicle. This allows to online reconfigure the provided suspensions forces in the four corners of the vehicle to reach the desired performance objective. Simulations performed on a complex nonlinear model have shown the efficiency of the proposed approach.
The authors stress that using the LPV framework allows to simplify the implementation procedure. The next step of this work is being started with the implementation of this strategy on a test benchmark, available at Gipsa-lab in Grenoble, developed in collaboration with a high-technology start up "SOBEN". It consists of vehicle equipped with four semi-active Electro-Rheological dampers. Different road profile could be generated separately on each wheel and online control can be implemented.

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