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Fault Tolerant Control with Additive Compensation for Faults in an Automotive Damper ^{*}

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Abstract—A novel *Fault-Tolerant Controller* is proposed for an automotive suspension system based on a *Quarter of Vehicle (QoV)* model. The design is divided in a robust *Linear Parameter-Varying* controller used to isolate vibrations from external disturbances and in a compensation mechanism used to accommodate actuator faults. The compensation mechanism is based on a robust fault detection and estimation scheme that reconstructs a fault on the semi-active damper; this information is used to reduce the failure effect into the vertical dynamics to achieve good control performances. Validations have been made over a *QoV* model in CarSimTM. Results show the effectiveness of the fault-tolerant semi-active damper versus an uncontrolled damper; the improvement is 50.4% in comfort and 42.4% in road holding, by avoiding biases in the damper deflection.

I. INTRODUCTION

Because advanced technological processes increase constantly their complexity, the control community has been developing novel techniques of modeling and control with fault-tolerance to improve the process reliability. A Fault-Tolerant Controller (*FTC*) is designed to maintain the desired performance of the system when occur process failures; there are two major groups: passive (when the fault tolerance is designed off-line) and active (based on an automatic control reconfiguration mechanism).

The importance of using an active system is to design a Fault Detection and Isolation (*FDI*) strategy. The *FDI* module estimates the fault on-line, which signal is used into the controller synthesis in order to accommodate the malfunction; in contrast, a passive version can result very conservative when different fault scenarios exist.

A reconfiguration mechanism of compensation is one of the different ways to accommodate the fault [1]; for the fault estimation, *FDI* modules based on analytical redundancy are the most accepted frameworks, in particular parity space approaches for Linear-Time Invariant (*LTI*) systems.

During the last years, the modeling of Linear Parameter-Varying (*LPV*) systems has gained importance as a solution to include the non-linear dynamics into the control law. Recently, *LPV* based techniques have been extended to *FTC* approaches. For instance, in [2] it is proposed an *FTC* for polytopic *LPV* systems by considering multiple failures.

Particularly in automotive control applications, some researches on *FTC* are designed for active suspension systems: in [3] an *FTC* based on sliding mode observers is proposed, while in [4] a fault tolerant *LPV* control is designed to guarantee road holding and roll stability; both *FTC* approaches are passive with good robustness. For semi-active suspension systems, a control strategy under different faulty schemes in a Quarter of Vehicle (*QoV*) model is proposed in [5], an on-line parametric estimation is used to create a fault signature by using parity relations; however, the fault is not estimated.

This paper proposes an active *FTC* for a semi-active suspension system, by analyzing a *QoV* model that includes an experimental Magneto-Rheological (*MR*) damper model. The *FTC* structure is composed by a robust *LPV* controller and a reconfiguration mechanism based on a robust fault detector. Based on \mathcal{H}_∞ control theory, the *LPV* controller is robust to road disturbances and oriented to comfort and road holding; while, the proposed *FDI* approach is an adaptation of the classical parity-space theory for the considered uncertain system, i.e. with un-modeled dynamics. The estimated fault obtained by the *FDI* module is used in a reconfiguration mechanism to compensate the fault effect in the vertical dynamics of the vehicle. Simulation results in CarSimTM shows the effectiveness of the proposed approach.

TABLE I. DEFINITION OF VARIABLES.

Variable	Description
ρ_1	Varying parameter to represent hysteresis of the damper
ρ_1^*	Varying parameter to represent hysteresis of the damper by using a filter in the control input
ρ_2^*	Varying parameter to represent saturation of actuation
a_i	Pre-yield viscous damping coefficients in <i>MR</i> damper
b_i	Post-yield viscous damping coefficients in <i>MR</i> damper
f_c	Dynamic yield force in the <i>MR</i> damper model
F_{MR}	<i>MR</i> damper force
F_o	Additive actuator fault
I, I_0	Electric current, Mean of I
I_{LPV}	Electric current derived from the <i>LPV</i> controller
I_{FTC}	Electric current derived from the <i>FTC</i>
I_{FDI}	Electric current derived from the <i>FDI</i>
\mathcal{K}_{LPV}	Polytopic <i>LPV</i> controller structure
k_s, k_t	Spring & wheel stiffness coefficient
m_s, m_{us}	Sprung & unsprung mass in the <i>QoV</i>
r, \bar{r}	Parity space residual
z_{def}	Damper piston position
\dot{z}_{def}	Damper piston velocity
z_r	Road profile
z_s, z_{us}	Vertical position of m_s, m_{us}
\dot{z}_s, \dot{z}_{us}	Vertical velocity of m_s, m_{us}
$\ddot{z}_s, \ddot{z}_{us}$	Vertical acceleration of m_s, m_{us}

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The paper is organized as follows: the problem statement is described in the next section. Sections III and IV describe the methodology of fault detection and estimation and the design of the *FTC*, respectively. Section V presents the simulation results. Finally, conclusions are presented in section VI. All variables are defined in Table I.

II. PROBLEM STATEMENT

A classical *QoV* model of a pick-up truck is used as test-bed. An experimental *MR* damper model represents the semi-active suspension between the sprung and unsprung masses, given by:

$$\begin{aligned} m_s \ddot{z}_s &= -k_s(z_s - z_{us}) - F_{MR} \\ m_{us} \ddot{z}_{us} &= k_s(z_s - z_{us}) - k_t(z_{us} - z_r) + F_{MR} \end{aligned} \quad (1)$$

The semi-active damping force used for this approach is an extension of [6], but with inclusion of the manipulation signal (electric current). Its non-linear dynamics can be modeled by the damping force F_{MR} as:

$$F_{MR}(t) = I f_c \rho_1(t) + b_1 \dot{z}_{def}(t) + b_2 z_{def}(t) + F_o(t) \quad (2)$$

where the non-linear part $\rho_1(t)$ is given by:

$$\rho_1(t) = \tanh(a_1 \dot{z}_{def} + a_2 z_{def})$$

By using (1) and (2), a state-space representation of the *QoV* model can be obtained as:

$$\begin{aligned} \underbrace{\begin{bmatrix} \ddot{z}_s \\ \dot{z}_s \\ \ddot{z}_{us} \\ \dot{z}_{us} \end{bmatrix}}_x &= \underbrace{\begin{bmatrix} -\frac{b_1}{m_s} & -\frac{c_1}{m_s} & \frac{b_1}{m_s} & \frac{c_1}{m_s} \\ 1 & 0 & 0 & 0 \\ \frac{b_1}{m_{us}} & \frac{c_1}{m_{us}} & -\frac{b_1}{m_{us}} & -\frac{c_1+k_t}{m_{us}} \\ 0 & 0 & 1 & 0 \end{bmatrix}}_A \underbrace{\begin{bmatrix} \dot{z}_s \\ z_s \\ \dot{z}_{us} \\ z_{us} \end{bmatrix}}_x \\ &+ \underbrace{\begin{bmatrix} -\frac{f_c \rho_1}{m_s} \\ 0 \\ \frac{f_c \rho_1}{m_{us}} \\ 0 \end{bmatrix}}_{B_I} I + \underbrace{\begin{bmatrix} -\frac{1}{m_s} \\ 0 \\ \frac{1}{m_{us}} \\ 0 \end{bmatrix}}_{B_F} F_o + \underbrace{\begin{bmatrix} 0 \\ 0 \\ \frac{k_t}{m_{us}} \\ 0 \end{bmatrix}}_{B_r} z_r \\ \underbrace{\begin{bmatrix} \ddot{z}_s \\ \ddot{z}_{us} \\ z_{def} \end{bmatrix}}_y &= \underbrace{\begin{bmatrix} -\frac{b_1}{m_s} & -\frac{c_1}{m_s} & \frac{b_1}{m_s} & \frac{c_1}{m_s} \\ \frac{b_1}{m_{us}} & \frac{c_1}{m_{us}} & -\frac{b_1}{m_{us}} & -\frac{c_1+k_t}{m_{us}} \\ 0 & 1 & 0 & -1 \end{bmatrix}}_C \underbrace{\begin{bmatrix} \dot{z}_s \\ z_s \\ \dot{z}_{us} \\ z_{us} \end{bmatrix}}_x \\ &+ \underbrace{\begin{bmatrix} -\frac{f_c \rho_1}{m_s} \\ \frac{f_c \rho_1}{m_{us}} \\ 0 \end{bmatrix}}_{D_I} I + \underbrace{\begin{bmatrix} -\frac{1}{m_s} \\ \frac{1}{m_{us}} \\ 0 \end{bmatrix}}_{D_F} F_o + \underbrace{\begin{bmatrix} 0 \\ \frac{k_t}{m_{us}} \\ 0 \end{bmatrix}}_{D_r} z_r \end{aligned} \quad (3)$$

A. Problem definition

Design an active *FTC* for the semi-active suspension system of a *QoV* model capable to mitigate the effects of an additive fault in the *MR* damper. The *FDI* module aims to compensate the malfunction of the *MR* damper by adding a compensatory signal to the *LPV* controller. Both approaches (*LPV* controller and *FDI* module) must be designed to be robust to road disturbances and model uncertainties, and include

the nonlinearities of the semi-active damper into its design, such that:

$$\begin{aligned} \dot{x} &= \mathcal{A}(\rho) \cdot x + \mathcal{B} \cdot [I_{FTC} F_o z_r]^T \\ y &= \mathcal{C}(\rho) \cdot x + \mathcal{D} \cdot [I_{FTC} F_o z_r]^T \\ I_{FTC} &= \mathcal{K}_{LPV}(\rho) \cdot x - I_{FDI}(\hat{F}_o) \end{aligned} \quad (4)$$

with $\mathcal{K}_{LPV}(\rho) = \sum_{i=1}^N \xi_i(\rho) \mathcal{K}_i$ by appropriately choosing the gains \mathcal{K}_i , $i = 1, \dots, N$ such that the closed-loop system (4) be asymptotically stable in all parameter variations, and I_{FDI} depends on the estimated fault based on a nonlinear relation force - electric current.

III. FAULT DETECTION AND ESTIMATION

The fault detection scheme is based on the parity-space approach, because of its applicability to actuator faults in nonlinear systems, speed of detection, isolability property, robustness and computational complexity [1]. The aim is to synthesize a residual which is sensitive to the fault F_o and insensitive to bounded exogenous inputs, e.g. the road profile or load transfers on the vehicle.

The outputs $y(t)$ are expressed along the horizon s of the time derivatives as:

$$\underbrace{\begin{bmatrix} y \\ \dot{y} \\ \ddot{y} \\ \vdots \\ y^{(s)} \end{bmatrix}}_y - \mathcal{G}_I \underbrace{\begin{bmatrix} I \\ \dot{I} \\ \ddot{I} \\ \vdots \\ I^{(s)} \end{bmatrix}}_{\mathcal{I}} = \mathcal{H}x + \mathcal{G}_F \underbrace{\begin{bmatrix} F_o \\ \dot{F}_o \\ \ddot{F}_o \\ \vdots \\ F_o^{(s)} \end{bmatrix}}_{\mathcal{F}} + \mathcal{G}_r \underbrace{\begin{bmatrix} z_r \\ \dot{z}_r \\ \ddot{z}_r \\ \vdots \\ z_r^{(s)} \end{bmatrix}}_{\mathcal{Z}} \quad (5)$$

where

$$\mathcal{G}_x = \begin{bmatrix} D_x & 0 & \cdots & \cdots & \cdots & 0 \\ CB_x & D_x & 0 & \cdots & \cdots & 0 \\ CAB_x & CB_x & \ddots & \ddots & \ddots & \vdots \\ CA^2 B_x & CAB_x & \ddots & \ddots & \ddots & \vdots \\ \vdots & \ddots & \ddots & \ddots & \ddots & 0 \\ CA^{s-2} B_x & CA^{s-3} B_x & \cdots & CAB_x & CB_x & D_x \end{bmatrix}$$

$$\text{and } \mathcal{H} = \begin{bmatrix} C \\ CA \\ CA^2 \\ \vdots \\ CA^{s-1} \end{bmatrix} \text{ ensures the observability condition.}$$

To fulfill the requirement of sensitivity to the fault F_o and insensitivity to road disturbances, the parity-matrix \mathcal{W} is:

$$\mathcal{W} \cdot [\mathcal{H} \quad \mathcal{G}_r] = 0 \quad (6)$$

Equation (6) guarantees the perfect decoupling of residuals from the states x of the system and from the road disturbances z_r . The expression of the residuals is:

$$r(t) = \mathcal{W}(\mathcal{Y} - \mathcal{G}_I \mathcal{I}) = \mathcal{W} \mathcal{G}_F F_o(t) \quad (7)$$

where residuals $r(t)$ are perfectly sensitive to $F_o(t)$. By computing the null-space matrix \mathcal{W} it will be possible to detect any additive fault in the damper without considering the road disturbances or the manipulation variable.

The filter design (A_f, B_f, C_f, D_f) and LPV modeling with constraints in the actuator can be reviewed in [8], [9].

The polytopic approach for LPV control design bounds the nonlinearities of a closed-loop system into a finite dimension problem. A polytope of 4 LTI controllers is designed by using \mathcal{H}_∞ control theory; the controller is quadratically stable for all trajectories of the varying parameters by using LMI techniques, [10]. The weighting functions are designed to reduce the vertical acceleration of the sprung mass (comfort) and the displacement of the unsprung mass (road holding):

$$W_{\dot{z}_s} = \frac{K_s(s^2 + 2\zeta_{1s}\omega_{1s}s + \omega_{1s}^2)}{s^2 + 2\zeta_{2s}\omega_{2s}s + \omega_{2s}^2}, \quad W_{z_{us}} = \frac{K_{us}(s + \omega_{1us})}{s + \omega_{2us}}$$

Taking into account the control specifications, the generalized system \mathcal{P} for the LPV control synthesis is:

$$\begin{aligned} \dot{x}_p &= A_p(\rho_1^*, \rho_2^*) \cdot x_p + B_{p1} \cdot u_1 + B_{p2} \cdot u_2 \\ z &= C_z(\rho_1^*, \rho_2^*) \cdot x_p \cdot \begin{bmatrix} W_{\dot{z}_s} & 0 \\ 0 & W_{z_{us}} \end{bmatrix} \\ y &= C_p \cdot x_p + D_{p1} \cdot u_1 + D_{p2} \cdot u_2 \end{aligned} \quad (19)$$

where $C_z(\rho_1^*, \rho_2^*) = [C_{z1} \quad \rho_1^* D_{z1} C_f]$, $D_{z1} = \begin{bmatrix} -\frac{fc}{m_s} \\ 0 \end{bmatrix}$ and,

$$C_{z1} = \begin{bmatrix} -\frac{b_1}{m_s} & -\frac{c_1}{m_s} & \frac{b_1}{m_s} & \frac{c_1}{m_s} \\ 0 & 0 & 1 & 0 \end{bmatrix} + \begin{bmatrix} -\frac{\rho_2^* I_0}{m_s} C_{I0} \\ 0_{1 \times 4} \end{bmatrix}$$

The reconfiguration of compensation uses the estimated fault \hat{F}_o for computing the required electric current to correct the malfunction. This non-linear relation is given by the inverse dynamics of the MR damper model, eq. (2), such that:

$$I_{FDI} = I_0 \cdot \tanh\left(\frac{\hat{F}_o - b_{10}\dot{z}_{def} - b_{20}z_{def}}{f_0 \cdot \rho_1}\right) \in [-I_0, I_0] \quad (20)$$

where, the \tanh function bounds the non-linear relation $I - F_{MR}$ around the average of the electric current (I_0), by considering $fc = f_0$, $b_1 = b_{10}$ and $b_2 = b_{20}$ from an experimental test at passive damping conditions.

Equation (20) represents the electric current required to reduce the fault propagation into the vehicle dynamics; because this fault is additive, I_{FDI} must subtract the effect of the estimated fault \hat{F}_o , as a compensation, in the closed-loop of (3). Because the actuation over the damper is saturated, the magnitude of the fault affects directly the bandwidth of control. Thus, the fault-tolerant controller output is composed by the robust LPV controller output (I_{LPV}) and the compensation (I_{FDI}) as:

$$I_{FTC} = I_{LPV} - I_{FDI} \quad (21)$$

such that $B_F \cdot F_o - B_I \cdot I_{FDI} \sim 0$ in the closed-loop system in order to eliminate the fault effects. The resulting closed-loop dynamics is given by:

$$\dot{x} = Ax + B_I(\mathcal{K}_{LPV}x - I_{FDI}) + B_F F_o + B_r z_r \quad (22)$$

where, \mathcal{K}_{LPV} is the polytopic LPV controller obtained by the LMI solution.

V. RESULTS AND DISCUSSION

The considered suspension system (test-bed) is obtained from a customized pick-up truck model by using a $K\&C$ test, which dynamics is generated via CarsimTM. The analysis is carried on the front-left corner of the vehicle, whose QoV model parameters are: $m_s = 470 \text{ Kg}$, $m_{us} = 110 \text{ Kg}$, $k_s = 86,378 \text{ N.m}^{-1}$ and $k_t = 270,000 \text{ N.m}^{-1}$. For damper modeling, experimental data were identified at two different electric current values from a standardized test (road elevation *chirp*), the model parameters are: $fc = 600.95$, $a_1 = 37.85$, $a_2 = 22.15$, $b_1 = 2,830.86$ and $b_2 = -7,897.21$.

The vehicle is moving straight forward at constant speed $v = 30 \text{ km/h}$. The road profile is composed of two successive bumps of 0.1 m height at $t = 0.6 \text{ s}$ and $t = 3.7 \text{ s}$, which are considered as disturbances for the FDI module. The fault occurs between both bumps, at $t = 1 \text{ s}$. The fault is a negative force appearing on the damper: $F_0 = -4000 \text{ N}$, i.e. around 40% of the operation span in the rebound effect. This kind of fault could be caused by a bias of reference into the inner controller of the semi-active damper, i.e. it is a fault in the actuation system.

The measurements of the QoV suspension system, using CarSimTM as software-in-the-loop in Matlab/SimulinkTM, are used for fault estimation according to the methodology presented in section III. Figure 2 shows that the fault estimation is not perfect, but it converges to the implemented fault. Note that the FDI module is insensitive to the bumps (road disturbances). This information can be directly compared with a threshold to warn the driver about the faulty damper.

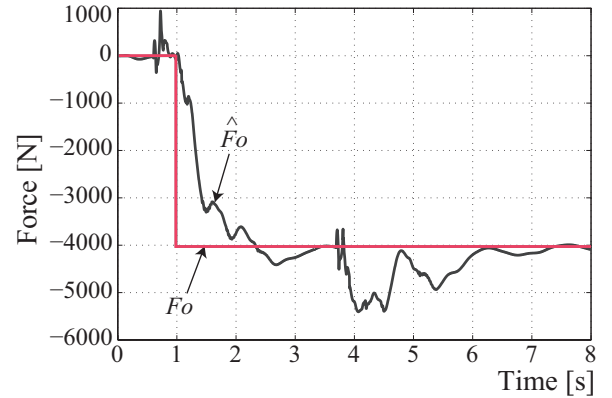


Fig. 2. Fault estimation.

Moreover, a suspension control system of the vehicle could take into account this fault estimation in order to maintain the desired performance of comfort and/or road holding.

Plots in Fig. 3 compare the performance of an uncontrolled shock absorber respect to the controlled one with fault-tolerance features (Fault-Tolerant Semi-Active: $FTSA$ damper) based on the control design proposed in section IV.

Note in Fig. 3 that before $t = 1 \text{ s}$ (fault free case) the $FTSA$ and uncontrolled system behave identically. When the fault occurs, $t > 1 \text{ s}$, the sprung mass acceleration is reduced when the $FTSA$ damper is used, Fig. 3a; the oscillations due to the bumps are more attenuated. The comfort of the passengers is increased in contrast to the uncontrolled damper that loses damping capability when the fault occurs. By computing the

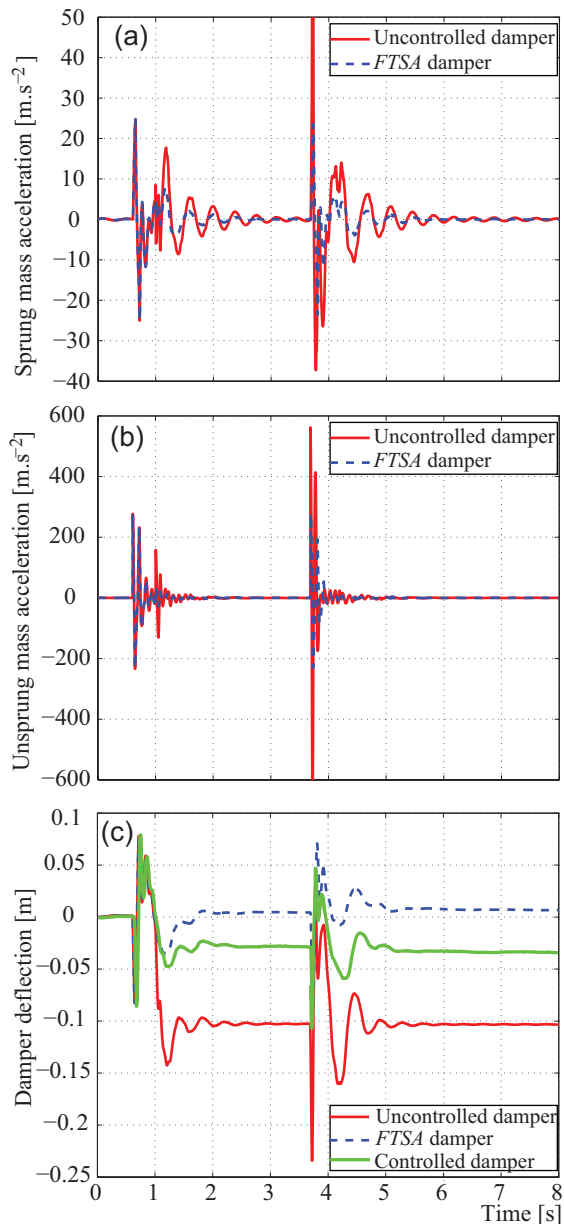


Fig. 3. Comparison between an uncontrolled and FTSA damper.

Root Mean Square (RMS) value of the \ddot{z}_s , the FTSA damper presents 3.31 m/s^2 versus to 6.68 m/s^2 of the uncontrolled damper, i.e. an improvement of 50.4% of comfort.

At the same time, Fig. 3b shows that the unsprung mass acceleration is also better due to the fault accommodation. These lower accelerations cause lower z_{us} and consequently lower deflection between the tire and road, i.e. an improvement of 42.4% in road holding (using the RMS of \ddot{z}_{us}).

Figure 3c shows the deflection between the masses, i.e. it is a monitor of the damper stroke. The robust fault information plus the robust control compensates the effect of the fault. During the second bump ($t = 3.7\text{s}$), the magnitude of the deflection is more attenuated. The uncontrolled damper is clearly affected by the fault, its stroke will remain roughly 10 cm lower than its normal position. This bias into the stroke

does not only affect the motion in the sprung and unsprung mass; it also could affect dramatically the life-time of the damper, i.e. this bias could cause that the damper piston arrives until the jounce-rebound stops.

Also Fig. 3c presents a comparison between the FTSA damper and a controlled damper which only uses the robust LPV feature without the reconfiguration mechanism. Although the LPV controller is designed to reduce the vibrations in the sprung and unsprung masses when external disturbances (e.g. the road profile) and faults occur, results show that this control approach is not good enough; in this case, the fault causes a bias in the damper stroke around 4 cm lower than its normal position. Inclusion of all possible faulty schemes is a drawback of passive FTC approaches. On the other hand, the proposed active FTC compensates the lack of effectiveness of the damper and permits to behave like a safe actuator. Early results show promising real implementations; the sampling frequency of the electronic control unit must ensure at least the response time of the MR damper (25 ms).

VI. CONCLUSIONS

An effective fault tolerant semi-active damper is achieved by synthesizing separately a robust fault estimator and controller. The idea was validated with a front-left QoV model using CarSimTM. Once the fault is detected, its estimation is used to compensate the effect; however, the magnitude of the fault could limit the bandwidth of control. The robustness of the controller/fault estimator shows good performances even if the model is not perfect. Simulation results show that the proposed fault-tolerant controller improves comfort 50.4% and road holding 42.4% and maintains the nominal position of the stroke when an additive fault occurs in the damper.

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