

Applicative Fault Tolerant Control for Semi-Active Suspension System : Preliminary Results \star

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Abstract—This paper presents applicative results about fault detection and control for semi-active suspension system. The control is extended from the “Acceleration Driven Damping” (ADD) controller combined with a robust fault detection scheme in order to estimate the fault. The approach, based on a quarter of vehicle model includes the nonlinearities of the shock absorber. The robust fault estimation module is based on \mathcal{H}_∞ background providing robustness. It aims to estimate a sensor fault on the system, especially a bias on accelerometers. Combination of both strategies allows to attenuate the effect the fault on the system. This approved structure aims to be implemented in real time. Validations has been made in a F-Class vehicle in CarsimTM software which provides realistic non-linear vehicle behavior. Results show the effectiveness of the fault tolerant strategy for semi-active dampers.

I. INTRODUCTION

Nowadays, modern cars become more and more complex in its structure in order to improve passenger comfort and security. As a consequence, different vehicle control strategies including robustness and/or fault tolerance [1] have been proposed to enforce safety in vehicles. Additionally, Fault-Tolerant Controller (FTC) are designed to maintain the stability of the system and the desired performance when occur process and/or instrument failures. It has been observed two different groups of FTC, [2]: passive (when the fault tolerance is designed off-line) and active (based on an automatic control reconfiguration mechanism).

Active fault tolerance often relies in the design of an external fault detection unit. Its signal, also called residual serves to reconfigure the controller to achieve good specifications. In the field of fault detection, several works have received considerable attention. Analytical redundancy-based methods are the most handled approaches [3], [4], [5]. Some statistical and geometrical techniques as proposed in [6], [7] and some observer-based methods [8] complete the field of fault detection.

In most cases, ground vehicles are modeled as Linear Parameter Varying (LPV) systems. This modeling allows to rewrite a non-linear system like a Linear one, scheduled by some parameters. The most commonly handled methodologies are the gridding approach, polytopic or Linear

Fractional Representation (LFR) one as proposed in [29] and references therein. However, those control strategies remain hard in terms of implementation and computation and are reserved for high performances computers. Recently, this modeling/control technique has been extended to FTC approaches. For instance, a comparison between an active and passive fault-tolerant LPV controller is presented in [10]. In [11], an FTC under multiple failures is proposed for polytopic LPV systems. Similarly, [12] proposed an FTC based on an LPV-model reference adaptive controller.

Particularly in automotive control applications, some researches on FTC are designed for active suspension systems: in [13] an FTC based on sliding mode observers is proposed, while in [14] a fault tolerant LPV control is designed to guarantee road holding and roll stability; in both proposals the system observability must be preserved. For semi-active suspension systems, a control strategy under different faulty schemes in a Quarter of Vehicle (QoV) model is proposed in [15], an on-line parametric estimation is used to create a fault signature by using parity relations; however, the fault is not estimated.

In this work, it is proposed a combination of an approved controller for a semi-active suspension and a fault estimator. In effect, all semi-active control strategies need information from sensors, like accelerometers and/or deflection sensors. However, a faulty information from one of the measurement can lead to worst results than a classical passive suspension. The study case exposed in this paper highlight this observation by analyzing the faulty case. In order to overpass this situation, the fault information taken into account to switch to a passive situation in order to attenuate the effect of the fault. The fault estimator is based on \mathcal{H}_∞ design, adapted from control design in order to achieve fault estimation.

The paper is organized as follows : sections II recalls the modeling of a quarter of vehicle equipped with a semi-active magneto-rheological (*MR*)-damper. Then section III presents the principle of the fault estimator, followed by the details about the controller under consideration in section IV. Thus an applicative study case developed in Carsim shows the effectiveness of the approach by comparing healthy and faulty situations in section V. Finally, section VI concludes the paper and exposes some perspectives as real implementation.

II. VEHICLE MODELING AND ANALYSIS

A. Modeling of the system

This work uses independent corner control system for semi-active suspension. A corner of a vehicle can be rep-

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resented as a lumped parameter *Quarter of Vehicle (QoV)* model illustrated on figure 1. It consists of the sprung mass

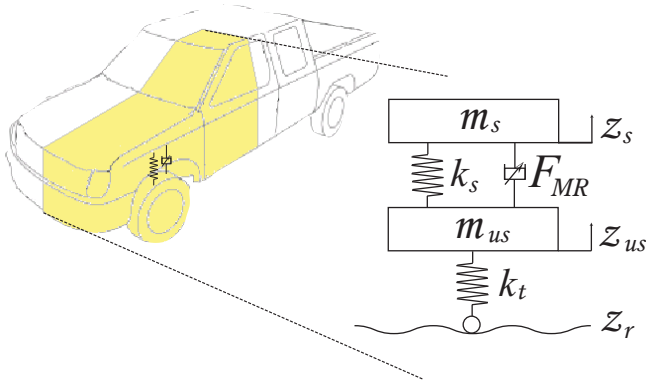


Fig. 1. Quarter of vehicle

m_s (weight of cabin, power engine, etc), unsprung mass m_{us} (weight of tire, wheel, arms), the suspension strut and the stiffness of the tire k_t . A suspension strut is composed of a spring k_s , a passive damping element c_p , and a variable damping c_v . The road disturbances, z_r , excites the *QoV* model. The vertical displacements of sprung, and unsprung masses, z_s and z_{us} , result from the ground excitation.

Finally, the dynamic behavior of a Quarter of Vehicle with a semi-active suspension is described by the following system of ordinary differential equations:

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

where m_s and m_{us} are the sprung and unsprung mass, respectively; z_{def} is the suspension deflection; z_r represents the road profile; k_s and k_t are respectively the stiffness of the suspension and the tire and F_{MR} the semiactive damping force.

The semi-active suspension is represented by an *MR*-damper, which can be modeled as the sum of various forces: a linear viscous friction (F_c), a spring force (F_k) and an electrically variable force (F_I) which is subject to saturations. Thus, the resulting model for the damper force (F_M) can be stated as:

$$\begin{aligned} F_M &= F_I + F_c + F_k \\ F_M &= \underbrace{I f_c \tanh(a_1 \dot{z}_{def} + a_2 z_{def})}_{F_I} + \underbrace{b_1 \dot{z}_{def}}_{F_c} + \underbrace{b_2 z_{def}}_{F_k} \end{aligned} \quad (2)$$

which can be rewritten:

$$F_M = I f_c \rho_1 + b_1 \dot{z}_{def} + b_2 z_{def} \quad (3)$$

where $\rho_1 = \tanh(a_1 \dot{z}_{def} + a_2 z_{def})$. The parameters f_c , ρ_1 , a_1 , a_2 , b_1 and b_2 characterize the magneto-rheological damper and has been obtained thanks to experimental results [32]. Its value are :

Variable	Description	Value
I	Electric current	[A]
f_c	Dynamic yield force in the MR damper model	600.95 [N.A ⁻¹]
ρ_1	Varying parameter to represent hysteresis of the damper	
a_1	Pre-yield viscous damping coefficient in MR damper	37.85 [N.m ⁻¹]
a_2	Pre-yield viscous damping coefficient in MR damper	22.15 [N.m ⁻¹]
b_1	Post-yield viscous damping coefficient in MR damper	2,830.86 [N.s.m ⁻¹]
b_2	Post-yield viscous damping coefficient in MR damper	-7,897.21 [N.s.m ⁻¹]

In the study, each corner of the vehicle is equipped with 3 sensors: chassis acceleration \ddot{z}_s , wheel acceleration \ddot{z}_{us} and a deflection sensor z_{def} . However, the chassis acceleration sensor \ddot{z}_{us} is faulty. The fault is considered to be additive. In effect, most of accelerometers are subject to biases which can be represented by an additive fault. The dynamical model given in (1) is completed with the dynamics of the damper (3). It results a state-space representation of the *QoV* model as :

$$\begin{aligned} \underbrace{\begin{bmatrix} \ddot{z}_s \\ \dot{z}_s \\ \ddot{z}_{us} \\ \dot{z}_{us} \end{bmatrix}}_{\dot{x}} &= \underbrace{\begin{bmatrix} -\frac{b_1}{m_s} & -\frac{c_1}{m_s} & \frac{b_1}{m_s} & \frac{c_1}{m_s} \\ 1 & 0 & 0 & 0 \\ \frac{b_1}{m_{us}} & \frac{c_1}{m_{us}} & -\frac{b_1}{m_{us}} & -\frac{c_1+k_t}{m_{us}} \\ 0 & 0 & 1 & 0 \end{bmatrix}}_A \underbrace{\begin{bmatrix} z_s \\ z_{us} \\ z_{us} \end{bmatrix}}_x \\ &+ \underbrace{\begin{bmatrix} -\frac{f_c \rho_1}{m_s} \\ 0 \\ \frac{f_c \rho_1}{m_{us}} \\ 0 \end{bmatrix}}_{B_I} I + \underbrace{\begin{bmatrix} -\frac{1}{m_s} \\ 0 \\ \frac{1}{m_{us}} \\ 0 \end{bmatrix}}_{B_F} F_o + \underbrace{\begin{bmatrix} 0 \\ 0 \\ \frac{k_t}{m_{us}} \\ 0 \end{bmatrix}}_{B_r} z_r \end{aligned} \quad (4a)$$

where the measurement vector y provided by the 2 accelerometers \ddot{z}_s , \ddot{z}_{us} and the deflection sensor z_{def} is given by :

$$\begin{aligned} y &= \begin{bmatrix} \ddot{z}_s \\ \ddot{z}_{us} \\ z_{def} \end{bmatrix} \\ &= \underbrace{\begin{bmatrix} -\frac{b_1}{m_s} & -\frac{c_1}{m_s} & \frac{b_1}{m_s} & \frac{c_1}{m_s} \\ \frac{b_1}{m_{us}} & \frac{c_1}{m_{us}} & -\frac{b_1}{m_{us}} & -\frac{c_1+k_t}{m_{us}} \\ 0 & 1 & 0 & -1 \end{bmatrix}}_C \underbrace{\begin{bmatrix} \dot{z}_s \\ z_s \\ z_{us} \\ z_{us} \end{bmatrix}}_x \\ &+ \underbrace{\begin{bmatrix} -\frac{f_c \rho_1}{m_s} \\ \frac{f_c \rho_1}{m_{us}} \\ 0 \end{bmatrix}}_{D_I} I + \underbrace{\begin{bmatrix} 0 \\ \frac{k_t}{m_{us}} \\ 0 \end{bmatrix}}_{D_r} z_r + \underbrace{\begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix}}_{D_F} F_o \end{aligned} \quad (4b)$$

with $c_1 = k_s + b_2$.

In this model, non linearity of the *MR*-damper ρ_1 appear in a linear way in the matrices B_I and D_I of the system,

so the system is LPV. Moreover, the system is subject to the road excitation z_r which is considered as a disturbance for control and fault estimation. The control input of the system is the electric current I inside the MR-damper.

B. Analysis of control laws

The QoV model suspension tuning looks for good comfort, road holding and safe suspension deflections, [26]:

- The vertical chassis acceleration (\ddot{z}_s) response to road disturbances (z_r), between 0 and 20 Hz, represents the felt acceleration by the driver, i.e. ride comfort specification.
- The vertical wheel deflection ($z_{us} - z_r$) response to road disturbances (z_r), between 0 and 30 Hz represents the ability of the wheel to stay in contact with the road, i.e. the road-holding specification.

For road holding the hard damping suspension is desirable, but it deteriorates the ride comfort. The performance will be obtained through of the control system focused on comfort and road holding. Hence the goals of the control system are: (1) the minimization of the vertical acceleration (\ddot{z}_s) gain in the frequency of resonance of the sprung mass, and (2) the minimization of the tire deflection ($z_{us} - z_r$), in the frequency of resonance of the tire deflection

III. FAULT ESTIMATION

The objective of the fault estimation is to estimate the fault F_o in the sensor. The proposed method is based on the \mathcal{H}_∞ theory. Since the classical \mathcal{H}_∞ approach is generally used for controller synthesis, it can be easily extended for fault detection and estimation. The objective of the synthesis is to minimize the error of estimation between the fault and its estimation, Fig. 2.

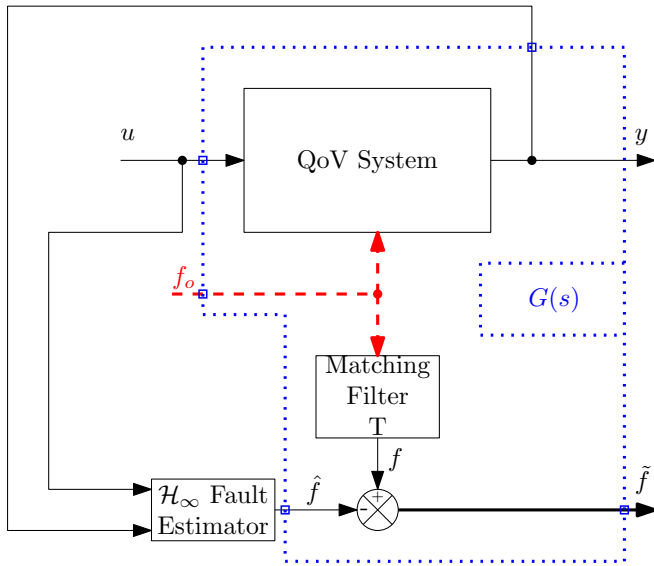


Fig. 2. Fault estimation problem

The optimization problem is rewritten in the “standard form” problem, Fig. 3, where it is added “weighting filters” W_I and W_O .

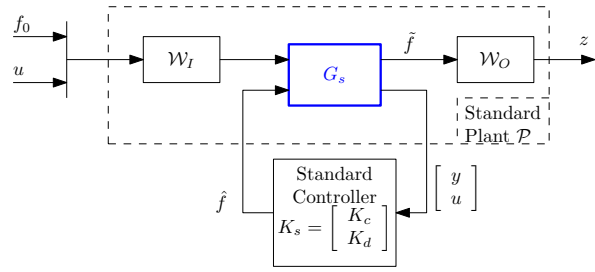


Fig. 3. Problem under standard form

These “weighting filters” W_I and W_O allow us to specify the frequency behavior of the closed loop system.

According to the scheme in Fig. 2, the fault estimation error is given by :

$$\begin{aligned}
 \tilde{f} &\triangleq f - \hat{f} \\
 &= TF_o - K_e \begin{bmatrix} y \\ u \end{bmatrix} \\
 &= TF_o - K_{e1}y - K_{e2}u \\
 &= TF_o - K_{e2}u - K_{e1}G_{QoV}u - K_{e1}G_f F_o \\
 &= (T - K_{e1}G_f)F_o - (K_{e2} + K_{e1}G_{QoV})u \quad (5)
 \end{aligned}$$

It can be shown that the fault estimation is directly affected by both the fault itself F_o and also the control input u .

A good estimation is achieved when the fault error \tilde{f} is as small as possible. In order to achieve this objective, a weighting filter W_f W_f as illustrated in Fig. 4 is introduced in the output \tilde{f} .

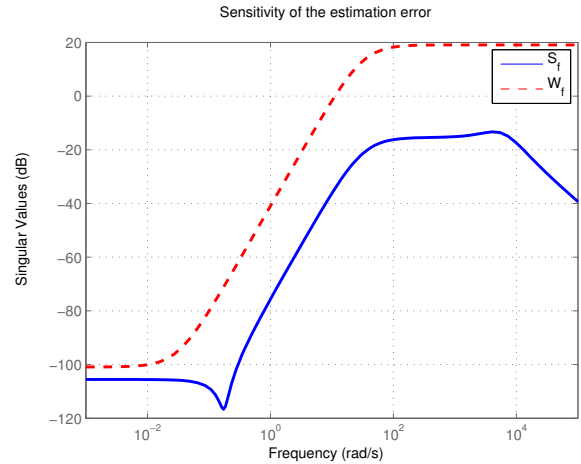


Fig. 4. Sensitivity of the estimation error S_f and its corresponding sensitivity filter W_f

Its strong attenuation in low frequencies permits to have a low estimation error \tilde{f} , and so a good fault estimation. On the other hand, the reader can easily appreciate the interest of the matching filter T since it allows to specify the frequency specification of the fault estimation.

Remark 1: In the structure of the fault estimator, it has been chosen to consider as inputs both : 1 - the output of

TABLE I
PARAMETERS OF A F-CLASS VEHICLE IN CARSIM™ MODEL.

Variable	Description	Value	Unit
$m_{s\ front}$	Front QoV sprung mass	546.5	Kg
m_{us}	QoV unsprung mass	50	kg
$k_{s\ f}$	Front spring stiffness	83	N/mm
k_t	All tire stiffness	230	N/mm

(m_{us}), and (m_s) are obtained from the full vehicle parameters. The lumped parameters of the QoV model follows a distribution of the body weight of 60 % in front and 40 % rear. Hence, a single front corner of the vehicle has a 30 % of the total body weight.

B. Results

It is added an additive fault F_o on the sprung mass accelerometer :

$$\ddot{z}_s = \ddot{z}_{s,acc} + F_o$$

The fault of $F_o = +10m.s^{-2}$ arrives at time $t = 10s$. Figure 7 shows the performance of the \mathcal{H}_∞ fault estimator.

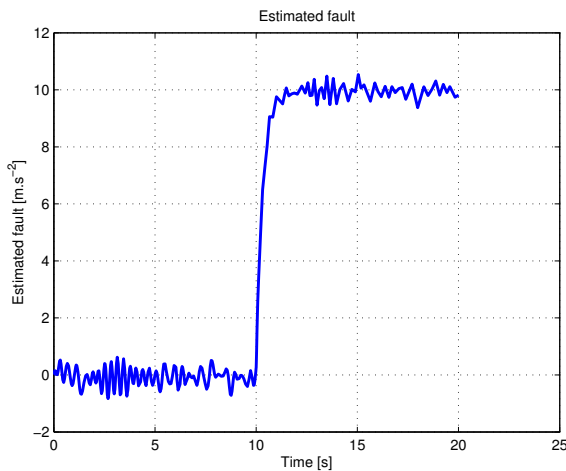


Fig. 7. Estimated fault

A reconfiguration mechanism has been also considered in this study. It consists of a simple switch. In fault free case, the semi-active control allows to increase the comfort of the passengers. When a fault is detected, the semi-active control is no more used, and a passive suspension is considered :

$$I = \begin{cases} I_{ADD} & \text{in fault free case} \\ I_0 & \text{if there is a fault} \end{cases}$$

where I_0 represents the nominal value of the current.

In effect, when the fault occurs the control strategy is degraded since its computation is based on the measurements.

The following figures show the roll (Fig. 8) and vertical acceleration (Fig. 9) of the vehicle. The classical ADD controller (solid blue lines) show that the performances of the controller are degraded in the presence of the fault ($t \geq 10s$).

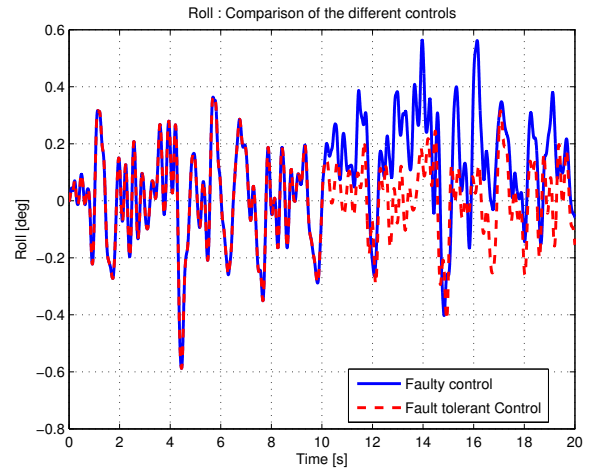


Fig. 8. Comparison of the roll

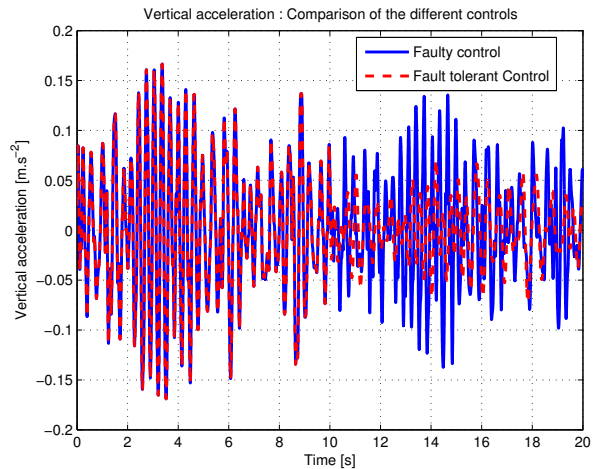


Fig. 9. Comparison of the vertical acceleration

Nevertheless, red dashed lines show the response of the accommodated Controller. When a fault is detected, the controller output is kept constant to a nominal value in order to achieve performances of a passive damper. Thanks to this approach, the performances are not so degraded.

VI. CONCLUSION

A robust Fault-Tolerant methodology for semi-active suspension control is presented in this paper. The innovation in this paper relies in the combination of an approved control law (namely ADD) and a Fault detection approach. In this paper, the fault detector is made from the \mathcal{H}_∞ background. The interest of this approach relies in the robust design. On the other hand, the ADD controller presents good performances for comfort. However, this paper highlight the effect of a sensor fault on the system. In effect, it is shown that the performances of the ADD controller are degraded when a fault occurs. The fault accommodation controller switch to a passive damper when a fault is detected. This

control strategy allows to reduce the bad performances due to the fault.

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