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HARDWARE-IN-THE-LOOP TESTING OF ON-OFF CONTROLLERS IN SEMI-ACTIVE SUSPENSION SYSTEMS*

Juan C. TUDÓN-MARTÍNEZ¹, Jorge de J. LOZOYA-SANTOS¹,
Ruben MORALES-MENENDEZ¹, Ricardo A. RAMIREZ-MENDOZA¹,
Oliver SENAME² and Luc DUGARD²

¹Tecnológico de Monterrey, Av. E. Garza Sada 2501, 64849, Monterrey N.L. México
{jc.tudon.phd.mty, jorge.lozoya, rmm, ricardo.ramirez}@itesm.mx

²GIPSA-lab, Control Systems Dept. CNRS-Grenoble INP, ENSE3,
BP 46, F-38402 St martin d'Hres cedex, Grenoble France
{olivier.sename, luc.dugard}@gipsa-lab.grenoble-inp.fr

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ABSTRACT

This paper presents an experimental validation of a proposed Frequency Estimation-Based (*FEB*) controller for semi-active suspensions by using a Hardware-in-the-Loop (*HiL*) platform of a Quarter of Vehicle (*QoV*) model. The *FEB* approach is compared with three commercial *On-Off* controllers that have shown good results in comfort and road holding: *Sky-Hook* (*SH*), *Groud-Hook* (*GH*) and *Mix-1-sensor* (*MIS*). The comparison was done under the same experimental tests; the standards ISO-2631 and BS-6841 are used to evaluate the comfort and the Root Mean Square (*RMS*) index to quantify the road holding. The *QoV* model belongs to a front-left corner of a pick-up truck; the used experimental Magneto-Rheological (*MR*) damper is not symmetric and only has 2 manipulation states. Experimental results show that the *FEB* controller has the best comfort performance at low frequencies (outperforms the benchmark controllers at 11.2%); while, for road holding, the improvement is slight; however, *FEB* controller works better for both goals simultaneously. By analyzing the suspension deflection, the *FEB* controller reduces up to 32.8% of motion respect to the *GH* controller. Additionally, the manipulation of the *SH* and *GH* controllers have several changes of actuation that do not allow the stabilization of the force in its desirable value; while *FEB* controller has a soft actuation defined on bandwidths.

Keywords: semi-active suspension control, hardware-in-the-loop, quarter of vehicle, MR damper

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1. INTRODUCTION

The main function of an Automotive Suspension Control System (*ASCS*) is to provide comfort by isolating the vehicle motion from the road irregularities, and to ensure the road holding in any driving condition. Due to the highly non-linear behavior of a shock absorber, this device is the key element in the *ASCS*. According to the properties for dissipating/absorbing energy, the automotive suspensions can be classified as passive, semi-active and active.

Passive dampers are the most used in commercial vehicles, the damping properties are designed mechanically; however, the constant damping coefficient has many limitations for ensuring comfort and safety simultaneously. On the other hand, semi-active and active dampers regulate the damping force through an *ASCS*. Normally, active suspensions achieve better comfort performance because they exploit the full *map force-velocity*; however, they require an external power system and its implementation could result more expensive than semi-active suspension systems.

Thus, semi-active dampers have variable damping force with lower power requirements; seminal results in [1] show that a semi-active suspension can decrease until 52% of vertical sprung mass acceleration. Commercially, there are two types of semi-active dampers: mono and double-tube. The mono-tube dampers are the most used; they can be Magneto-Rheological (*MR*), Electro-Hydraulic (*EH*) or pneumatic devices. The main advantages of the *MR* damper are: 1) fast time response (20-40ms), 2) large range of damping force and, 3) long bandwidth of control. An *MR* damper is a highly non-linear component with dissipative capability; its damping coefficient depends on the supplied electric current used to manipulate a magnetic field.

During last decades, there are several proposals of active and semi-active suspension control systems, some of them based on non-linear control theory [2-5], robust control solutions [6-8], optimum controllers [9], heuristic approaches [10-11] and *On-Off* strategies [12-14]. They have shown good results in comfort and some of them include criteria of safety. However, in an experimental evaluation, it has proved that semi-active dampers, with up to 25 operating states, only activates two states for an average of 92% of the driving time [15], concluding that an *On-Off* controller offers enough potential benefit to evaluate the semi-active property; additionally, the *On-Off* strategies are efficient and more feasible (light on-line computation and model-free synthesis) for being implemented in comparison with complex controllers. The *On-Off* control law computes the two values of manipulation as hard/soft damping.

Classical *On-Off* controllers such as *Sky-Hook* (*SH*) [12] and *Ground-Hook* (*GH*) [13] techniques have been exhaustively validated experimentally; however, they are not multi-objective. On the other hand, the *Mix-1-sensor* (*MIS*) controller [14] offers the possibility of selecting an exclusive control goal (comfort or road holding). In the aforementioned controllers, the controller output is not the damper manipulation; thus, the control law could compromise the controller practical feasibility by using a mapping algorithm from controller output to manipulation units. Hence, the design of a feasible and multi-objective *ASCS* is still an opportunity area. Recently, a novel controller based on the frequency estimation of motion is proposed in [16], the *Frequency Estimation-Based* (*FEB*) controller showed an efficient response for comfort and road holding in simulations by decreasing the trade-off between them.

This paper presents an experimental validation of the *FEB* controller in terms of comfort and road holding by using a Hardware-in-the-Loop (*HiL*) platform of a Quarter of Vehicle (*QoV*) model, the work is an extension of [16]. The *HiL* platform allows the validation of electronic control units and their algorithms by receiving and sending analogical signals from a mechanical component [17]. A comparative analysis among the *SH*, *GH*, *MIS* and *FEB* control strategies under the same experimental tests is presented; the standards ISO-2631 and BS-6841 are used to evaluate the comfort and the Root Mean Square (*RMS*) index to quantify the road holding. The *QoV* model is a front-left corner of a commercial pick-up truck; the experimental *MR* damper, manufactured by BWITM, is not symmetric and only has two manipulation states.

The outline of this paper is as follows: the next section presents briefly the control law of all *On-Off* control strategies used in this comparative analysis. Section 3 describes the *QoV* model, the experimental system and the design of experiments. The results are discussed in section 4. Finally, conclusions are presented in section 5.

2. ON-OFF CONTROLLERS IN SEMI-ACTIVE SUSPENSIONS

On-Off control strategies have been analyzed exhaustively in experimental tests, e.g. in *test-rigs* or *HiL*. They are considered free of models because only need limit conditions in the required measurements for being implemented; they are robust in the experimental phase. At difference to the *test-rig*, a *HiL* platform uses a model of concentrated parameters of a vehicle which is programmed into a microprocessor, i.e. the non-linear vehicle dynamics is ensured into the model. The microprocessor receives/sends signals to a hydraulic system that acts over the semi-active damper.

Based on their published experimental performance, the classical *SH* and *GH* controllers are used as benchmark in this work; additionally, the *MIS* control strategy offers interesting simulation results with the innovation of using only one sensor. The control laws of these approaches are briefly described.

Sky-Hook (SH). It minimizes the vertical acceleration of the sprung mass by connecting a virtual damper between the chassis and sky. The control law is given by:

$$C_{SH} = \begin{cases} C_{max} & \text{if } \dot{z}_s(\dot{z}_{def}) \geq 0 \\ C_{min} & \text{if } \dot{z}_s(\dot{z}_{def}) < 0 \end{cases} \quad (1)$$

Ground-Hook (GH). It reduces the dynamic *tire-road* forces for optimizing the road holding index. A fictitious damping element is proposed between the wheel and the ground. It is a concept similar to the *SH* controller. The control law is:

$$C_{GH} = \begin{cases} C_{max} & \text{if } -\dot{z}_{us}(\dot{z}_{def}) \geq 0 \\ C_{min} & \text{if } -\dot{z}_{us}(\dot{z}_{def}) < 0 \end{cases} \quad (2)$$

Mix-1-sensor (MIS). It is a control strategy that only demands one sensor in the control system of a *QoV*: an accelerometer on the sprung mass. The control law selects, at the end of each sampling, the maximum or minimum damping coefficient according to a cross-over frequency α , i.e. the term $(\ddot{z}_s^2 - \alpha^2 \dot{z}_s)$ is used as an instantaneous approximation in the time domain of the frequency range; when the controller is oriented to comfort α must be the frequency of resonance of the sprung mass. The control law is defined as,

$$C_{MIS} = \begin{cases} C_{max} & \text{if } (\ddot{z}_s^2 - \alpha^2 \dot{z}_s) \leq 0 \\ C_{min} & \text{if } (\ddot{z}_s^2 - \alpha^2 \dot{z}_s) > 0 \end{cases} \quad (3)$$

The damping in these reference controllers is considered directly proportional to the limit values of electric current of the experimental *MR* damper, i.e. $C_{min} \sim 0$ A means a soft *MR* suspension and $C_{max} \sim 2.5$ A means a hard *MR* suspension.

On the other hand, the proposed *FEB* controller is multi-objective (comfort and road holding) and its output is directly the manipulation signal over the semi-active damper, i.e. electric current. Following is briefly described its design.

2.1 Design of the *FEB* control strategy

This control strategy is based on the idea of monitoring the excitation frequency in the suspension for applying the required electric current into the semi-active damper in

order to achieve the desired control goal [16]. It is proposed to estimate the frequency of motion as:

$$\hat{f} = \sqrt{\frac{(z_{def_1}^2 + z_{def_2}^2 + \dots + z_{def_n}^2)}{(z_{def_1}^2 + z_{def_2}^2 + \dots + z_{def_n}^2) \cdot 4\pi^2}} \quad (4)$$

where, n is the number of samples to compute of the relative position and velocity. The manipulation is obtained by analyzing the frequency response of the controlled variables, i.e. a *look-up* table between frequency and electric current is defined in order to assure the desired performances for comfort and road holding.

Four bandwidths of control have been defined; the first two are related to comfort and the remaining two to road holding and suspension deflection. If the applied electric current is not the required in a specific bandwidth, the control goals are not achieved. This is the reason for having an accurate online estimation of the frequency. The electric current values selected in the *look-up* table, Table 1, are the optimum for ensuring the minimum gain in the load transfer, i.e. optimum for comfort and road holding by using the experimental *MR* damper of two states.

Tab. 1. Look-up table based on comfort and road holding simultaneously.

Bandwidth	1	2	3	4
$\hat{f}(=)Hz$	0-2	2-6	6-12	12-15
$I(=)A$	2.5	0	2.5	0

3. EXPERIMENTAL SETUP

The *HiL* configuration was implemented by simulating a *QoV* model which was programmed in an FPGA device embedded in a Compact Reconfigurable I/O (*cRIO*) system of NITM. The model was discretized by using a sampling frequency of 300 Hz.

The vertical dynamics of the *QoV* model is given by (5). The model parameters are: the sprung mass ($m_s = 711$ Kg), unsprung mass ($m_{us} = 81.5$ Kg), spring stiffness coefficient ($k_s = 42,230$ N/m) and stiffness coefficient of wheel tire ($k_t = 295,200$ N/m), that correspond to a commercial full size pick-up truck (front-left corner).

$$m_s \ddot{z}_s = -k_s(z_s - z_{us}) - F_{MR} \quad \text{and} \quad m_{us} \ddot{z}_{us} = k_s(z_s - z_{us}) - k_t(z_{us} - z_r) + F_{MR} \quad (5)$$

where \ddot{z}_s and \ddot{z}_{us} represent the vertical accelerations of m_s and m_{us} ; z_s and z_{us} are the vertical positions and z_r is the simulated road profile. F_{MR} is the experimental *MR* damping force given by: $F_{MR} = F_d / \sqrt{R_d}$, where F_d is the measurement of force obtained from the load cell and R_d is the motion ratio (i.e. ratio between the vertical position of the damper experimentally and its position in a McPherson suspension).

Thus, in the *HiL* configuration, the suspension deflection generated by the simulation of the *QoV* is the input of a hydraulic actuator *MTS*TM in order to move the *MR* damper; the suspension deflection is limited for the damper stroke in its effects jounce/rebound (± 50 mm). The measured *MR* force represents the *QoV* input and it is considered linearly distributed for the deflection between the masses. The *cRIO* monitors all process variables in a Human-Machine Interface (*HMI*).

Figure 1 shows the experimental system which is composed by 5 sections: (1) an operation and monitoring system by using an *HMI* designed in LabVIEW[®], (2) the *MR* damper which has two states of actuation (0 and 2.5 A) and an impedance of 2 Ω , (3) an hydraulic system of actuation *MTS*TM 407 for controlling the damper rod position with capacity of 20,690 kPa and bandwidth of 15 Hz, (4) an electric current controller *HCT*TM that uses a *PWM* signal of 2 kHz from 0 to 10V and, (5) the simulator of a *QoV* model in real-time and Electronic Control Unit (*ECU*) by using a *cRIO* system.

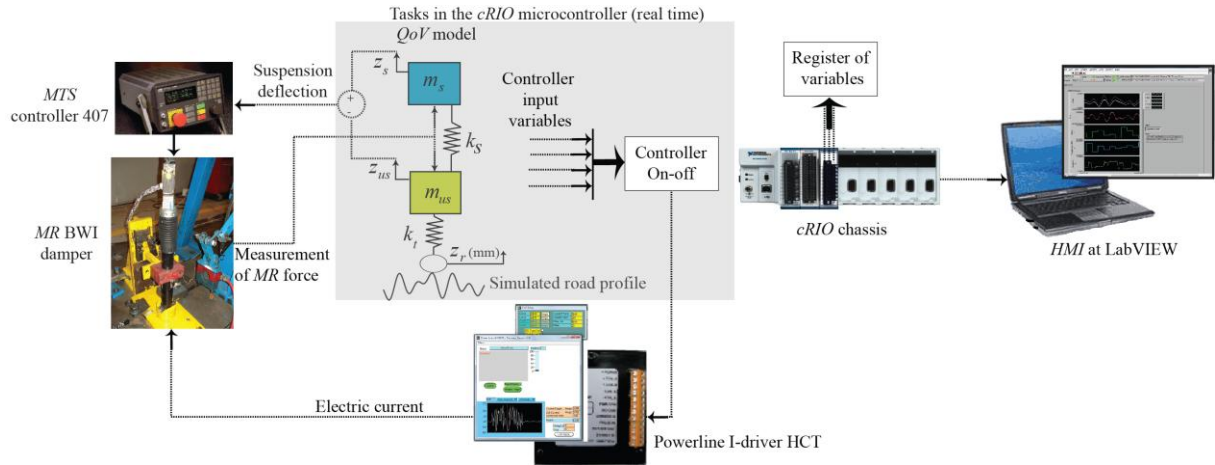


Fig 1. Experimental setup of the *HiL* configuration.

Two different tests of road profile over all control strategies were implemented in the *HiL* system. The test 1 is a road type F (rough road) according to the standard ISO-8606:1995, Fig. 2(a); this signal allows the analysis of the suspension motion in normal driving conditions. The test 2 represents a *Chirp* signal (0.25-5Hz) with decreasing amplitude (15-1mm), Fig. 2(b); this signal allows to explore uniformly the resonance motions of the sprung mass (comfort) and unsprung mass (road holding).

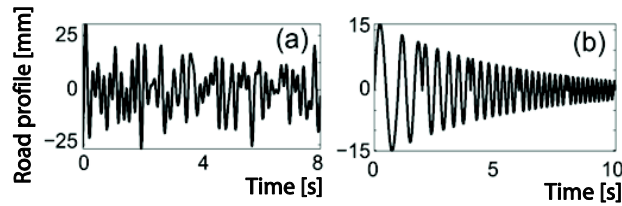


Fig 2. Implemented road profiles in the *HiL* configuration.

4. RESULTS

Experimental results of the *chirp* test are presented in Fig. 3 and 4. Figure 3 (a) shows that the *FEB* controller has the lowest gain in the vertical acceleration at low frequencies (<2Hz); after 2Hz, practically all suspension systems have same performance. For road holding, Fig. 3(b), *FEB* controller has acceptable performance, slightly better to the *GH* controller; however, the soft *MR* suspension (at 0A) presents the best road holding index at low frequencies. The difference between the *FEB* controller and the soft *MR* suspension is around 3 mm. Additionally, Fig. 3 shows the trade-off between comfort and road holding at low frequencies, i.e. the hard *MR* suspension (2.5 A) is good for comfort but limited for road holding and vice versa.

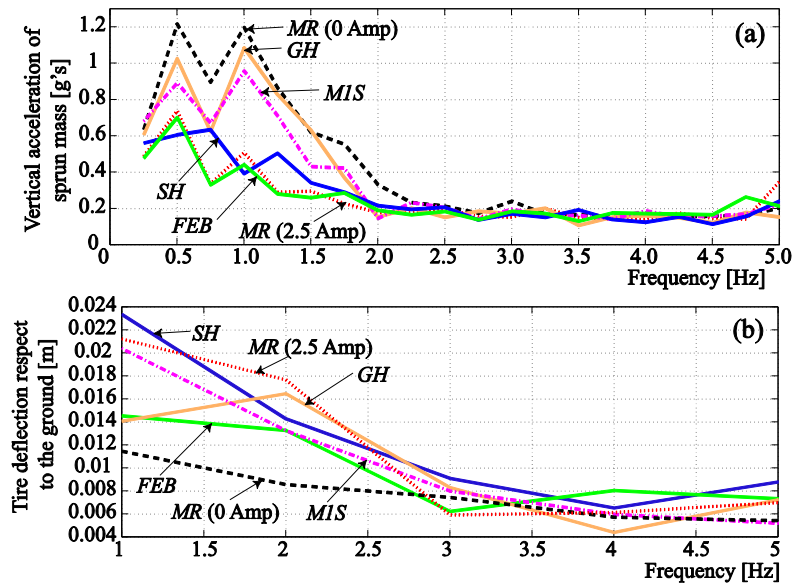


Fig 3. Analysis of comfort (up) and road holding (bottom) in the frequency domain (pseudo-Bode), by considering a *chirp* test as road profile in the *HiL* configuration.

The transient response of \ddot{z}_s , Fig. 4(a), shows that the magnitude is decreased up to 1 g whit the *FEB* controller in comparison with *GH* and *MIS*; this improvement is presented close to the resonance frequency of m_s (~ 1 Hz). By design of the *chirp* test, at beginning more suspension deflection is demanded, Fig. 4(b); however, *FEB* and *SH* do not show saturation in the displacement of the damper rod.

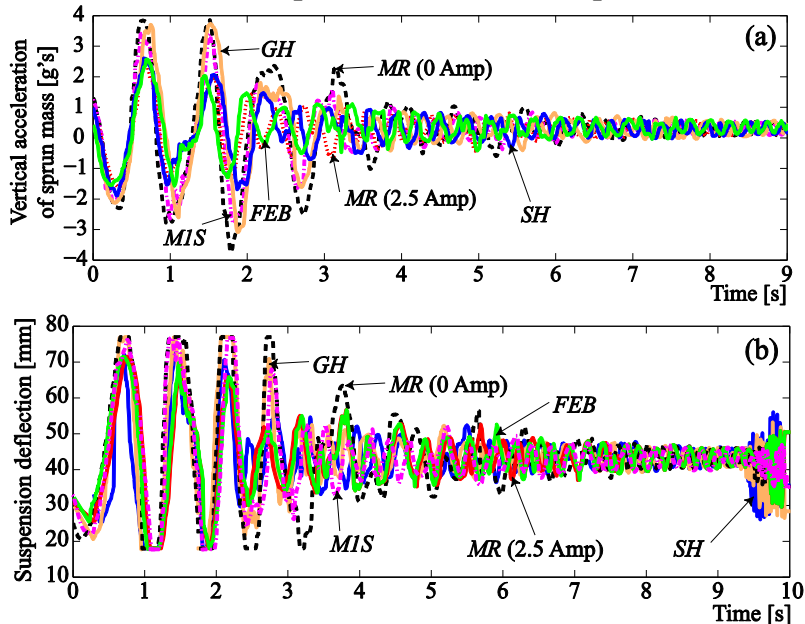


Fig. 4. Analysis of comfort (up) and suspension deflection (bottom) in the time domain, by considering the *chirp* test as road profile in the *HiL* configuration.

Table 2 shows the quantitative performance of comfort in both tests by using the standard BS 6841. In the *chirp* test, the *FEB* controller has the best performance i.e. an improvement between 11.2% and 39.4%, depending of the benchmark controller; practically few perception of movement according to the ISO 2631. For the ISO road test, *MIS* and *FEB* have the better comfort index. By analyzing the road holding in

both tests, *RMS* index, there is not a significant difference among the controllers. Finally, the *FEB* controller showed the best performance in the suspension deflection for the *chirp* test (reduction between 7.2% and 32.8%); while, for the ISO road test, all controllers presented similar results.

Tab. 2. Analysis of comfort, road holding and suspension deflection in both tests.

Controller	Maximum filtered acceleration by the standard BS6841 [m/s ²]		<i>RMS</i> of the road holding index [cm]		<i>RMS</i> of the suspension deflection index [mm]	
	<i>Chirp</i> test	ISO road test	<i>Chirp</i> test	ISO road test	<i>Chirp</i> test	ISO road test
<i>SH</i>	0.107	0.190	0.7	1.4	9.7	2.8
<i>GH</i>	0.156	0.170	0.7	1.4	13.4	2.8
<i>MIS</i>	0.157	0.161	0.8	1.4	12.0	3.0
<i>FEB</i>	0.095	0.165	0.8	1.3	9.0	2.9

Figure 5 shows the manipulation of all controllers in the ISO road test. Clearly, the *SH* and *GH* controllers have several changes in the level of actuation caused by the sign of the deflection velocity in the semi-active operating zone; in some cases, these sudden changes do not allow the stabilization in the force signal in its desirable value. The *FEB* strategy did not present this problem because the manipulation is defined by bandwidths.

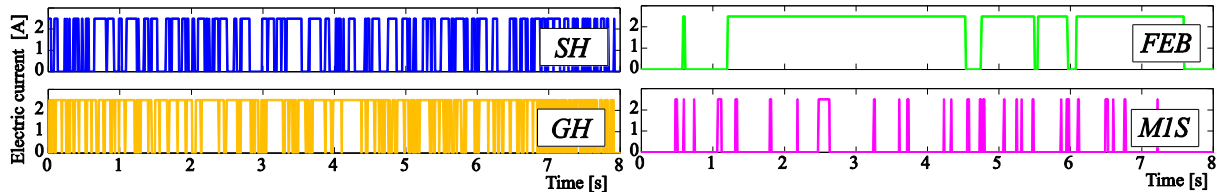


Fig. 5. Controller output in the control strategies for the ISO road test.

5. CONCLUSIONS

The *Frequency-Estimation-Based (FEB)* controller, which regulates the damping level based on an estimated frequency of motion, was experimentally validated by using a *Hardware-in-the-Loop (HiL)* configuration. The *FEB* controller was compared versus three commercial control strategies of semi-active suspension systems: *Sky-Hook*, *Ground-Hook* and *Mix-1-Sensor*. Experimental tests and standard monitoring indexes were designed in this comparative analysis for analyzing comfort and road holding. The *FEB* controller improved a 25% of comfort in average respect to the benchmark controllers and equaled the road holding performance. The controller design free of model, an easy computing, the generation of a soft continuous manipulation and, the demand of few measurements (low instrumentation cost) are advantages that make attractive this controller for comfort and road holding in an automotive semi-active suspension system.

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