A Comparison between a Model-free and Model-based Controller of an Automotive Semi-active Suspension System : Independent Wheel-stations

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Abstract: A comparison between two control strategies of automotive semi-active suspension systems is presented by using a pick-up truck model in CarSim^{TM}; the controllers are based on different frameworks. The Linear Parameter Varying (LPV) controller, considered as model-based controller, includes the constraints of the semi-active damper by using two scheduling parameters; while, the Frequency Estimation-Based (FEB) controller only requires measurements of a wheel-station for defining the damping force according to the objective controls, i.e. is free of a vehicle and actuator model. Since each wheel-station has an independent controller, the global semi-active suspension control system does not include the coupling effect among the four vehicle corners; however this effect is embedded into the controller performance. Experimental data are used to model a MR damper, which is used in each quarter of vehicle. A bounce sine sweep test is used to compare the performance in comfort and road holding of both controllers. Simulation results in CarSim^{TM} shows that the FEB controller has the best comfort and road holding performance; in comparison with the passive suspension system, the pitch angle is reduced 19%, the front and rear suspension deflection decrease 23% and 58% respectively and the tire compression is reduced 3% (front wheel) and 10% (rear wheel).

Keywords: Semi-active Suspension Control, Magneto-Rheological Damper, Linear Parameter Varying Control, Heuristic Control.

1. INTRODUCTION

Recently, suspension systems play a key role in vehicle design. In the standard ISO 2631, it has been established that the human body could suffer different health damages if this is exposed to constant vibrations, specially at frequencies between 0.5 - 20 Hz. Thus, the design in automotive suspension systems is focused to ensure not only the wheel-road contact (road holding) but also to improve the passengers comfort by isolating the road irregularities of the vehicle body. In contrast with passive suspension systems, which are designed mechanically and have constant damping ratio, the semi-active and active suspension systems can regulate the vehicle motions by using a control strategy over the damping force.

By good performance, low price, low energy consumption and easy implementation, the semi-active suspension systems are widely used in automobile industry nowadays. Commercially, there are different technologies of semi-active dampers, the main advantages of a Magneto-Rheological (MR) damper versus others semi-active dampers are: (a) fast time response (20-40 ms), (b) large force range and, (c) long bandwidth of control.

An MR damper contains metallic particles into the oil that modify the rheological properties of the fluid when a magnetic field is manipulated by an electric current signal; the flow resistance variation causes a highly nonlinear behavior in the damping ratio. The MR damper in an Automotive Suspension Control System ($ASCS$) is used to dissipate the energy for reducing the motion in the sprung mass and maintaining the wheel-road contact.

The semi-active suspension control design has been tackled with many approaches based on different frameworks during last decades. For its facility of design and widely experimental evaluation, the classical Sky-Hook (SH) controller proposed by Karnopp et al. (1974) and its improvements such as the modified SH [Hong et al. (2002)], mixed SH-ADD [Savaresi and Spelta (2007)], Mix-1-$sensor$ [Spelta et al. (2010)] and others, offer a good alternative to improve comfort; however, such control strategies neglect the road holding objective into the control law. In Sung et al. (2008), Sankaranarayanan et al. (2008) and Tudón-

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Martínez et al. (2012) is showed the performance of some of these controllers by using a full vehicle semi-active suspension system with independent wheel-stations.

On the contrary, the Ground-Hook (GH) controller [Valasek et al. (1997)], designed in a similar way to SH, improves only the road holding. By combining the SH and GH techniques, the hybrid controller proposed in Ahmadian (1997) allows to improve both control objectives; however, a comparison between the hybrid controller and an heuristic control strategy, named *Frequency Estimation-Based* (FEB) controller proposed by Lozoya-Santos et al. (2011), shows that the heuristic approach has better results of comfort and handling in a full vehicle semi-active suspension system with softer changes in the manipulation of the semi-active dampers. Other heuristic controllers [Ikenaga et al. (2000), Swevers et al. (2007), Dong et al. (2009)], considered as free of model, have been designed to improve comfort in a vehicle, but they use a master controller to include the coupling features among the four wheel-stations and have been analyzed for small or medium-size vehicles and not pick-up trucks.

On the other hand, the nonlinear control techniques such as Linear Parameter-Varying (LPV) control [Poussot-Vassal et al. (2008), Do et al. (2010)] and Sliding Mode Control (SMC) [Assadsangabi et al. (2009)] offer interesting simulating results in a Quarter of Vehicle (QoV) design, where the objectives of comfort and road holding are considered into the control law by including the properties of hysteresis, saturation and semi-activity of the damper, e.g. the LPV approach presented in Do et al. (2010). These nonlinear techniques also are evaluated in full-vehicle models [Chamseddine et al. (2006), Yoon et al. (2010), Poussot-Vassal et al. (2011)] with good results; however, their design is considerable complex since two levels of control are demanded or different control modes are defined.

In this manner, by their good results in a QoV model, a comparative analysis between an LPV controller (model-based approach) and the FEB controller (model-free approach) is presented in this paper by using a full semi-active suspension system of a pick-up with independent controllers in the wheel-stations. In Lozoya-Santos et al. (2011) is presented a comparison between these controllers, but the analysis of performance does not include the nonlinear dynamics of a full-vehicle model and the QoV parameters correspond to a medium-size car. In Dong et al. (2010), a comparative research in semi-active control strategies for an MR suspension is presented, a nonlinear controller (SMC) showed the best performance in comfort but an heuristic controller (fuzzy) had the best road holding; the analysis is also based on a QoV system. Similarly, a quarter car test rig is used in Hudha et al. (2005) to compare the SH, GH and hybrid controller.

Based on robust control theory, the LPV controller is synthesized to improve comfort and road holding by using two scheduling parameters that include the constraints of the MR damper; thus, several experiments over a commercial MR damper are considered to model the nonlinearities of the semi-active damper dynamics. On the other hand, the FEB controller defines the electric current value to apply on the MR damper by bandwidths according to the desired objective control (comfort or road holding) and using an estimated frequency signal of motion in the suspension. Physical insights of the vehicle are used to model it in CarSim^{TM}, which is used as Software-in-the-Loop (SiL). The Bounce Sine Sweep (BSS) test is used to quantify the comfort and road holding performances in both controllers; the Root Mean Square RMS value over the variables of interest is used as performance index.

The outline of this paper is as follows: the formulation of the LPV controller is described in the next section. Section 3 presents the FEB controller design. Section 4 shows the simulation tests and a discussion of results is presented in 5. Finally, conclusions are presented in section 6.

Table 1. Definition of Variables.

Variable	Description
a_i	Pre-yield viscous damping coefficients
b_i	Post-yield viscous damping coefficients
f_c	Dynamic yield MR force
F_{MR}	MR damping force
I	Electric current
k_s	Spring stiffness coefficient
k_t	Stiffness coefficient of the wheel tire
m_s	Sprung mass in the QoV
m_{us}	Unsprung mass in the QoV
u_{sat}	Filtered and bounded controller output
$\dot{z}_{def}$	Damper piston velocity (deflection velocity)
z_r	Road profile (input disturbance)
z_s	Vertical position of the mass m_s
$\dot{z}_s$	Vertical velocity of the mass m_s
$\ddot{z}_s$	Vertical acceleration of the mass m_s
z_{us}	Vertical position of the mass m_{us}
$\dot{z}_{us}$	Vertical velocity of the mass m_{us}
$\ddot{z}_{us}$	Vertical acceleration of the mass m_{us}
ρ_1^*	Varying parameter for hysteresis/saturation of F_{MR}
ρ_2^*	Varying parameter for dissipativity constraint of u

2. LPV FORMULATION

In LPV controller design, first it is described the nonlinear behavior of the QoV system and then the gain scheduling parameters, used to achieve an LPV model, are defined.

2.1 QoV representation

An experimental MR damper model represents the suspension between both masses, Fig. 1. It is assumed that the wheel-road contact is ensured. All variables are described in Table 1. The system dynamics is given by:

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

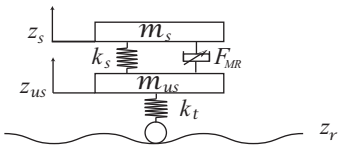

Fig. 1. Model for a QoV with a semi-active damper.

The MR damper model embedded in the QoV is represented by the parametric model of Guo et al. (2006), which emphasizes the hysteresis dependence on the position and

velocity of the piston by using a *tanh* function. The characteristic of a linear elastomer is included, by representing a stiffness factor. The non-linear model is defined by:

$$F_{MR} = f_c \cdot \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) + b_1 \cdot \dot{z}_{def} + b_2 \cdot z_{def} \quad (2)$$

where, the 5 coefficients have physical meaning, Table 1.

2.2 LPV modelling

Based on the *LPV* control framework proposed by Do et al. (2010), the varying parameter ρ_1 is defined by the nonlinear term of the *MR* damping force of (2), such that:

$$\rho_1 = \tanh\left(a_1 \dot{z}_{def} + a_2 z_{def}\right) \in [-1, 1] \quad (3)$$

A general state-space representation of the *QoV* model can be obtained, rewriting (1), (2) and (3) as:

$$\begin{aligned} \dot{x}_s &= A_s \cdot x_s + B_s \cdot u + B_{s1} \cdot z_r \\ y &= C_s \cdot x_s \end{aligned} \quad (4)$$

where, u is the control input in N and,

$$y = z_{def} = z_s - z_{us},\ C_s = [\,1\ 0\ -1\ 0\,]$$

$$B_s = \begin{bmatrix} 0 \\ \dfrac{-\rho_1}{m_s} \\ 0 \\ \dfrac{\rho_1}{m_{us}} \end{bmatrix},\ B_{s1} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \dfrac{k_t}{m_{us}} \end{bmatrix},\ x_s = \begin{bmatrix} z_s \\ \dot{z}_s \\ z_{us} \\ \dot{z}_{us} \end{bmatrix}$$

$$A_s = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\dfrac{k_s + b_2}{m_s} & -\dfrac{b_1}{m_s} & \dfrac{k_s + b_2}{m_s} & \dfrac{b_1}{m_s} \\ 0 & 0 & 0 & 1 \\ \dfrac{k_s + b_2}{m_{us}} & \dfrac{b_1}{m_{us}} & -\dfrac{k_s + b_2 + k_t}{m_{us}} & -\dfrac{b_1}{m_{us}} \end{bmatrix},\ u = f_c$$

In order to satisfy the dissipativity constraint in the control input, the state-space representation of (4) is rewritten in (5) and $u = f_c - f_0$, details in Do et al. (2010).

$$\begin{aligned} \dot{x}_s &= (A_s + B_{s2} \cdot \rho_2 \cdot C_{s2}) \cdot x_s + B_s \cdot u + B_{s1} \cdot z_r \\ y &= C_s \cdot x_s \end{aligned} \quad (5)$$

$$B_{s2} = \begin{bmatrix} 0 & -\dfrac{f_0}{m_s} & 0 & \dfrac{f_0}{m_s} \end{bmatrix}^T,\ C_{s2} = [\,a_2\ a_1\ -a_2\ -a_1\,]$$

where, f_0 is the average of f_c and $\rho_2 = \frac{\rho_1}{C_{s2} x_s} \in [0, 1]$.

Since the controlled input matrix B_s depends on ρ_1, it is required to add a low pass filter into de *QoV* model for getting a proper structure for the *LPV* based controller synthesis, Poussot-Vassal et al. (2008). Additionally, a saturation constraint (ρ_{sat}) is embedded into the filter for does not affect the closed-loop behavior.

A filter with bandwidth of 25 Hz was used for ensuring the time response of the *MR* damping force (~ 40 ms):

$$\mathcal{F}: \begin{bmatrix} \dot{x}_f \\ u_f \end{bmatrix} = \begin{bmatrix} A_f & B_f \\ C_f \rho_{sat} & 0 \end{bmatrix} \begin{bmatrix} x_f \\ u \end{bmatrix} \quad (6)$$

where, $\rho_{sat} = \frac{\tanh(C_f x_f / f_0)}{C_f x_f / f_0} \in [0, 1]$ approximates the saturation function of u_f in:

$$u_{sat} = \begin{cases} f_0 & if \quad u_f >> f_0 \\ u_f & if \ -f_0 \le u_f \le f_0 \\ -f_0 & if \quad u_f << -f_0 \end{cases} \quad (7)$$

Thus, the new structure (8), presented in Figure 2, takes into account the saturation and semi-activity of the damper into the varying parameters.

$$\begin{aligned} \dot{x}_{sf} &= A(\rho_1^*, \rho_2^*) \cdot x_{sf} + B \cdot u + \begin{bmatrix} B_{s1} \\ 0 \end{bmatrix} \cdot z_r \\ y &= [\,C_s\ 0\,] \cdot x_{sf} \end{aligned} \quad (8)$$

$$x_{sf} = \begin{bmatrix} x_s \\ x_f \end{bmatrix},\ A(\rho_1^*, \rho_2^*) = \begin{bmatrix} A_s + \rho_2^* B_{s2} C_{s2} & \rho_1^* B_s C_f \\ 0_{1\times 4} & A_f \end{bmatrix},\ B = \begin{bmatrix} 0_{4\times 1} \\ B_f \end{bmatrix}$$

where, $\rho_1^* = \rho_1 \rho_{sat} \in [-1, 1]$ and $\rho_2^* = \rho_2 \in [0, 1]$.

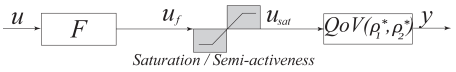

Fig. 2. *QoV* model with a semi-active saturated input.

2.3 LPV controller design.

In order to bound the nonlinearities of the closed-loop system, the *LPV* controller computes the manipulation signal by interpolation of a polytope whose vertices are Linear Time-Invariant (*LTI*) controllers. In this case, a polytope of 4 *LTI* controllers is designed by using the H_∞ control framework; the controller is quadratically stable for all trajectories of the scheduling parameters by solving an optimization problem with *LMI* techniques, Scherer et al. (1997). Figure 3 shows the block diagram for designing the H_∞ controller in each wheel-station. The weighting functions, based on a priori knowledge of the performance objectives, are designed as:

$$W_{z_{r,i}} = 3 \times 10^{-2},\ W_{\ddot{z}_{s,f}} = \frac{K_{s,f}\left[s^2 + 2\omega_{s1,f}\zeta_{s1,f} s + \omega_{s1,f}^2\right]}{s^2 + 2\omega_{s2,f}\zeta_{s2,f} s + \omega_{s2,f}^2}$$

$$W_{z_{us,f}} = \frac{K_{us,f}\left[s^2 + 2\omega_{us1,f}\zeta_{us1,f} s + \omega_{us1,f}^2\right]}{s^2 + 2\omega_{us2,f}\zeta_{us2,f} s + \omega_{us2,f}^2}$$

$$W_{\ddot{z}_{s,r}} = \frac{K_{s,r}\omega_{s1,r}^2}{s^2 + 2\omega_{s2,r}\zeta_{s2,r} s + \omega_{s2,r}^2},\ W_{z_{us,r}} = \frac{K_{us,r}\omega_{us1,r}}{s + \omega_{us2,r}}$$

where $W_{\ddot{z}_{s,i}}$ allows to reduce the amplification of $\ddot{z}_s$ in each quarter of vehicle for achieving the desired comfort, $W_{z_{us,i}}$ is shaped to ensure the road holding and $W_{z_{r,i}}$ increases the sensitivity to the road profile. By considering the control specifications, the generalized model $\mathcal{P}$ for the *LPV* control synthesis in each wheel-station is,

$$\begin{aligned} \dot{x}_{sf} &= A(\rho_1^*, \rho_2^*) \cdot x_{sf} + B \cdot u + \begin{bmatrix} B_{s1} \\ 0 \end{bmatrix} z_r \\ z &= C_1(\rho_1^*, \rho_2^*) \cdot x_{sf} \cdot \begin{bmatrix} W_{\ddot{z}_{s,i}} & 0 \\ 0 & W_{z_{us,i}} \end{bmatrix} \\ y &= C_2 \cdot x_{sf} \end{aligned} \quad (9)$$

where $C_1(\rho_1^*, \rho_2^*) = [\,C_{s1}\ \rho_1^* D_{s1} C_f\,]$ and,

$$C_2 = \begin{bmatrix} C_{s3} & 0 \end{bmatrix},\ C_{s3} = \begin{bmatrix} 1 & 1 & -1 & -1 \end{bmatrix},\ D_{s1} = \begin{bmatrix} \dfrac{-1}{m_s} \\ 0 \end{bmatrix}$$

$$C_{s1} = \begin{bmatrix} -\dfrac{k_s + b_2}{m_s} & -\dfrac{b_1}{m_s} & \dfrac{k_s + b_2}{m_s} & \dfrac{b_1}{m_s} \\ 0 & 0 & 1 & 0 \end{bmatrix} + \begin{bmatrix} -\dfrac{f_0}{m_s}\rho_2^* C_{s2} \\ 0_{1\times 4} \end{bmatrix}$$

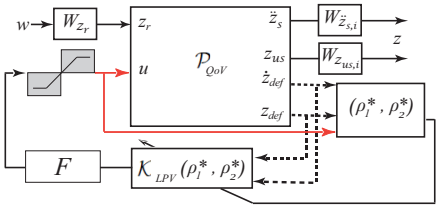

Fig. 3. *LPV* control structure based on H_∞ framework for an *LTI* model of a *QoV*.

3. *FEB* FORMULATION

The *FEB* controller is considered free of model, only measurements or estimated signals are used to monitor the excitation frequency in the suspension system in order to define the electric current value required to apply over the *MR* damper, and thus achieve the desired control objectives, Lozoya-Santos et al. (2011). This control strategy is considered multi-objective (for comfort and road holding).

Since the frequency of motion in the suspension system (f) is an indicator of its performance, with f it is possible to assign different values of damping coefficient in the semi-active shock absorber according to the frequency responses of the control objectives. Thus, it is possible to hold a specific suspension condition based on an estimated frequency ($\hat{f}$), which is used to manipulate the electric current in bandwidths. The measurements of the deflection velocity ($\dot{z}_{def} = \dot{z}_s - \dot{z}_{us}$) and suspension deflection ($z_{def} = z_s - z_{us}$) are used to estimate the frequency of motion and to decide which damping improves the comfort and/or road holding conditions. By using the *RMS* values of z_{def} and $\dot{z}_{def}$ of the *MR* damper piston, the estimation of the frequency in each *QoV* is given by,

$$\hat{f} = \sqrt{\frac{\left(\dot{z}_{def_1}^2 + \dot{z}_{def_2}^2 + \ldots + \dot{z}_{def_n}^2\right)}{\left({z_{def_1}}^2 + {z_{def_2}}^2 + \ldots + {z_{def_n}}^2\right) \cdot 4\pi^2}} \qquad (10)$$

where, $n = 256$ is the number of samples to compute the *RMS* value with a sampling time of 1/512. Since the damping coefficient is proportional to the electric current value in an *MR* damper, a *look-up* table between the manipulation signal (electric current) and the estimated frequency of the suspension motion is defined in order to assure the desired performances for comfort and road holding. Therefore, the *FEB* control law is completely independent of a vehicle or semi-active damper model.

4. SIMULATION TESTS

The vehicle model used to evaluate the controllers was characterized from real data (camber angles, caster, damping force, stiffness, etc.) obtained by a *K&C* (Kinematics and Compliance) test over a commercial pickup truck. The model was customized from a generic full-size light-load vehicle model in CarSim^{TM} software, Fig. 4, which is used as Software-in-the-Loop (*SiL*). The customization is composed by the physical dimensions (weight, long, width, height, wheel base, front and rear track and so on) and suspension system (independent wheel stations at the front side and a rear solid axle at the back).

Fig. 4. Full-size pickup truck model in CarSim^{TM}, by including *MR* dampers in the suspension system.

The Kerb weight is around 2000 Kg, the sprung and unsprung masses as well as the *QoV* model parameters described in eqn. (1) are presented in Table 2.

Table 2. *QoV* model parameters.

Front *QoV*		Rear *QoV*	
Parameter	Value	Parameter	Value
m_s	630 (Kg)	m_s	387 (Kg)
m_{us}	81.5 (Kg)	m_{us}	139.5 (Kg)
k_s	42,500 (N/m)	k_s	37,300 (N/m)
k_t	295,200 (N/m)	k_t	295,200 (N/m)

For *SiL* simulations, the semi-active suspension system is composed by four experimental *MR* damper models described by (2). The characterized *MR* damper is not symmetric in the jounce-rebound effects and only has two levels of actuation: low damping force at 0 A and high damping force at 2.5 A; the range of force is [−6000 to 11000] N (peak to peak), the damper stroke is around 50 mm with a time constant of 12 ms. The model parameters of the shock absorber are shown in Table 3; they were identified from a *chirp* signal with decreasing amplitude from 10 mm to 1 mm at [0.5-8] Hz. Figure 5 shows the modeling performance of the *Guo* structure.

Table 3. *MR* damper model coefficients.

Coefficient	Value	Units
a_1	21.3843	s/m
a_2	14.8223	1/m
b_1	4,630	s/m
b_2	-3,948.6	1/m
f_c	[951.5 - 3067]	N

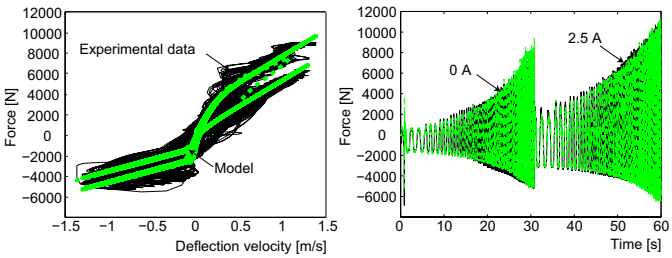

Fig. 5. Comparison between the *MR* damper model and experimental data.

Table 4 shows the parameters of the weighting functions used in the *LPV*-H_∞ control design, the cut frequencies have been selected according to the resonance frequencies of the sprung and unsprung masses, while the damping factors and gains are designed by loop-shaping.

The *BSS* test at 100 Km/h, which explores the most important range of frequencies for comfort and road holding, is used to evaluate the controller performances. The *RMS* value over the variables of interest is used as performance index in order to compare the controller performances respect to the passive one, according to:

$$x = \frac{v_{RMS_{Controller}}}{v_{RMS_{Passive}}} \qquad (11)$$

Table 4. LPV-H_∞ controller parameters.

Front QoV			
Parameter	Value	Parameter	Value
$K_{s,f}$	1	$K_{us,f}$	10
$\zeta_{s1,f}$	0.3	$\zeta_{us1,f}$	0.3
$\omega_{s1,f}$	$1(rad/s)$	$\omega_{us1,f}$	$9(rad/s)$
$\zeta_{s2,f}$	1	$\zeta_{us2,f}$	1
$\omega_{s2,f}$	$3(rad/s)$	$\omega_{us2,f}$	$9(rad/s)$
Rear QoV			
Parameter	Value	Parameter	Value
$K_{s,r}$	0.1	$K_{us,r}$	0.1
$\omega_{s1,r}$	$1(rad/s)$	$\omega_{us1,r}$	$9(rad/s)$
$\zeta_{s2,r}$	1	$\omega_{us2,r}$	$9(rad/s)$
$\omega_{s2,r}$	$3(rad/s)$		

where v is the RMS value of the variable of interest (pitch, heave and vertical acceleration for comfort, and suspension deflection and tire compression for road holding) during the test, $RMS_{Controller}$ refers to the controller under analysis and *passive* refers to the baseline suspension system by using a set of passive dampers. When the performance criterion value is greater than the unity, the passive suspension system is better than the semi-active suspension system, i.e. the index x monitors the improvement or deterioration of the $ASCS$.

5. RESULTS AND DISCUSSION

Figure 6 shows the transient response of both controllers respect to the baseline suspension system by considering the BSS test. For comfort, the semi-active suspension systems have lower oscillations in the pitch and heave signals close t=2 s (related to 2 Hz), while at t>3 s the vertical acceleration is reduced when the controllers are used. For road holding, the improvement is more significative in whole range of frequencies with both controllers, the damper compression in the front QoV is reduced up to 20 mm at different time instants (similar results are obtained in the rear damper), while the jounce motion of the rear solid axle is reduced up to 100 mm; by analyzing the tire compression, the improvement is presented between t=3-5 s, the reduction is up to 10 mm of displacement.

For a quantitative comparison between the FEB and LPV controllers, Table 5 shows the index of improvement (x) obtained by each controller in the BSS test. In all variables that monitor the comfort and road holding objectives, both control strategies have better performance than the passive suspension, except the vertical acceleration which in some time instants has lower amplitude in the oscillations caused by the soft suspension.

By comparing the FEB and LPV transient responses, in almost all variables the FEB controller has the best performance. For comfort, the FEB controller reduces the pitch angle 19 %; while for road holding, the front suspension deflection is reduced 23% and the rear up to 58%, similarly the tire compression is also reduced up to 10%. On the other hand, Figure 7 shows that the manipulation signal in the FEB controller is more persistent with softer changes than the LPV controller, whose manipulation depends on the time instant values of the scheduling parameters that could increase the natural waste in the actuation system.

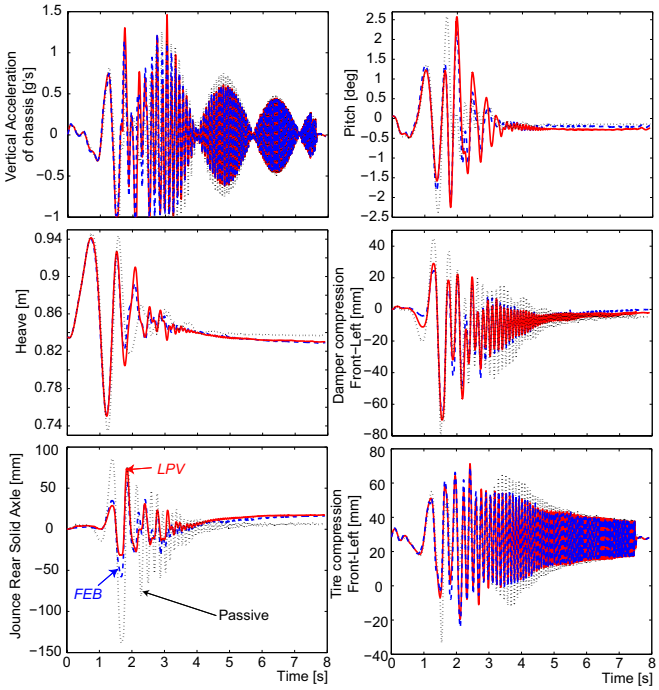

Fig. 6. Performance of the controllers in the BSS test.

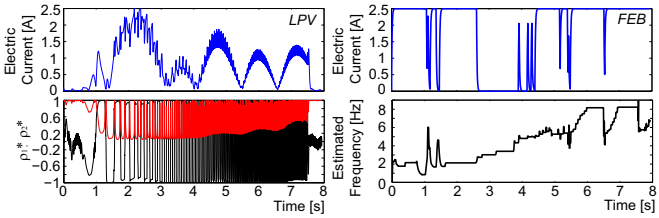

Fig. 7. Control input and control output in both $ASCS$, by analyzing a single wheel station.

6. CONCLUSIONS

A comparative analysis between a model-free and a model-based controller is presented in this paper; the full semi-active suspension system considers the four independent wheel stations, each Quarter of Vehicle (QoV) has an experimental Magneto-Rheological (MR) damper. The Frequency Estimation-Based (FEB) controller uses measurements to monitor the frequency of suspension motion in order to apply the required electric current over the MR damper; on the other hand, the Linear Parameter-Varying (LPV) controller requires a model of the QoV by including the semi-active damper dynamics. Two measurable scheduling parameters are included in the LPV control design to represent the nonlinear behavior (saturation and semi-activeness) of the MR damper.

Simulation results of the transient response show that both control strategies improve the comfort and road holding in comparison with the passive suspension system. By comparing the controllers, the FEB approach shows slightly the best comfort and road holding performance; the improvement of comfort respect to the passive system includes a reduction of 19% in the pitch angle; while for road holding, the front suspension deflection is reduced 23% and the rear up to 58%, i.e. the lifetime of the dampers is increased. Additionally, FEB controller reduces the tire compression up to 10% and LPV 17%.

Table 5. Performance comparison respect to the passive suspension system by using the x index.

Analysis of comfort					
Control Approach	Heave	Pitch		Vertical acceleration	
FEB	**0.99**	**0.81**		**1.13**	
LPV	**0.99**	0.84		1.14	
Analysis of road holding					
Control Approach	Damper Compression front QoV	Damper Compression rear QoV	Tire Compression front QoV	Tire Compression rear QoV	Jounce rear solid axle
FEB	**0.77**	**0.45**	**0.97**	0.90	**0.42**
LPV	0.84	0.46	0.98	**0.83**	0.46

Finally, the *LPV*-H_∞ controller is more complex because requires a reliable model of the whole system, but by design, the asymptotic stability is ensured; on the other hand, the *FEB* controller is more feasible to be implemented because only requires process measurements.

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