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Force Control System for an Automotive Semi-active Suspension *

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Abstract: A new semi-active suspension control system is proposed. This control system includes a Linear Parameter Varying (LPV) controller which was designed to improve the ride comfort. It also incorporates a Force Control System (FCS) to transform the force command from the LPV controller to a input signal for the Electro-Rheological (ER) semi-active damper. This FCS was assessed by its tracking performance of the desired force command, with a 7 % of tracking error. Then the semi-active control system was evaluated in a Quarter of Vehicle (QoV) model under two tests: a Bump and a Road Profile. The results were a reduction up to 19 % (Bump test) and 29 % (Road Profile test), of the sprung mass position compared with a passive suspension. Additionally, an improvement up to 14 % was obtained when compared with a LPV controller using a simple model inversion Force-Manipulation transformation.

Keywords: LPV systems, Automotive semi-active dampers, Electro-Rheological shock absorber

1. INTRODUCTION

The way a suspension system is tuned in the vehicle design process can affect ride comfort and road holding. Therefore, a passive suspension has to be designed to achieve a good compromise between these goals.

To overcome these passive damper limitations, semi-active shock absorbers can be used. These type of devices can online change their dissipation characteristics. Technologies such as Electro-Rheological (ER) or Magneto-Rheological (MR) are the most commercially used because of their advantages: fast time response (40 ms), large force range, wide bandwidth of control and cost.

ER damper force dynamic is highly non-linear (i.e. saturation, hysteresis, etc.). Figure 1 presents the Force-Velcity (FV) map of an ER shock absorber. These effects can be well modelled by equations that mimic the damper force \( F_D \) as a function of the damper deflection \( z_{def} \), deflection velocity \( \dot{z}_{def} \), and manipulation signal \( v \), Guo et al. (2006):

\[
F_D = c_p(\dot{z}_{def}) + k_p(z_{def}) + F_{sa}
\]

where \( F_{sa} = v \cdot f_e \cdot \tanh(a_1(\dot{z}_{def}) + a_2(z_{def})) \) is the semi-active force due to \( v \). Table 1 resumes the description of the variables.

Semi-active suspensions require a control system to maintain a desired performance. Normally, those controllers are designed to calculate a force that meets that performance, but a transformation from force to manipulation \( v \) is needed. Because of the non-linear damper characteristics, it is possible to achieve the same level of force at different conditions. As an example, the three red points in Fig. 1 (a, b, and c) correspond to the same \( F_D = 10 \text{ N} \) but, all are achieved at different average velocities and manipulations: (a) \( 0.12 \text{ m/s} \) with 10 %, (b) \( 0.08 \text{ m/s} \) with 20 %, and (c) \( 0.02 \text{ m/s} \) with 35 %. Moreover, the actuator dynamics cannot be neglected, Priyandoko et al. (2009). This condition makes the mapping from force to manipulation not a trivial task hence a tracking control system is needed.

This problem has been addressed in previous works. Kitching et al. (1998) used a cascade control system, the master controller tracks the desired damper force, whereas the

* Authors thank CONACyT and CRNS for their partial support in the bilateral México-France PCP projects 03/10 and 06/13.
slave one controls the opening valve position of the flow. Only the damper velocity was considered to compute the control algorithm and the damper force loop had a considerable delay. A neural network of the inverse model of the damper force was proposed in Chang and Zhou (2002). The desired force and two steps of the damper displacement are needed to obtain the manipulation. Hudha et al. (2005) used a PI controller coupled with conditional rules, however they do not take into account the dynamic of the damper force during the PI tuning. Similarly, Sam and Hudha (2006) proposed a PI controller and incorporated the dynamics of the damper flow valve; but the considered actuator was active thus did not consider the dissipativity restrictions of the semi-active dampers. Finally, Pellegrini et al. (2011) proposed an inverse model of the damper to obtain the manipulation, but they neglected the dynamic of the damper.

Unlike previous authors, this work proposes a Force Control System (FCS) that overcomes the non-linear force constraints of the damper by considering its dynamical response. This FCS is designed to operate along with a Linear Parameter Varying (LPV) controller for a semi-active suspension control system.

This paper is organized as follows. Section 2 describes the automotive semi-active suspension and its control system. Section 3 shows the design of the proposed FCS. Section 4 discusses the results. Finally, section 5 concludes the paper.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>zr</td>
<td>Road profile</td>
<td>m</td>
</tr>
<tr>
<td>z, zua</td>
<td>Sprung/Unsprung mass position</td>
<td>m</td>
</tr>
<tr>
<td>zdef, zdef</td>
<td>Damper/Tire deflection</td>
<td>m/s</td>
</tr>
<tr>
<td>z, zdef, zdef</td>
<td>Sprung/Unsprung mass velocity, damper deflection velocity</td>
<td>m/s</td>
</tr>
<tr>
<td>F_D</td>
<td>Damper Force</td>
<td>N</td>
</tr>
<tr>
<td>c_p</td>
<td>Viscous damping coefficient</td>
<td>N/s/m</td>
</tr>
<tr>
<td>k_p</td>
<td>Stiffness coefficient</td>
<td>N/m</td>
</tr>
<tr>
<td>F_sa</td>
<td>Semi-active component of the damper force</td>
<td>N</td>
</tr>
<tr>
<td>u</td>
<td>% of Manipulation</td>
<td>%</td>
</tr>
<tr>
<td>a_1</td>
<td>Hysteresis coefficient due to velocity</td>
<td>N/s/m</td>
</tr>
<tr>
<td>a_2</td>
<td>Hysteresis coefficient due to displacement</td>
<td>N/m</td>
</tr>
<tr>
<td>f_c</td>
<td>Damping coefficient</td>
<td>N/%</td>
</tr>
<tr>
<td>k_s, k_t</td>
<td>Spring/Tire stiffness</td>
<td>N/m</td>
</tr>
<tr>
<td>m_s</td>
<td>Sprung/Unsprung mass</td>
<td>kg</td>
</tr>
<tr>
<td>ω_d</td>
<td>Natural frequency</td>
<td>rad</td>
</tr>
<tr>
<td>k_d</td>
<td>Static gain</td>
<td></td>
</tr>
<tr>
<td>m_d</td>
<td>Dynamic damping coefficient</td>
<td></td>
</tr>
</tbody>
</table>

2. AUTOMOTIVE SUSPENSION SYSTEM

Figure 2 presents the block diagram of the LPV control system. This control system is designed using the Quarter of Vehicle (QoV) model.

2.1 QoV Model

The QoV model is used to analyse the vertical dynamics of a vehicle. It represents a quarter of the vehicle body with the wheel, tire, and suspension elements (spring and damper) with the following equations:

\( m_s \ddot{z_s} = -F_D - k_s(z_s - z_us) \)
\( m_us \ddot{z_us} = k_s(z_s - z_us) + F_D - k_t(z_us - z_r) \)  
(2)

where \( F_D \) is computed with two equations: 1) a static model and 2) a dynamical model, Fig. 3.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>z_r</td>
<td>Road profile</td>
<td>m</td>
</tr>
<tr>
<td>z_s, z_us</td>
<td>Sprung/Unsprung mass position</td>
<td>m</td>
</tr>
<tr>
<td>z_def, z_def</td>
<td>Damper/Tire deflection</td>
<td>m/s</td>
</tr>
<tr>
<td>z_s, z_def, z_def</td>
<td>Sprung/Unsprung mass velocity, damper deflection velocity</td>
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</tr>
<tr>
<td>F_D</td>
<td>Damper Force</td>
<td>N</td>
</tr>
<tr>
<td>c_p</td>
<td>Viscous damping coefficient</td>
<td>N/s/m</td>
</tr>
<tr>
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<td>N/m</td>
</tr>
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</tr>
<tr>
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<td>% of Manipulation</td>
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<td>N/s/m</td>
</tr>
<tr>
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<td>Hysteresis coefficient due to displacement</td>
<td>N/m</td>
</tr>
<tr>
<td>f_c</td>
<td>Damping coefficient</td>
<td>N/%</td>
</tr>
<tr>
<td>k_s, k_t</td>
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<td>kg</td>
</tr>
<tr>
<td>ω_d</td>
<td>Natural frequency</td>
<td>rad</td>
</tr>
<tr>
<td>k_d</td>
<td>Static gain</td>
<td></td>
</tr>
<tr>
<td>m_d</td>
<td>Dynamic damping coefficient</td>
<td></td>
</tr>
</tbody>
</table>

2.2 LPV Semi-Active Suspension Controller

A Linear Parameter Varying (LPV) controller is used. This controller incorporates in its design the characteristics of saturation and hysteresis of the semi-active damper to fulfill its force constraints, Do et al. (2010).

Substituting (1) in (2), and using \( f_v = v \cdot f_c \):

\[
\begin{align*}
    m_s \ddot{z_s} &= -k_s(z_s - z_us) - c_p(\dot{z_s} - \dot{z_us}) - k_p(z_s - z_us) - f_c \tanh(a_1(\dot{z_{def}}) + a_2(\dot{z_{def}})) \\
    m_us \ddot{z_us} &= k_s(z_s - z_us) + c_p(\dot{z_s} - \dot{z_us}) + k_p(z_s - z_us) + f_c \tanh(a_1(\dot{z_{def}}) + a_2(\dot{z_{def}})) - k_t(z_{def}) \\
\end{align*}
\]

To satisfy the dissipativity constraint of a semi-active damper, \( f_v \) must be constrained by

\[ 0 < f_{v_{min}} \leq f_v \leq f_{v_{max}} \]  
(5)

defining \( u_v = f_v - F_0 \), with \( F_0 = (f_{v_{min}} + f_{v_{max}})/2 \), the dissipativity constraint on \( f_v \) is recast as a saturation constraint on \( u_v \), i.e.

\[ -F_I \leq u_v \leq F_I \]  
(6)
where $F_I = (f_{v,\text{max}} - f_{v,\text{min}})/2$.

Denoting $k_f = k_s + k_p$, $z_{def} = z_s - z_{us}$, $z_{defI} = z_{us} - z_r$, and $\rho = \tanh(a_1(\dot{z}_s - \dot{z}_{us}) + \ldots + a_n(\dot{z}_s - \dot{z}_{us}))$, a state space representation of (4) is:

$$
P : \begin{cases} 
\dot{x} = A_s x + B_s \rho f_v + B_{sw} w \\
y = C_{sz} x 
\end{cases} 
$$

(7)

where $x = (x_1, \dot{x}_s, x_s, x_{us}, \dot{z}_s, \dot{z}_{us}, \dot{z}_{def})^T$, $w = \dot{z}_r$, $y = (z_{def}, \dot{z}_{def})^T$.

$$
A_s = \begin{pmatrix}
-\frac{k_f}{m_s} & -\frac{c_p}{m_s} & \frac{c_p}{m_s} & 0 & 1
0 & -1 & 0 & 0 & 1
\frac{k_f}{m_s} & -\frac{c_p}{m_s} & \frac{c_p}{m_s} & 0 & 0
\end{pmatrix},
B_s = \begin{pmatrix}
0
0
0
-\frac{1}{m_s}
\end{pmatrix},
C_s = \begin{pmatrix}
1 & 0 & -1 & 0 & -1
\end{pmatrix}^T
$$

(8)

The control input of (7) is parameter-dependent and it can be rewritten as follows:

$$
\begin{align*}
\dot{x} &= A(p_1, p_2) x + Bu + B_1 w \\
y &= C x
\end{align*}
$$

where:

$$
x = \begin{pmatrix} x_1, x_s, x_s, x_{us}, x_{us}, x_s, x_s, x_s, x_{us}, x_{us} \end{pmatrix}, \\
A(p_1, p_2) = \begin{pmatrix}
A_s + p_2 B_{s1} C_{s1} & p_1 B_s C_f
0 & 0
\end{pmatrix},
B = \begin{pmatrix}
B_{sw}
B_{sw}
0
\end{pmatrix},
C = \begin{pmatrix}
C_{sz}
0
\end{pmatrix},
\begin{align*}
B_{s1} &= \begin{pmatrix}
-\frac{F_0}{m_s} & 0 & \frac{F_0}{m_s}
0 & \frac{F_0}{m_s} & 0
\end{pmatrix},
C_{s1} &= \begin{pmatrix}
a_2 & a_1 & -a_1 & -a_1
\end{pmatrix},
D_{s1} &= \begin{pmatrix}
0
\end{pmatrix},
\rho_1 = \tanh(C_{s1} x_s) \tanh(C_{s1} x_s)
\end{align*}
$$

$x_f, A_f, B_f, C_f$ are the matrices corresponding to a state space representation of the low-pass filter $W_{\text{filter}} = w_f(s + w_f)$ which is added to the system to make the control input matrices parameter-independent.

The LPV controller, scheduled by $p_1, p_2$, has the form:

$$
K_c(p_1, p_2) : \begin{cases} 
\dot{x}_c = A_c(p_1, p_2) x_c + B_c(p_1, p_2)y \\
y = C_c(p_1, p_2) x_c
\end{cases} 
$$

(9)

This controller minimizes the $H_\infty$-norm of the transfer function between the input disturbances $w$ and controlled outputs $z$. The synthesis of the controller is made in the LPV/$H_\infty$ framework based on the LMI solution, see Scherer et al. (1997), for polytopic systems with quadratic stabilization, Do et al. (2010).

3. FORCE CONTROL SYSTEM

The LPV controller output $(F_{s\text{des},d})$ cannot go directly to the semi-active damper as an input, it needs to be transformed into a manipulation signal $(\nu)$. This signal is considered as a percentage of the possible range manipulation input. This transformation becomes complicated because of the non-linear behaviour of the damper, Fig. 1.

In addition to the LPV controller output $(F_{s\text{des},d})$, $z_{def}$ and $\dot{z}_{def}$ are needed to obtain the corresponding $(\nu)$. Hence, a simple inverse model is not enough, then a FCS is proposed. The FCS has two objectives: 1) to bound $F_{s\text{des},d}$ in a possible force range $(F_{\text{sa,des}})$ and 2) to compute the required $\nu$ input to achieve the desired damper force. Figure 4 shows the proposed semi-active suspension control system; the FCS is shown in detail.

![Fig. 4. LPV semi-active control system including the FCS.](image)

3.1 Control System Design

The FCS takes as reference the LPV output $(F_{\text{sa,des}})$ and sends the corresponding manipulation $(\nu)$ to the damper, Fig. 4. In the first place it is necessary to ensure that $F_{s\text{des},d}$ can be delivered by the ER damper. This is done by defining an admissible region $\mathcal{D}$ which includes the achievable force range of the real damper, Pouso et al. (2008). To bound the force a clipping function is used:

$$
\mathcal{D}(F_D, \dot{z}_{def}) \rightarrow F_D = \begin{cases}
F_D & \text{if } F_D \in \mathcal{D} \\
F_D & \text{if } F_D \notin \mathcal{D}
\end{cases}
$$

(10)

where an orthogonal projection of the desired force $(F_D)$ over the region $\mathcal{D}$ is assumed, this projection is driven...
by the current measure of the damper velocity. Figure 5 presents the admissible region $D$ and the simulated force for different percentages of manipulation.

Fig. 5. Admissible force region ($D$) and damper force at different manipulation levels.

Considering the ER damper model, the following control law is proposed:

$$v = g(x(t))^{-1} \dot{v}$$

(11)

with

$$G_c(s) = \frac{\hat{v}(s)}{e(s)}$$

(12)

where $g(x(t)) = f_c \cdot \tanh(a_1 \dot{z}_{def} + a_2 z_{def})$ and $e(t) = F_{sa_{des}}(t) - F_{sa}(t)$.

The controller $G_c(s)$, (12), is designed using the dynamic model (3), and classical control techniques, considering the following specifications: bandwidth around 100 rad/s, gain and phase margin greater than 12 dB and 45$^\circ$ respectively. A controller which fulfill these specifications is given by:

$$G_c(s) = \frac{86(s + 120)}{s(s + 80)}$$

(13)

4. RESULTS

The evaluation of the semi-active suspension control system is made in two steps: 1) Assessment of the FCS using the tracking error of the force as the performance index, and 2) evaluation of the control system at different tests, this evaluation is held in time and frequency domains.

4.1 Force Control System Assessment

To evaluate the performance of the FCS, a reference of force $(F_{sa_{des}})$ that mimics the characteristics and behaviour of an automotive semi-active damper under normal operating conditions is needed. For this purpose an ER shock absorber was simulated under different displacements $(z_{def})$, velocities $(\dot{z}_{def})$ and manipulations $(v)$ inputs to generate a force reference $(F_{sa_{des}})$. The selected signals are summarized in Table 2.

Figure 6 shows the response of the FCS. The Root-Mean-Square (RMS) value of the tracking error is used as performance index. Table 3 presents the RMS index of the tracking error and its normalization against the force range $(F_{D_{max}} - F_{D_{min}})$ corresponding to each test.

Fig. 6. Comparison of reference force (solid red) and the achieved force (dashed blue) at different displacement inputs to the damper model.

A qualitative analysis can be made based on Fig. 6. It can be seen that the proposed FCS was able to follow the reference signal, even for the Chirp signal, where the frequency of the wave increased up to 10 Hz.

Table 3. Tracking error of different tests.

<table>
<thead>
<tr>
<th>Test</th>
<th>#1</th>
<th>#2</th>
<th>#3</th>
</tr>
</thead>
<tbody>
<tr>
<td>RMS</td>
<td>1.63</td>
<td>2.01</td>
<td>1.42</td>
</tr>
<tr>
<td>Normalization RMS</td>
<td>5.66 %</td>
<td>3.96 %</td>
<td>13.14 %</td>
</tr>
</tbody>
</table>

A quantitative analysis was made by using Table 3. The RMS errors from the different experiments show that the FCS was able to track the reference for the different signals. In the test #2 with the Chirp signal, the error is higher, but not considerably. The error in Test #3 is considerably high, but in this case the force reference

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behaves as a square signal (i.e. the worst case). It can be concluded that the controller is able to follow the reference signal and the tracking error can be neglected.

4.2 Semi-active Suspension Control System Evaluation

To evaluate the performance of the proposed control system, three cases were compared: 1) A passive suspension system where the ER damper manipulation was fixed at 20 % (Passive), 2) the proposed control system (LPV + FCS) and 3) a LPV controller coupled with a Simple Model Inversion (LPV + SMI) function. The comparison was based on: ride comfort index, and road-holding index. In the LPV + SMI control system, the $F_{sa_{des}}$ command was transformed from force to manipulation by using the next simple inverse model function:

$$v(F_{sa_{des}}) = \begin{cases} 
35\% & \text{if } F_{sa_{des}} \geq f_c \cdot 35\% \\
F_{sa_{des}} & \text{if } f_c \cdot 10\% < F_{sa_{des}} < f_c \cdot 35\% \\
10\% & \text{if } F_{sa_{des}} \leq f_c \cdot 10\%
\end{cases} \tag{14}$$

this function substitutes the FCS block in Fig. 4.

Time Domain Evaluation. Figure 7 shows the results of a test with a Bump of 5 mm height, and Fig. 8 a test with a Road Profile input signal. Table 4 summarizes the RMS index. The reported indexes are computed as improvement of each variable against the Passive case, as:

$$\% \text{ of Improvement} = 1 - \frac{RMS(X_i_{Controlled})}{RMS(X_i_{Passive})} \tag{15}$$

where $X_i$ is the corresponding controlled variable ($z_s$ or $z_{def}$) for the control systems.

Figure 7 shows the displacement of the sprung mass and the deflection of the tire for the Bump test. This test was used to evaluate the response of the system against a highly uncomfortable situation. Figure 7a shows that the LPV + FCS control system was able to compensate the effect of the Bump with a smooth transition, while the Passive case presents amplification of the Bump effect and oscillations in its transient response. For the LPV + SMI control system, the simplistic transformation of the force to manipulation introduces a negative impact in the performance. Figure 7b shows, in both cases an improvement on the tire deflection and in their transient behaviour, meaning less tire bounce. The Passive case presents higher oscillations in its transient response.

Figure 8 compares the control systems for the Road Profile test. This test evaluates the performance of a control system in a common automotive operation condition during riding. Figure 8a shows how both control systems LPV + SMI and LPV + FCS reduce the movement of the sprung mass, having better performance the LPV + FCS case. Figure 8b shows the response of $z_{def}$; it can be observed in the detail view that the LPV + FCS control system has a better performance.

Table 4 shows a quantitative comparison, it can be seen that the LPV + FCS controller has the better suspension performance. Regarding comfort, in the Bump test the improvement is considerably higher (19.14 %) compared with the LPV + SMI case (5.49 %), in both cases they were better than the Passive case. The same occurs in the Road Profile test where the LPV + FCS has a 29.38 % of improvement compared with the Passive case, against
Table 4. RMS index performance improvement.

<table>
<thead>
<tr>
<th>Case</th>
<th>Variables</th>
<th>( z_s / z_r )</th>
<th>( z_{def} / z_r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bump Test</td>
<td>LPV + FCS</td>
<td>19.14%</td>
<td>12.09%</td>
</tr>
<tr>
<td></td>
<td>LPV + SMI</td>
<td>15.67%</td>
<td>11.87%</td>
</tr>
<tr>
<td>Road Profile Test</td>
<td>LPV + FCS</td>
<td>29.38%</td>
<td>21.92%</td>
</tr>
<tr>
<td></td>
<td>LPV + SMI</td>
<td>19.14%</td>
<td>12.09%</td>
</tr>
</tbody>
</table>

the 15.67 % obtained by the LPV + SMI control system. These results are consistent with the qualitative ones.

Frequency Domain Evaluation. Figure 9 presents the frequency response of \( z_s / z_r \), and \( z_{def} / z_r \) functions, Poussot-Vassal et al. (2012). It can be seen that the LPV + FCS control system has better performance in comfort \( (z_s/z_r) \) than both Passive and LPV + SMI cases, specially in the resonance frequencies. For the road-holding \( (z_{def}/z_r) \) the LPV + FCS control system has also better performance in the range of 0-5 Hz. It can be seen that the use of the FCS improves the performance of the LPV control system.

Remark 1. In the frequency analysis the range beyond 15 Hz was not take into account due to unmodeled dynamics in the mathematical model beyond that point.

5. CONCLUSIONS

A Force Control System (FCS) was proposed to improve a Linear Parameter Varying (LPV) control system. The FCS considers the non-linear dynamic behaviour of an Electro-Rheological (ER) damper, Fig. 1. Due to these non-linearities, the damper can deliver wrong output manipulations in different conditions, the FCS adjusts the manipulation to reach the force reference, regardless of the uncontrolled variables in the force control loop \( (z_{def}, \text{and } \dot{z}_{def}) \).

In order to validate the proposal, the LPV + FCS control system was compared with a LPV plus a Simple Model Inversion (SMI) mapping function, taking the Passive case as reference. The LPV + FCS control system proved its effectiveness by maintaining the original control objectives with better performance in comfort and road-holding.

REFERENCES


