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A LPV control approach for a semi-active hydraulic damper

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ABSTRACT

In this paper, a new control strategy is developed to improve comfort and roadholding of a ground vehicle equipped with an industrial damper. This damper can be controlled by a small servomechanism which adjusts the damping rate. The main controller is a Linear Parameter Varying static state-feedback controller synthesized in the H_∞ /LPV framework to compute the required damping force that minimizes the movements of the vehicle's body on one hand, and the deflection of the tire on the other hand. A scheduling strategy is developed on the basis of the real damper behavior to improve performances without using active damping forces which would be useless for such a semi-active system. Here the controller takes into account the technological constraints, the damper behavior and is easy to implement in an industrial application. The comfort and roadholding levels of the semi-active suspension are studied using some adapted criteria and compared with the passive suspension. Some simulations emphasize the improvements of this control strategy that will be tested by SOBEN on a testing car in the near future.

Keywords: automotive suystems, identified models, H_∞ /LPV control, semi-active, scheduling strategy

1. INTRODUCTION

The main role of suspensions is to improve comfort by isolating the vehicle chassis from an uneven ground and by providing a good roadholding to ensure the safety of the passengers, especially during a bend. Suspension control based on quarter (1/4) vehicles has been widely explored in the past years to improve vertical movements. Active control laws have been developed : Skyhook [13], [5], H_∞ control [7], LPV [6] or mixed synthesis [1], [15], and semi-active control laws [14], [16]. Semi-active suspensions are very interesting because of their low energy consumption when compared to active ones and their high performances when compared to passive ones.

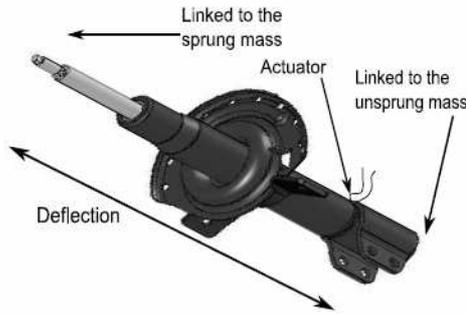
This paper develops a semi-active control strategy that optimizes the vehicle behavior considering the actuator's constraints and the damper behavior in the controller. A semi-active suspension control strategy based on H_∞ /LPV techniques has already been developed in [9]. Here this previous study is completed: the performance specifications are scheduled and the damper limitations are determined using identified models. Finally, an efficient and complete industrial solution including a high-level controller and a low-level force control-loop is proposed.

This paper is organized as follows: the SOBEN damper and its actuator are described and modelled in Section II. In Section III, the control strategy is developed. In Section IV, some simulation results are given in time and frequency domains: they show the interest of semi-active suspensions when compared with passive ones. Conclusions are given in Section V and some possible future works are proposed.

2. PRESENTATION AND MODELING OF SOBEN DAMPER

2.1 SOBEN damper

The system under study is the semi-active hydraulic damper designed by SOBEN. The oil flow in the damper is controlled with a single servomechanism. Some experiment results have been used to identify a simplified model of the damper given by Equation 1. This model has been proposed by [12] for magneto-rehological dampers. This is a static nonlinear model that gives the damping force using the deflection and deflection speed:



$$F_s = A_1 \tanh(A_2 v + A_3 x) + A_4 v + A_5 x \quad (1)$$

Fig. 1. New SOBEN damper

In Equation 1, F_s is the damping force, v the deflection speed, x the deflection and $A_i, i \in [1,5]$ the identified parameters. The model is used by the high-level controller to compute a realistic required damping force taking into account the behavior of the damper.

2.2 Actuator

The actuator chosen for SOBEN damper is the flow control solenoid valve. The average damping rate of the damper can be controlled by changing the input current of the servomechanism according to a second order model given by Equation 2. The parameters of this model have been identified.

$$G_{act}(s) = \frac{C(s)}{I(s)} = \frac{G}{\left(\frac{s}{\omega_0}\right)^2 + 2m \frac{s}{\omega_0} + 1} \quad (2)$$

2.3 Vehicle Model

The vehicle model used in this paper is a vertical linear $\frac{1}{4}$ car model represented on Figure 2. The identified damper model given by Equation 1 has been embedded in the quarter car model.

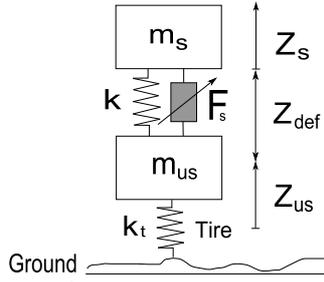


Fig. 2 Vertical quarter car vehicle

m_s, m_{us}	Sprung, unsprung mass
k, k_t	Suspension, tire stiffness
z_r	Ground vertical position
$\ddot{z}_s, \ddot{z}_{us}$	Sprung, unsprung mass acceleration
z_s, z_{us}	Sprung, unsprung mass position
$z_{def} = z_s - z_{us}$	Suspension deflection
F_s	Damping force

Table. 1 Quarter car parameters and variables

The equations of this model are given by Equation 3. This model will be used later as reference model for simulations.

$$\begin{cases} m_s \ddot{z}_s = k(z_{us} - z_s) + F_s \\ m_{us} \ddot{z}_{us} = k(z_s - z_{us}) - F_s + k_t(z_r - z_{us}) \end{cases} \quad (3)$$

3. CONTROL ARCHITECTURE OF THE DAMPER

Here the controller developed to improve the vehicle performances aims at controlling the new semi-active SOBEN damper using a semi-active control strategy based on its real behavior.

3.1 Control strategy

The overall control architecture is presented on Figure 3. S_1 is the controlled model and includes the model of the $\frac{1}{4}$ car given by Equation 3, the nonlinear identified damper model given by Equation 1 and the model of the servomechanism given by Equations 2. S_2 is the observer designed by [16] and presented in Section III-B. This observer estimates the state variables $x = [z_s - z_{us}, \dot{z}_s, z_{us} - z_r, z_{us}]^T$ of the $\frac{1}{4}$ vehicle. The LPV static state-feedback force controller S_3 receives the observed state variables x as an input and gives the required damping force in order to improve the vehicle performances.

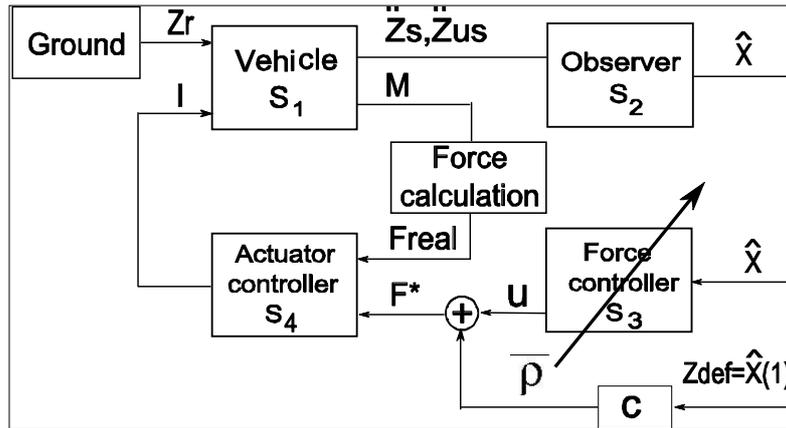


Fig. 3 Control architecture

This controller is scheduled by the parameter ρ that increases or decreases the performances required, in such a way that the required force F^* is always semi-active and adapted to the actuator abilities. The controller S_4 computes the proper servo mechanism input current allowing the damper to provide the required damping force F^* . This controller needs the real damping force F_{real} which is obtained using some measurements M and through a calculation procedure. This part is confidential due to patented results.

3.2 State observer

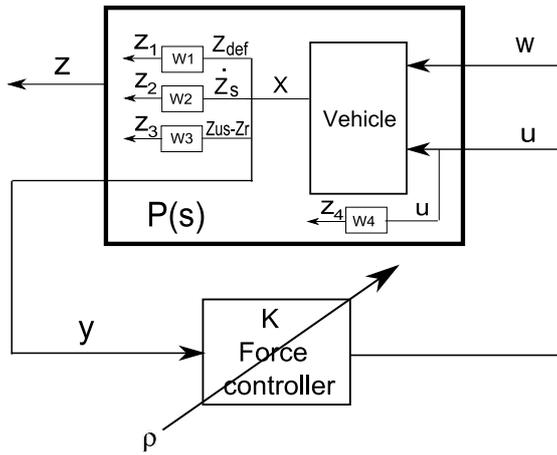
A disturbance decoupled nonlinear state observer ($\hat{x} = [z_s - z_{us}, \dot{z}_s, z_{us} - z_r, z_{us}]^T$) for a semi-active suspension was designed by [17] assuming the measurements of sprung and unsprung mass accelerations. They are available in the application considered here.

3.3 H_∞ /LPV force controller

The H_∞ approach is interesting to tackle frequency specifications. Here, the objective is to minimize the 4 transfer functions $\ddot{z}_s / z_r, z_s / z_r$ (comfort) $z_{us} / z_r, z_{def} / z_r$ (roadholding) at given frequencies, see [3] for details. LPV techniques can be used to schedule the controller according to measured varying parameters. It was used in [6], [9] to adapt the performance specifications and to improve the controlled system's robustness [16]. The solution proposed here aims at improving the 4 performances using an H_∞ /LPV controller with varying performance specifications. This completes the preliminary results [9]. The controller was synthesized using a linear $\frac{1}{4}$ car and damper model. The scheduling parameter is computed according to the difference between the real damping force and the force the damper can actually provide, on the basis of identified models. This solution allows the controller S_3 , given in Figure 3, to compute a realistic and semi-active required force that the damper is able to provide, using an identified damper model. The performance objectives are adapted on-line to the damper abilities. The required force received by the actuator controller S_4 as an input is $F^* = u + C \cdot \dot{z}_{def}$, where the damping rate C can be seen as the average damping rate of the damper, and u as the added energy to achieve the varying performance, computed by the H_∞ /LPV

force controller. The generalized parameter dependent plant $P(\rho)$ considered for the synthesis is given by Figure 4 and the equation below:

$$P(\rho): \begin{pmatrix} \dot{x} \\ z \\ y \end{pmatrix} = \begin{pmatrix} A(\rho) & B_1 & B \\ C_1 & D & E \\ C & F & O \end{pmatrix} \begin{pmatrix} x \\ w \\ u \end{pmatrix}$$



$$\begin{aligned} W_1(s)(\rho) &= \frac{z_1}{z_{def}} = (\rho_{max} - \rho) \cdot \frac{G_1}{\frac{s}{2\pi \cdot f_{c1}} + 1} \\ W_2(s)(\rho) &= \frac{z_2}{z_s} = (\rho_{max} - \rho) \cdot \frac{G_2}{\frac{s}{2\pi \cdot f_{c2}} + 1} \\ W_3(s)(\rho) &= \frac{z_3}{z_{ur}} = (\rho_{max} - \rho) \cdot \frac{G_3}{\frac{s}{2\pi \cdot f_{c2}} + 1} \\ W_4(s)(\rho) &= \frac{z_4}{u} = (\rho_{max} - \rho) \cdot \frac{G_4}{\frac{s}{2\pi \cdot f_{c2}} + 1} \end{aligned} \quad (4)$$

Fig. 4 Generalized plant and weighting functions

where the state variables vector $x = [x_{quarter}, x_{weighting}]^T$ includes the state variables vector of the $\frac{1}{4}$ car (3) and the state variables vector of the weighting functions performance specifications). $z = [z_1, z_2, z_3, z_4]$ are the weighed performance outputs to minimize, $z = \hat{x}$ is the observed state variables of the $\frac{1}{4}$ car model, $w = z_r$ is the ground variation and $\rho \in [0, \rho_{max}] = [0, 1]$ is the varying parameter used to schedule the controller. The weighting functions given by Equation 4, where $fc_1 = 1\text{Hz}$, $fc_2 = 20\text{Hz}$, $G_1 = 1$, $G_2 = 2$, $G_3 = 1$ and $G_4 = 1$, include the performance specifications detailed in [3].

$$\begin{cases} F_s^1 = A_1 \tanh(A_2 \dot{z}_{def} + A_3 z_{def}) + C_{min} \dot{z}_{def} + A_5 z_{def} \\ F_s^2 = A_1 \tanh(A_2 \dot{z}_{def} + A_3 z_{def}) + C_{max} \dot{z}_{def} + A_5 z_{def} \end{cases}$$

$$\begin{cases} F_s^{min} = \min(F_s^1, F_s^2) \\ F_s^{max} = \max(F_s^1, F_s^2) \end{cases}$$

$$F^* = \min(\max(F^*, F_s^{min}), F_s^{max})$$

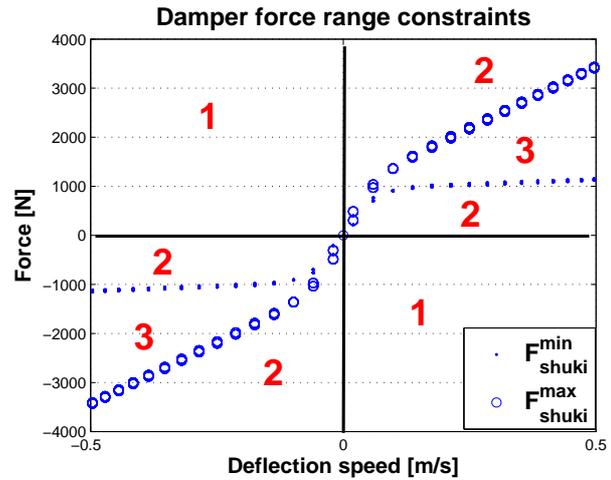
$$\rho = \frac{\min(\epsilon_{max}, |F_{real} - F^*|)}{\epsilon_{max}} \in [0, 1]$$

(5-8)

The weighting functions W_1 and W_2 have been chosen to minimize the accelerations and vertical movements of the sprung mass in order to improve the comfort of the

vehicle. W_3 and W_4 aim at reducing the tire and suspension deflections in order to reach a better roadholding. A scheduling strategy is proposed in [9] to avoid active forces by changing the weighting function of the control signal u . Here, the four weighting functions are scheduled by the parameter ρ which allows the performance objectives to be decreased if ρ is small, and increased if ρ is high. This parameter is computed according to (5-8), where ε_{\max} is a given maximal force error. In [9], the actuator constraints are only two extremal linear damping rates. Here ρ is evaluated using the identified damper model presented in Section II in order to determine the upper and lower reachable forces obtained with the extremal outputs of the actuator: the extremal damping rates C_{\min} and C_{\max} . The reachable force range of the damper is represented on the Figure below.

Zone 1 is active and unreachable, Zone 2 is semi-active but unreachable and Zone 3 is the reachable damper force range. The minimum and maximum of the extremal forces F_s^1 and F_s^2 computed in (6) are determined with (7) and used as limits for the saturation of the required force F^* given in (8). Therefore this saturated required force is a reachable force reference. Then ρ is computed with (9): $\rho = 0$ if $F_{\text{real}} = F^*$, $\rho = 1$ if $|F_{\text{real}} - F^*| > \varepsilon_{\max}$ and ρ is proportional to the force error if $|F_{\text{real}} - F_r| < \varepsilon_{\max}$. If $\rho = 0$ the weighting functions have small gains and the specified performances are the lowest. If $\rho = 1$ they are the highest. This solution allows the controller to decrease the performance objectives if the damper is not able to provide the required force. The controller $K(\rho)$ synthesized is a LPV static state-feedback. Therefore with $u = K(\rho) x = K(\rho) [z_{\text{def}}, z'_{\text{s}}, z_{\text{us}} - z_{\text{r}}, z'_{\text{us}}]^T$, the closed-loop system is given by:



$$CL(\rho) = \begin{pmatrix} \dot{x} \\ z \end{pmatrix} = \begin{pmatrix} A(\rho) + BK(\rho) & B_1 \\ C + EK(\rho) & D \end{pmatrix} \begin{pmatrix} x \\ w \end{pmatrix}$$

The H_∞ problem consists in minimizing, or bounding to a given γ_∞ level, the system gain between $\|w\|_2$ and $\|z\|_2$ (L_2 to L_2 induced norm). The solution of this problem is given by the Bounded Real Lemma extended to LPV systems and the objective is to minimize γ_∞ such that:

$$X = X^T > 0, U = KX$$

$$\begin{pmatrix} XA(\rho)^T + U^T B + A(\rho)X + BU & B_1 & XC_1^T + U^T E^T \\ B_1^T & -\gamma_\infty^2 I & D^T \\ C_1 X + EU & D & -I \end{pmatrix} < 0$$

where the decision variables are X and U .

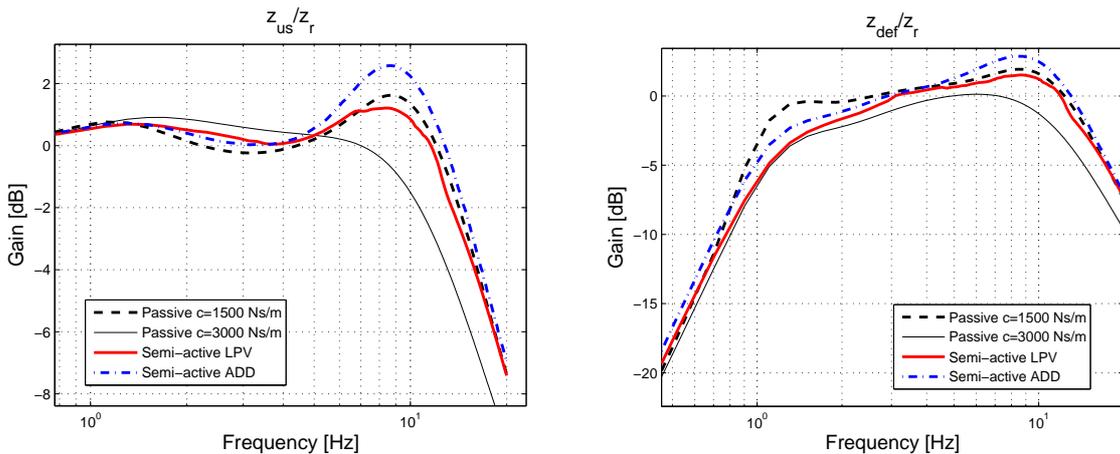
This inequality contains a parameter $\rho \in [\rho_{\min}, \rho_{\max}]$. Therefore this infinite set of LMIs established in [10], [11] has to be solved. The polytopic approach detailed in [2]

consists in finding the unknown matrices X , U and a scalar γ_∞ that solve a finite set of LMIs. This ensures the quadratic stability of the closed-loop system using a single Lyapunov function through the evaluation of the previous LMI at each corner of the polytope only. This polytope is defined by the extremal varying parameters. Then the LPV controller is a linear combination of the controllers computed at each corner. Here there is only one parameter: $\rho_{\min} < \rho < \rho_{\max}$. Thereafter, the controller is given by : $K = \rho K_{\min} + (1 - \rho) K_{\max}$ where ρ_{\min} and ρ_{\max} define the corners of the polytope.

The actuator controller S4, represented on Figure 3, is a simple PID controller that computes the servomechanism input current I .

4. SIMULATIONS

In this section, some simulation results are given and show the interest of the semi-active control proposed in this paper. The $\frac{1}{4}$ car model given by Equation 3, and the model of the damper with the actuator given by Equations 1-2 are used as a reference model for the following simulations. Here the performances obtained are analysed using the pseudo-Bode diagrams presented on Figure 5. The methodology to compute these diagrams is detailed in [3], [8]. On Figure 5, a passive linear damper with low damping rate: $C = 1500\text{Ns/m}$ (P1), high damping rate: $C = 3000\text{Ns/m}$ (P2), semi-active damper with LPV control proposed in Section 3 and semi-active damper with ADD control (Acceleration Driven Damper) are compared. The ADD semi-active control uses the measurements of the sprung mass acceleration and of the deflection. This control law detailed in [14] has been used in this paper for comparison. The comfort level of the vehicle has been increased by the controller proposed in this paper and by the ADD controller, but the roadholding is better with the LPV solution. The results are very satisfying when compared to the passive dampers. They are also better than the semi-active ADD comfort oriented control which degrades the roadholding.



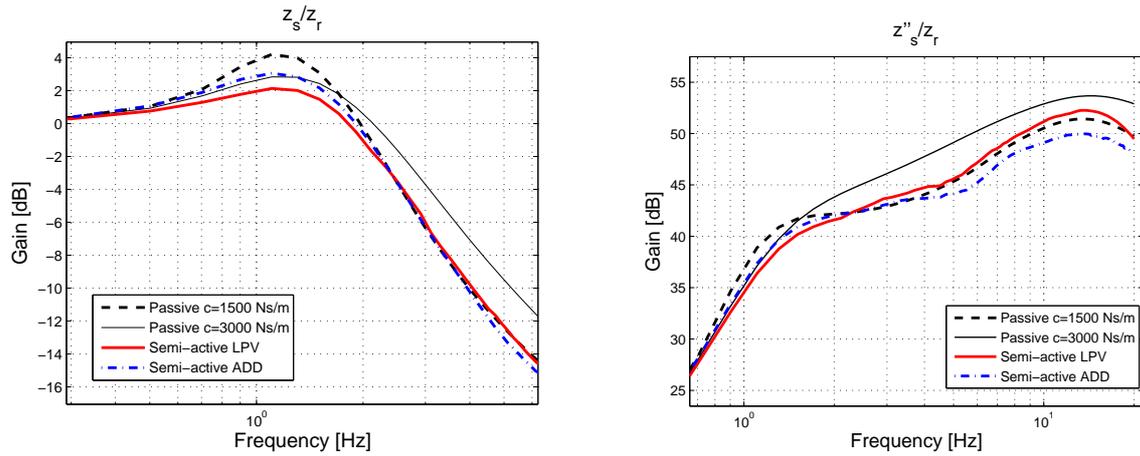


Fig. 5 Frequency results

Consider now some time results presented on Figure 6 where the $\frac{1}{4}$ car model is submitted to a given random ground profile.

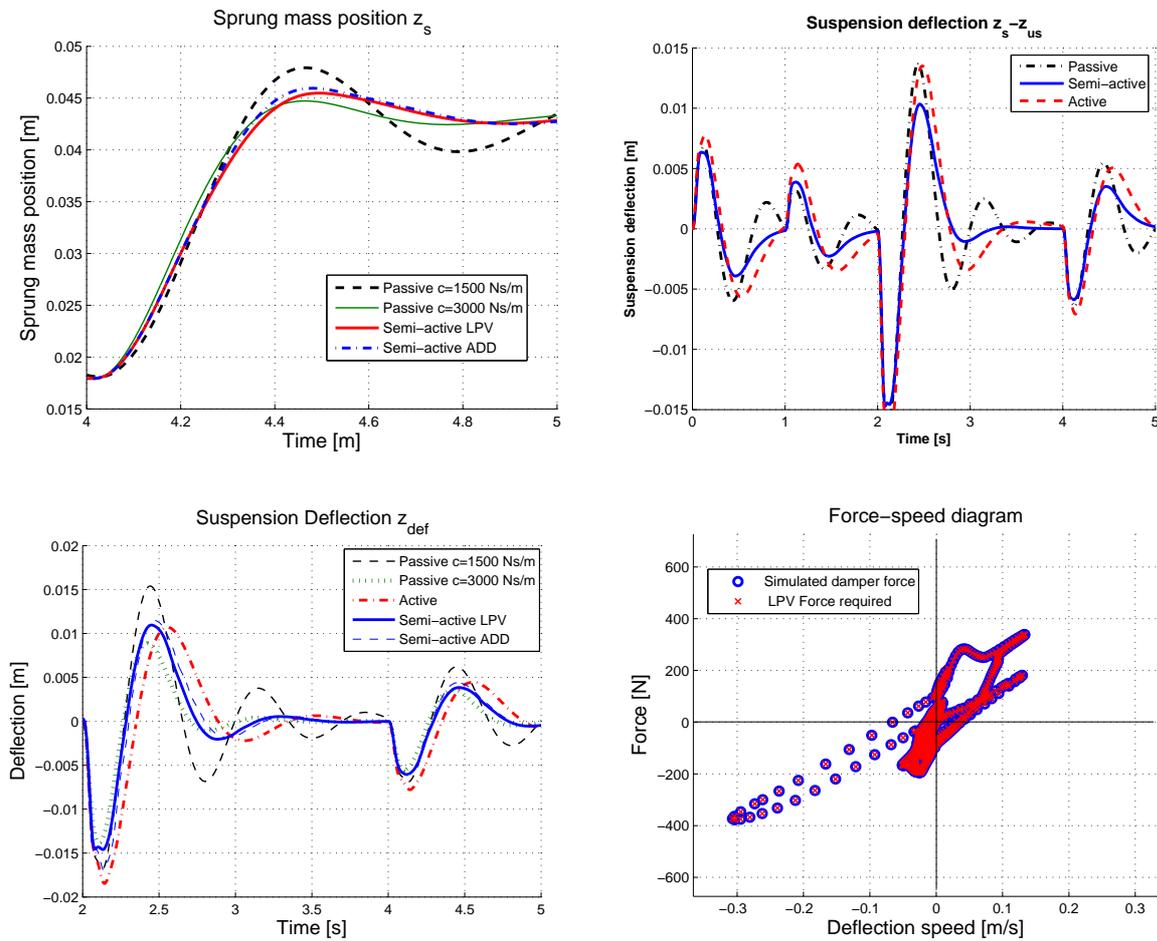


Fig. 6 Time results

Time and frequency results are equivalent. Figure 6 (force speed diagram) shows that the required force and the force actually provided are the same. It means on the one hand that the required force was semi-active and on the other hand that this force was reachable. The performance of the force controller is also illustrated with the results of Figure 7.

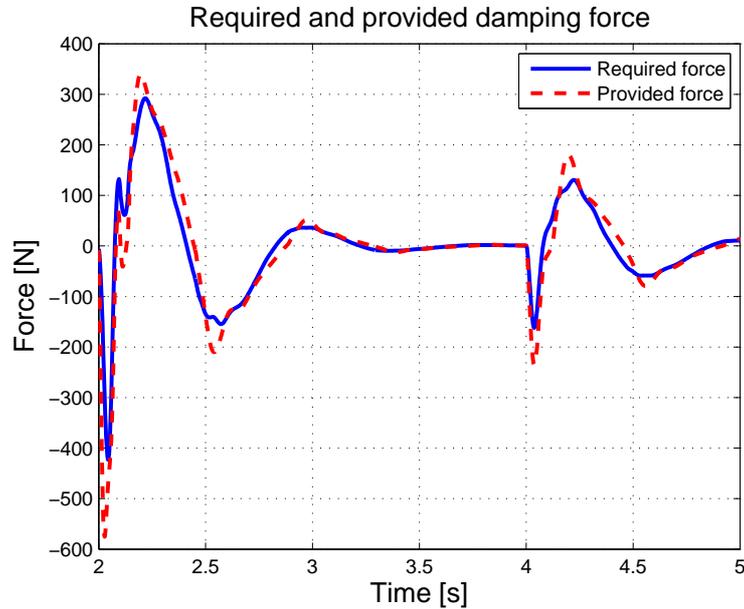


Fig. 7 Damping force control

5. CONCLUSIONS AND FUTURE WORKS

In this paper, an identified nonlinear static model of the damper and of its actuator have been developed using experimental results. Then a H_∞ /LPV static state-feedback controller was synthesized using a linear $\frac{1}{4}$ car model to compute the required damping force that minimizes given performance criteria. As this control strategy leads to an active force which is unreachable with such a semi-active damper, a scheduling parameter has been introduced to avoid the required force reference to be active. This parameter allows the controller to decrease the performance objectives if the required force is not in the reachable force range given by the identified model. The abilities of the real damper are taken into account in the controller. Therefore the proposed control architecture includes a global control of the vehicle behavior and a local control of the real damping force based servomechanism. The results emphasize the performance improvement of the proposed control strategy in terms of comfort and safety. Future works will consist in implementing and testing this control strategy with SOBEN on a testing car.

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