A MODEL FOR DYNAMIC FEED-FORWARD CONTROL OF A SEMI-ACTIVE DAMPER

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ABSTRACT
A new nonlinear model for a semi-active damper which describes a broad variety of hysteresis effects is presented in this paper. The novel part of the model is that the few parameters can be identified easily and cost effective for the automotive industry. Despite the few amounts of parameters, the validation of the model shows, that it describes the behavior of a real semi-active damper with strong hysteresis effects very well. Additionally, the influence of the unknown parameters is discussed in a detailed parameter study. In order to consider the hysteresis in the current control, the model is embedded in a dynamic feed-forward control structure.

Keywords: Damper model, hysteresis, dynamic feed-forward control

1. INTRODUCTION

The fundamental objectives of a car suspension are the isolation from vibrations introduced by the road irregularities and the improvement of road handling by means of a spring and a damper element. The suspension system supports the weight of the vehicle, provides directional control during handling maneuvers and provides effective isolation of passengers from road disturbances. These goals are generally at odds, so that the tuning of parameters in the suspension design involves finding a compromise.

The limitations of passive suspension can be enhanced by mechatronic systems, which can ease the conflict of the objectives ride comfort, ride safety and limited suspension deflection. In the last years, fully active suspension systems have been intensively studied. However, because of high costs and high energy demand their application in production vehicles is limited. Instead, because of the relatively low requested power, semi-active dampers are primarily integrated, which offer performance advantages over passive devices, see e.g. Ahmadian (1999). These suspensions feature "smart" shock absorber, which can vary the damping characteristic depending on the control strategy, mainly skyhook based comfort or handling oriented control laws. The generated force follows the passivity constraints, thus no energy can be introduced into the system. For more information and an overview on the semi-active control design refer to Guglielmino et al. (2008) and Savaresi et al. (2010) and the references therein.

Because of the complex damper mechanical construction and the switching elements in the valve, the behavior is highly nonlinear (see Duym and Reybrouck (1998) and Savaresi et al. (2010)). To exploit the potential of modern semi-active dampers a detailed damper model based on the static characteristics is desired, in order to be able to incorporate static and dynamical effects, such as hysteresis. In some works an approximation of the damper dynamics has been considered introducing first order lag elements (see e.g. Koch et al. (2010)). In the literature several models have been proposed to capture the hysteresis effects: A survey is given in Visintin (1994) and in Sain et al. (1997). To damper modeling purposes the Bouc-Wen model is frequently adopted. More details can be found in Guglielmino et al. (2008) and in Sain et al. (1997) and the references therein. A review of several idealized mechanical models for electrorheological and magnetorheological dampers based on a Bouc-Wen model is presented in Spencer et al. (1997). The models found in the literature are not able to reproduce the behavior of the twin tube hydraulic semi-active damper with internal switching valve.

The aim of this work is to model the adopted electromechanical device with strong hysteresis effects due to the design specifications and the interaction between fluid and moving mechanical components (valve). The parameter identification of the model is kept easy and only the measurement procedure adopted to obtain the static characteristics are utilized in order to produce no additional costs in the development process of the automotive industry. A model which fulfills these requirements is presented and the parameter influence on the model behavior and on the hysteresis shaping is discussed.

The remainder of the paper is organized as follows: The physical effects due to the fluid dynamics, the electromechanical valve effects and the construction characteristics are introduced in Section 2. In Section 3 the static damper characteristics are presented and compared to the measured damper forces. Based on the resulting insights the hysteresis model is presented in the same Section together with the validation of the damper model. In Section 4 a model based force tracking control in order to reach the desired force considering the nonlinearities is presented. In Section 5 the measurements results are discussed.

2. PHYSICAL EFFECTS CAUSING THE NON-LINEAR BEHAVIOR

In this Section, according to Duym and Reybrouck (1998), Lang and Segel (1981) and White (1986), some hydro-mechanical and thermodynamical aspects, which are able to modify the damper response, are considered. According to models of physical phenomena already described in the literature, a damper model is presented, which considers the major physical aspects in order to explain the measurement data which is shown in the next Section. Comparing the measurement results to the model output, the model performance is considered as basis for decision making of introducing or overcoming a physical effect. Aspects like model costs and complexity, time calculation needed for the determination of parameters, computer memory requirements and numerical problems in the simulation have been considered and thus only relevant effects are included in the modeling.

A common way to describe the behavior of a damper is the static force velocity characteristic which is shown in Figure 1. But as it can be seen in measurements (see e.g. Figure 3), the static relation is not sufficient to completely describe the coherence between the relative velocity and the damping force. In order to reproduce the real damping behavior, the physical characteristics of the fluid are taken into account. The oil flowing through the valve on the piston rod and building up pressure in the compression and rebound chambers is considered compressible and its impact can be modeled by a spring with linear stiffness. The oil compressibility leads to a lag of the pressure build-up in the tube and a phase loss of the damper force especially visible in the high range of velocity measurements.

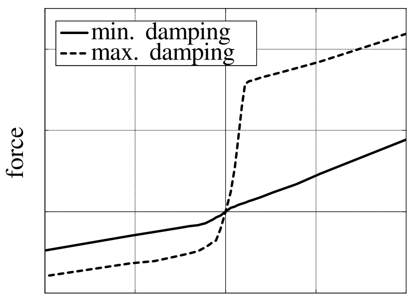

Figure 1: Static damper characteristic

As already analyzed in Duym and Reybrouck (1998) and Lang and Segel (1981), due to the damper design materials, the mechanical components are not perfectly rigid. Above all the rigidity of the cylinder walls has a direct effect on the oil pressure in the chambers. Their effect is similar to the oil compressibility and can be modeled as well as a spring. The two effects are considered together and will appear in the model as a spring component.

In addition to the mechanical friction also a hydrodynamical friction exits, which appears in elements moving in a viscous medium. This effect is known as Striebeck friction, Beitz and Küttner (1994). This kind of friction probably emerges between the valve mounted on the damper rod and the cylinder. Considering a standard Striebeck friction curve it can be observed, that the friction has a nonlinear dependency on the velocity and on the oil viscosity. Therefore, the amount of required parameters for modeling the friction is high and taking the minor effect on the damping force into account, friction is neglected for the presented model. Additionally, it has to be noted that the viscosity and density depend on the temperature in the chambers, which either has to be measured or estimated in order to consider these dependencies. As this would require high costs and efforts, it is assumed that the damper is heated up at a constant working temperature and thus temperature effects are also omitted.

The complete behavior of the damper depends not only on the characteristics of the fluid and the gas, but also on the mechanical switching element. By adjusting the rod valve, in order to change the damping coefficient of the damper, it can be noticed that the force behavior in this case also depends on the damper state. Defining the switching time, the time lapse in which 90% of the final force is reached it can be noticed that it varies between 10ms and 30ms. According to Heißing and Ersoy (2007), it depends on the absolute value of the damper velocities, on the rebound or compression direction as well as on the switching direction (from soft to hard or vice versa). This effect has to be implemented in the damper model because it describes the relation between the damper velocity (which is generally not the velocity of the oil through the valve assembly), the valve adjustment point (that means the current applied to the valve) and the resulting damping force. This phenomenon is strong nonlinear and very difficult to describe. In the presented model the switching time is approximated with a first order lag element for both directions and both switching directions.

Moreover, the structure of the inner valve assembly, which consists of different check valves, intake valves, port restrictions and blow-off valves (e.g. Duym and Reybrouck (1998)) is strongly nonlinear. The rod and the base valve assemblies are responsible for the static force velocity characteristic of the damper (Figure 1). In order to avoid switching noises of the valves, the pressure gradient is reduced by a damping element, which leads to a delay time while opening and closing. This effect causes nontypical hysteresis effects and peaks in the damper force, especially at high frequencies or high velocities. In order to improve ride and acoustic comfort, a special mount is attached between the damper and the chassis. This element is a flexible compound of metal and rubber and is designed in order to isolate the chassis from high frequency vibration induced by the damper (Heißing and Ersoy (2007)). However, this flexibility causes an additional stiffness, which also affects the damper force. As the measurement data used in this work has been obtained without damper mount, it is not considered in the model but it can easily be approximated by adjusting the upper stiffness ratio. The effect of the damper mount is that a spring-mass mechanical system is generated due to the fact that the damper rod is directly connected to the top mount. By non-optimal design parameters the system can be excited with high frequency vibration, which can deteriorate the damping action. Anyway, these frequencies are not in the range of interest, in which the suspension works. Therefore, the effect is not implemented in the proposed model.

Serveral major physical effects have been presented in this Section. However, there exist many other physical effects affecting the damper force (see e.g. Lang and Segel (1981), Reimpell and Stoll (1989)).

3. THE NONLINEAR DAMPER MODEL

3.1. Model equations

Figure 2 shows a semi-physical model for a semi-active damper, where the major physical effects for hysteresis that have been discussed in the previous Section, are considered. The stiffness of the cylinder walls, the damper mount and the fluid compressibility are substituted by a stiffness k and a damping factor b_l. This spring-damper element is connected in series with the main damper b_{nl}, where the static, nonlinear force-velocity characteristic (Figure 1) is considered.

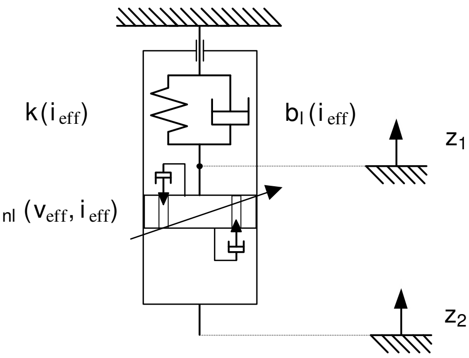

Figure 2: The nonlinear damper model

In order to take the opening time of the check valves into account, two relative velocities have to be distinguished. The velocity between the piston and the rod of the damper is generally used for calculating the damper force and will be denoted as $v_{pr} = \dot{z}_2 - \dot{z}_1$ in this work. As the check valves do not open instantly, the effective relative velocity on the main damper v_{eff} is slightly different. A first order lag element

$$v_{eff} = \frac{1}{T_{cv}s+1} v_{pr} \tag{1}$$

is suggested in order to approximate the coherence of these two velocities, with T_{cv} being the time constant of the check valves. This effect is symbolized by the dampers acting on the check valves in Figure 2. Also the dynamics of the current can be considered by a first order lag element

$$i_{eff} = \frac{1}{T_{cur}s+1} i_{des} \, , \tag{2}$$

with the time constant T_{cur}.

Balancing the forces and summarizing these equations leads to the differential equations for the nonlinear damper model

$$\dot{z}_1 = -\frac{k}{b_l + b_{nl}} z_1 + \frac{b_{nl}}{b_l + b_{nl}} \dot{z}_2 \tag{3}$$

$$\dot{i}_{eff} = -\frac{1}{T_{cur}} i_{eff} + \frac{1}{T_{cur}} i_{des} \tag{4}$$

$$\dot{v}_{eff} = -\frac{1}{T_{cv}} v_{eff} + \frac{k}{T_{cv}(b_l + b_{nl})} z_1 + \frac{b_l}{T_{cv}(b_l + b_{nl})} \dot{z}_2 \tag{5}$$

where $\dot{z}_2$ and i_{des} are the inputs of the system and the damping force

$$F_{damp} = \frac{k b_{nl}}{b_l + b_{nl}} z_1 + \frac{b_l b_{nl}}{b_l + b_{nl}} \dot{z}_2 \tag{6}$$

is considered as the output of the system.

3.2. Parameter identification

As it is shown in Section 3.4, the hysteresis behavior can be influenced by the model parameters k, b_l, b_{nl} and T_{cur}. In order to guarantee the highest flexibility of the model for different valve currents and therefore hysteresis types, it is suggested to introduce a dependency of the named parameters on the effective current i_{eff}. The damping coefficient of the main damper b_{nl} additionally depends on the effective relative velocity v_{eff} and is determined using the static force velocity characteristic of the damper (Figure 1). Therefore, it is considered to be known. The rise time of the current T_{cur} depends on the inductivity, the resistance and the controller parameters of the electrical circuit (Savaresi et al. (2010)). It can be either calculated if the parameters are known or read from a measured step response of the electrical system. The remaining parameters k, b_l and T_{cv} are estimated by optimization such that the difference between the output of the model, i.e. the damping force, O_{model} and the measured output O_{meas}, which can be expressed in the cost function

$$J(k, b_l, T_{cv}) = \frac{1}{n} \sum_{j=1}^{n} (O_{\text{model}}(t_j, \dot{z}_2, i_{des}, k, b_l, T_{cv}) - O_{\text{meas}}(t_j)) \tag{7}$$

is minimized for a fixed valve current i_{des} and a given velocity input $\dot{z}_2$ for all considered time steps t_j. The model parameters are then interpolated linearly for currents which have not been considered in the parametrization process.

For a cheap parametrization of the model in industrial application, it is desirable to use no additional measurements than those, which are used to obtain the static characteristic of the damper. The static behavior is determined by exciting the damper by a sinusoidal signal with a fixed stroke amplitude and several predefined velocity amplitudes for fixed valve currents. The damping force is measured for the point, where the velocity reaches its maximum what leads to one point in the force velocity diagram (Figure 1). For further information on the routine the reader is referred to Reimpell and Stoll (1989). In order to excite the hysteresis behavior of the damper, it is suggested to modify the stroke amplitude from 0.05 m to 0.01 m, what can be achieved easily by today's damper test rigs. The values for the amplitudes and the resulting frequencies are listed in Table 1.

Table 1: Input signal for parametrization according to Reimpell and Stoll (1989) and modified stroke amplitude

Stroke[m]	0.01				
Velocity [m/s]	0.052	0.131	0.262	0.393	0.524
Frequency[Hz]	0.83	2.08	4.17	6.25	8.34

By using the whole signals for the minimization of (7) instead of only one point of each data set, the parameters k, b_l and T_{cv} can be estimated. As this measurement data is sufficient, the requirement of producing no additional costs for the parametrization is fulfilled by this procedure.

3.3. Model validation

The model parameters are identified for the semi-active damper using the input signals according to Table 1. In order to validate the parameters, the model output is compared to measurements for different amplitudes and frequencies of the sinusoidal input signal (Table 2).

Table 2: Input signal for validation

Stroke[m]	0.005				
Velocity [m/s]	0.052	0.131	0.262	0.393	0.524
Frequency[Hz]	1.66	4.17	8.34	12.51	16.67

The result of this comparison can be seen in Figure 3 for the maximum damping, in Figure 4 for medium damping and in Figure 5 for the softest damping.

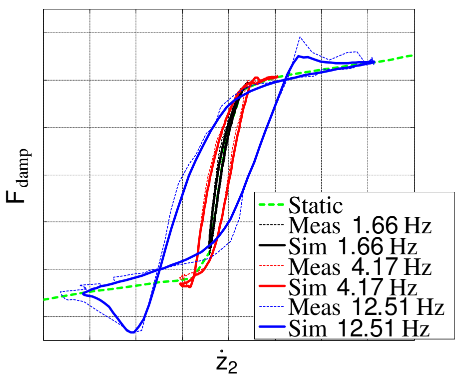

Figure 3: Fit to experimental data, $i_{des} = 0A$

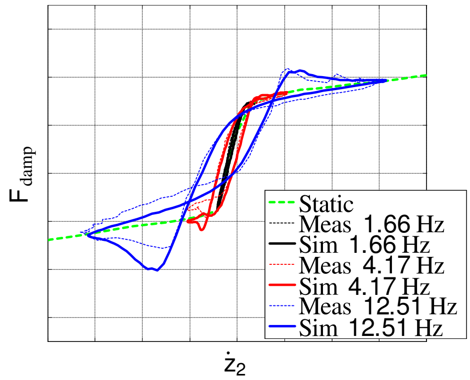

Figure 4: Fit to experimental data, $i_{des} = 0.9A$

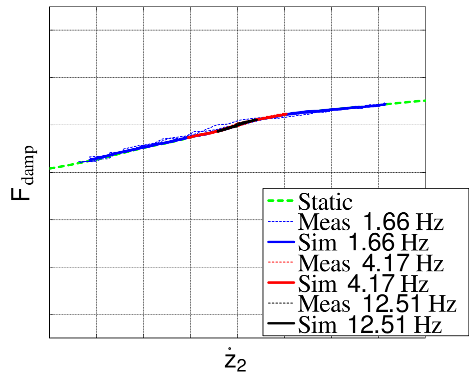

Figure 5: Fit to experimental data, $i_{des} = 1.8A$

Table 3: Fit values of the model in [%]

Frequency [Hz]	1.66	4.17	8.34	12.51	16.67
0A	84.8	95.4	91.7	91.1	86.4
1A	90.5	86.9	90.2	90.7	87.9
1.8A	85.1	92.6	90.9	90.5	86.9

The model output is compared in regions, where the damping forces nearly match the static characteristic (low velocities and/or soft damping) and also for increased velocities and damping forces, where minor (Figure 3, 4.17 Hz) and major (Figure 3, 12.51 Hz) hysteretic effects appear. It can be seen, that, in all cases, the proposed model matches the measurement data very well. In order to quantify the validity of the model, the fit value

$$fit = 1 - \frac{\left\| F_{meas} - F_{sim} \right\|_{rms}}{\left\| F_{sim} \right\|_{rms}} \tag{8}$$

for each considered frequency and current is given in Table 3, whereas a mean fit value of 89.4 % can be calculated.

3.4. Parameter variation

Depending on the set of parameters, a broad variety of hysteretic effects can be described using the new damper model. Figure 6 shows simulation results for three different parameter sets which result in minimal hysteresis, common hysteresis and hysteresis with an additional loop.

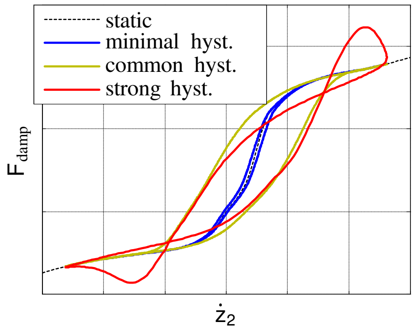

Figure 5: Model possibilities at 0 A and 16.67 Hz

The corresponding parameters are listed in Table 4 and it can be seen, that the hysteretic effects can be increased by increasing the response time of the check valves and reducing the stiffness and damping of the upper mount.

Table 4: Parameters for Figure 6

	k	b_l	T_{cv}
minimal hysteresis	9e7	9e4	2e-5
common hysteresis	5e6	5e3	1e-4
strong hysteresis	4e6	2e3	2.2e-3

In order to understand how the parameters k , b_l and T_{cv} affect the hysteresis behavior in the force velocity diagram, parameter variations have been made and the result is depicted in Figure 7. It can be seen, that the parameters k and b_l have an effect on the general hysteresis shape of the curve. Rising values of k or b_l lead to a smaller area which is enclosed by the damping curve. This seems plausible, because increasing stiffness and damping parameters stiffen the spring-damper element, which is connected in series with the nonlinear static damping element (Figure 2) and therefore, the influence of the static damper becomes higher. Most of the effects are influenced equally by k and b_l , except for the loop in the compression case, where slight differences can be seen. The size of both loops can be manipulated by the time constant of the check valves T_{cv} . Because the influence of the first order lag element (1) is getting smaller with lower time constants, the loop size decreases for faster response times.

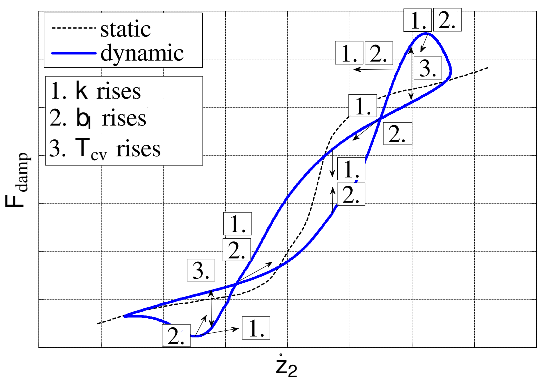

Figure 7: Influence of the model parameters

4. MODEL BASED CURRENT CONTROL

4.1. Dynamic feed-forward control

As seen in the previous Sections, the inversion of the static damping characteristics for a given desired damping force may lead to an improper resulting force due to the hysteresis of the damper. The previously described hysteresis model can be used to consider this nonlinear, dynamic behavior for the calculation of the appropriate valve current by embedding the model in a control structure, based on a dynamic feed-forward control structure, which has been proposed in Franklin et al. (2010) and is shown in Figure 8. It can be seen, that the static inversion of the damping characteristic, which is calculated in the feed-forward block in Figure 8, is extended by a controller, where the current is adjusted whenever an error between the desired and the calculated damping force occurs. The sign of the calculated force is multiplied with the force error because a positive error must lead to an increased damping, regardless of whether the damper is in compression or rebound. The corrected valve current is then applied to the real damper leading to a lower error between the desired force and the resulting force of the real damper. Of course, for proper tracking, it is required that the model matches the behavior of the real damper and that the dynamic feed-forward control is fast enough, i.e. the controller gain is high. A good performance can already be achieved by a proportional gain, which is also used in the present implementation. The gain is determined by optimization using a genetic algorithm such that the error between the desired force and the damping Force F_{damp} is minimized.

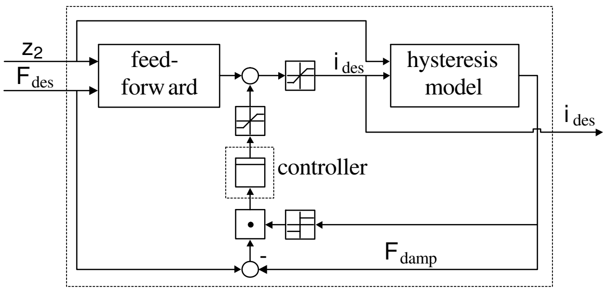

Figure 8: Dynamic feed-forward control structure

4.2. Numerical results

The dynamic feed-forward control approach is compared to the static calculation of the damper current using the hysteresis damper model for the simulation of the real damper. The desired force has been chosen to be proportional to the relative velocity, which has been measured from a real road profile. It is noted, that of course only desired forces, which are between the minimum and maximum characteristic of the damper, can be tracked. The results are shown in Figures 9 and 10 and it can be seen, that the error between the damper force and the desired force can be reduced significantly using dynamic feed-forward control. Especially the force peaks, which lead to high vertical accelerations and therefore are sensed most by the passenger, are tracked more accurately. The root mean square value for the force error is reduced by 38 % from 125.5 N to 77.2 N by using dynamic feed-forward control instead of the static inversion for this example. In order to estimate the robustness of the controlled system, the parameters of the simulated real damper have been varied by 30 %. The results show, that there is still an improvement of the rms value for the force error of 18.9 % for the variation of k, b_l, b_{nl} and T_{cv} in the positive direction and 5.6 % in the negative direction. A more detailed analysis of the robustness will be part of future work.

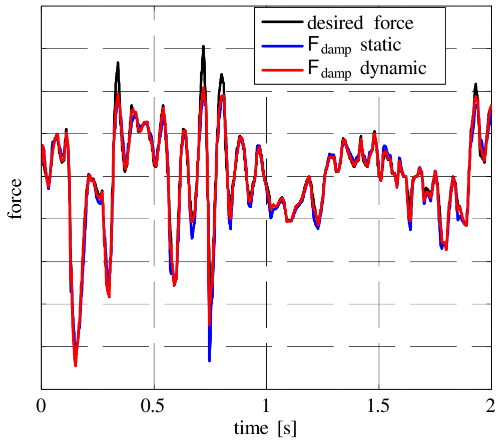

Figure 9: Comparison of absolute forces between static and dynamic current calculation

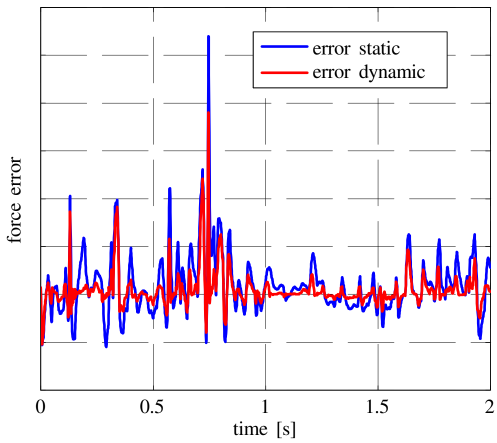

Figure 10: Comparison of force error between static and dynamic current calculation

5. CONCLUSION

In this paper, a novel hysteresis model for a semi-active damper has been presented, which can be used to describe a broad spectrum of dampers. The advantage of the model is that the few unknown parameters can be identified using only measurements, which are already made in the automotive industry in order to obtain the static characteristic of the damper. The identification process has been shown for a real damper with strong hysteresis effects and the validation showed that the model matches the measurement data very well. Furthermore, a study on the parameter variation has been presented and the influence on each parameter on the shape of the hysteresis curves has been discussed. Finally, it has been shown that the model can be used in order to improve the force tracking of the semi-active damper by embedding it into a dynamic feed-forward control structure.

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