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ON THE PERFORMANCE OF HYBRID ACTIVE-PASSIVE DAMPING TREATMENTS MECHANISMS FOR VIBRATION CONTROL OF BEAMS USING ADAPTIVE FEEDFORWARD STRATEGIES

C. M. A. Vasques[1] and J. Dias Rodrigues
Faculdade de Engenharia da Universidade do Porto
Departamento de Engenharia Mecânica e Gestão Industrial
Rua Dr. Roberto Frias s/n, 4200-465 Porto, Portugal
Email: (1)cvasques@fe.up.pt

SYNOPSIS

This paper concerns the adaptive feedforward control of vibration of a freely supported beam with two distinct surface mounted hybrid active-passive damping treatments. The first configuration concerns the use of an *Active Constrained Layer Damping* (ACLD) patch alone, where the piezoelectric constraining layer is actively utilized to increase the shear deformation of the sandwiched passive viscoelastic layer and at the same time to apply forces and moments into the structure, which will balance the power flows into the structure, and is denoted by ACLD configuration. The second configuration regards the use, as an active element in the control, of a piezoelectric patch alone, denoted by *Active Damping* (AD), and since the constraining layer of the ACLD treatment also bonded on the other side of beam is not actively utilized, a *Passive Constrained Layer Damping* (PCLD) treatment is utilized in combination with AD, and an AD/PCLD configuration is considered. A finite element (FE) model of the beam with the damping treatments is used for the simulation of the adaptive feedforward controller which is also tested in real-time. The aims are to compare the predicted and measured damping performances of the two treatments, in terms of vibration reduction, control effort, stability and robustness when a filtered-reference LMS algorithm is used to cancel the effects of a broadband voltage disturbance applied into a third surface mounted piezoelectric patch which is used to excite the beam.

INTRODUCTION

Over the past decades active control in conjunction with hybrid active-passive damping treatments has been widely applied to attenuate structural vibration and sound radiation from structures. The principle behind active vibration control is being able of producing some kind of secondary control action upon the structure that will cancel the effects of a primary excitation (disturbance). Hybrid active-passive damping treatments are characterized by a blend of piezoelectric and viscoelastic damping capabilities. Reviews, surveys and assessments on hybrid active-passive damping treatments and related technologies can be found, for example, in Park and Baz (1999b), Benjeddou (2001), Trindade and Benjeddou (2002) and Stanway et al. (2003).

Active Damping (AD) treatments comprise only piezoelectric material patches/layers. Usually they are surface mounted (or embedded) on a structural system and produce distributed forces and/or moments upon the structure which originate "negative" power flows to balance the effects of the primary disturbance(s). Some of the pioneer works concerning AD are the ones presented by Bailey and Hubbard (1985) and Crawley and de Luis (1987) in the 1980s, and over the following decades several works appeared in the open literature with further developments (see, for example, Collins et al., 1994; Sunar and Rao, 1999; Saravanos and Heyliger, 1999; Vasques and Rodrigues, 2005; Vasques and Dias Rodrigues, 2006c).

Passive damping treatments comprise viscoelastic layers sandwiched (or not) between a host structure and an elastic constraining layer. They add damping to the system due to heat dissipation mainly caused by the shearing of the viscoelastic layer. If an elastic constraining layer is considered, which makes the shear deformation become dominant to the detriment of the extensional one, the configuration is usually known as *Passive Constrained Layer Damping* (PCLD). However, in order to further increase the shearing and energy dissipation of the viscoelastic layer, an active constraining piezoelectric layer may be used to actively increase the shearing in a convenient way. That was originally proposed by Plump and Hubbard (1986) and later further developed and widely studied by Baz and his co-workers (e.g., Baz, 1993; Ray and Baz, 1997; Park and Baz, 1999a) and many others. These treatments are the so-called *Active Constrained Layer Damping* (ACLD) and combine the passive capabilities (fail-safe, robustness, simplicity) of viscoelastic materials to dissipate vibratory energy at high frequencies with the active capabilities (high performance, adaptability) of piezoelectric materials at low frequencies.

In the last decades the advances in digital signal processing and sensors and actuators technology have prompted interest in active control and a considerable effort has been put in the development and implementation of *Active Noise Control* (ANC) and *Active Vibration Control* (AVC) theories (see related textbooks: Inman, 1989; Meirovitch, 1990; Nelson and Elliott, 1993; Fuller et al., 1996; Hansen and Snyder, 1997; Clark et al., 1998; Elliott, 2001; Preumont, 2002). These might be divided into two fundamental classes, namely, *feedback* and *feedforward* control algorithms. The former control strategy has been shown to be most suitable in applications where the structure is under impulsive or stochastic unknown disturbances and the latter to the case where deterministic or correlated information about the disturbance is known. Variations of the two general methods exist, each with advantages, disadvantages and limitations. A review paper concerning active structural vibration control strategies is presented by Alkhatib and Golnaraghi (2003).

Studies concerning different control of vibration strategies (e.g., feedforward wave suppression, proportional and velocity feedback, optimal control) with different types of actuation (e.g., point forces, pair of moments, piezoelectric actuation) in different structural systems with hybrid active-passive treatments can be found (e.g., Baz and Chen, 2000; Trindade et al., 2001; Baz, 2001; Ray and Reddy, 2004). Comparisons of different classical and/or optimal feedback control strategies were performed, for example, by Gandhi and Munsky (2002) and Vasques and Dias Rodrigues (2006c).

Feedback theories have been vastly applied in the vibration control of structures with AD and ACLD treatments. Usually it is shown that ACLD treatments have better performances than AD ones, with higher damping values being obtained with lower control voltages. Works concerning adaptive feedforward control theory in AD applications were performed, for example, by Vipperman et al. (1993) and Vipperman and Burdisso (1995). However, only a few works utilizing ACLD treatments, concerning harmonic *Active Structural Acoustic Control* (ASAC) (Poh et al., 1996) or broadband (Illaire, 2004) and harmonic vibration reduction (Vasques and Dias Rodrigues, 2006b), might be found. The most typical application of feedforward control presented in textbooks concerns the noise attenuation in ducts and applications concerning vibration reduction are mainly devoted to ASAC. Furthermore, *hybrid* (combined feedback/feedforward) control strategies with AD treatments were successfully used in the past for noise and vibration suppression (see, for example, Saunders et al., 1993; Clark, 1995; Saunders et al., 1996; Man and Preumont, 1996). However, only one study concerning *hybrid* control with ACLD treatments applied to a beam was performed by Vasques and Dias Rodrigues (2006b).

The aims of this paper are to give new contributions to the understanding of how AD and ACLD treatments behave and to know which one is best and easily implemented in practice. To this end, a finite element (FE) model previously developed by the authors (Vasques et al., 2004, 2006a) is used for the simulation of an adaptive feedforward vibration control system using the filtered-reference LMS algorithm. A beam with AD and ACLD patch treatments, with the same dimensions and mounted on opposite sides, and excited by a piezoelectric patch, is considered. The two damping treatments are compared in their capacity to attenuate the effects of a broadband disturbance voltage applied into a piezoelectric patch. The treatments are compared and their damping performances assessed through simulated and experimental frequency response results.

MATHEMATICAL MODEL

When designing hybrid active-passive damping treatments it is important to know the configuration of the structure and treatment that gives optimal damping. Thus, for simulation, the designer needs a model of the system in order to define the optimal locations, thicknesses, configurations, control strategy, etc. However, the task of modeling structures with arbitrary ACLD treatments is not straightforward and requires the development of a piezo-visco-elastic coupled model of the host structure and damping treatments, comprising piezoelectric, viscoelastic and elastic materials, and different assumptions regarding the mechanical model, the damping introduced by the viscoelastic materials and the electro-mechanical coupling should be considered.

A FE solution of the mathematical model is usually employed in the simulation of the real system and a 1D FE model of beams with active-passive damping treatments is utilized in this work. The FE considers an arbitrary number of elastic, piezoelectric and viscoelastic layers attached to both sides of the beam, and a partial layerwise theory is used to define the displacement field. Furthermore, a fully coupled electro-mechanical theory is considered to model the piezoelectric layers, where a non-linear electric potential distribution is utilized. Regarding the viscoelastic damping behavior, the temperature and frequency dependent material properties of the viscoelastic materials increase the complexity of the underlying mathematical model. For simplicity, usually the temperature is assumed constant and only the frequency dependency is taken into account. In this work, the damping behavior of the viscoelastic layers is considered by a Laplace transformed *Anelastic Displacement Fields* (ADF) method. For the sake of brevity the mathematical model is not presented here and the reader is referred to Vasques and Dias Rodrigues (2006b), Vasques et al. (2006a) and Vasques et al. (2006b) for further details about the utilized coupled piezo-visco-elastic FE.

DAMPING MECHANISMS

Hybrid damping treatments attenuate vibrations and sound radiation through different damping mechanisms. In order to optimize the design of arbitrary hybrid active-passive damping treatments, where the damping layers are arbitrarily stacked and mounted on the host structure, it is essential to understand their phenomenological behavior and to be able to identify and quantify the efficiency of the different damping mechanisms (Vasques and Dias Rodrigues, 2006a). These treatments comprise, on the one hand, the damping effects due to the internal molecular interactions that occur during deformation, in general, and vibration, in particular, of the viscoelastic materials, which give rise to macroscopic properties such as stiffness and energy dissipation during cyclic deformation, and, on the other hand, the effects due to the piezoelectric actuation which applies forces and moments on the structure.

The damping achieved with ACLD treatments involves mainly three damping mechanisms: (1) the shearing of the viscoelastic layer in open-loop, (2) the increase of shearing in the viscoelastic layer due to the convenient motion of the active piezoelectric constraining layer, (3) and the decrease of total input power into the structure due to the forces/moments applied by the active piezoelectric constraining layer through the viscoelastic layer. Obviously, the latter mechanism for ACLD treatments has a reduced importance since the viscoelastic layer usually reduces the transmissibility of efforts to the host structure. In fact, usually the transmissibility (stiffness) increases with frequency. Thus, in order to dissipate energy also at low frequencies, the aim would be to actively increase the shearing in the viscoelastic layer. However, due to the loss factor behavior of viscoelastic materials (usually smaller at low frequencies), the treatment is usually more efficient for frequencies with higher loss factors, if we have also significant shearing strains. (The energy released is related with the loss factor times the shearing strain.) Regarding the AD treatment, the transmissibility of efforts to the host structure is higher, since the patch is bonded to the structure, and the aim is actively reducing the input power coming into the structure. Of course, some energy may also be lost due to the boundary conditions or the internal damping of the host structure, or even radiated into the ambient medium, which usually are neglected.

FEEDFORWARD CONTROL THEORY

Feedforward control system design and implementation rely upon digital filtering fundamentals and adaptive filter theory, which can be found in reference textbooks (see, for example, Widrow and Stearns, 1985; Elliott, 2001; Haykin, 2001). Contrarily to *feedback* systems, *feedforward* control assumes that one has information about the original (primary) excitation of the system and that a reference signal correlated with the excitation is available. The aims are then to produce a secondary excitation that will cancel the effects of the primary excitation at the chosen location(s).

In Figure 1 a schematic of the discrete-time (digital) generalized plant for a feedforward controller is presented. The *filtered-reference LMS* algorithm is the most widely accepted feedforward control algorithm because of its ease of implementation and remarkable performance. It is assumed that a detection system, which produces a signal correlated with the primary disturbance, exists (or not, and in that case the true excitation signal is utilized), and produces a reference signal $r(k)$ correlated with the excitation. This signal is then adaptively filtered to generate the necessary control action, u_ϕ^{ff}, to cancel the effect of the *primary excitation* (disturbance).

The control filtering process utilizes an adaptive *Finite Impulse Response* (FIR) filter whose ith coefficient at the kth sample time is $h_i(k)$. The filter output $u_\phi^{ff}(k)$ is obtained from

$$u_\phi^{ff}(k) = \sum_{i=0}^{N-1} h_i(k) r(k-i), \tag{1}$$

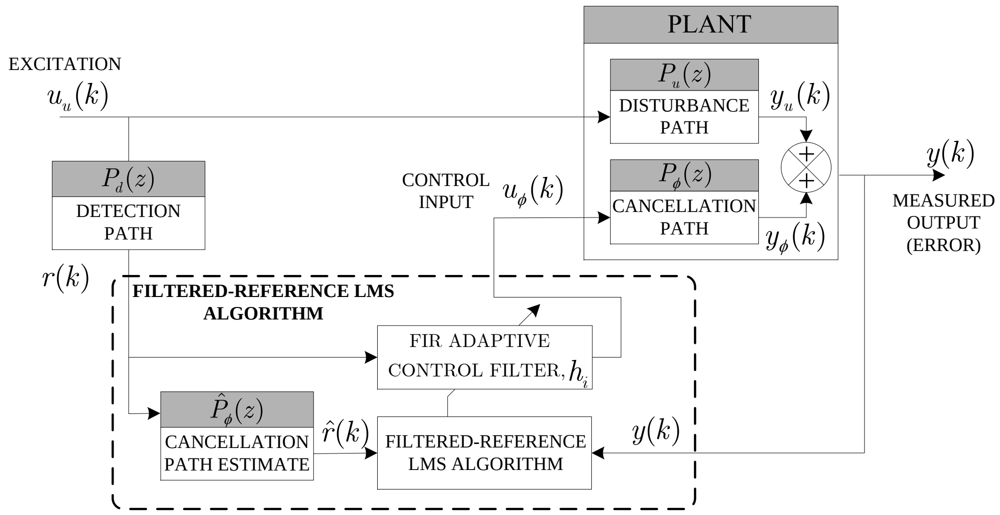

Figure 1: Schematic of the discrete-time (digital) generalized plant for a feedforward SISO controller.

where N is the number of filter coefficients. Then, the control signal has to pass through a part of the physical system before the error sensor measures the output. This physical path, $P_\phi(z)$, is called the *cancelation path* (or *error path*). Therefore, the output of the plant due to the feedforward control input only, $y_\phi(k)$, is given by

$$y_\phi(k) = \sum_{j=0}^{M-1} g_j \sum_{i=0}^{N-1} h_i(k) r(k-i-j), \tag{2}$$

where g_j is the discrete impulse response of the control input-to-output path $P_\phi(z)$, which is assumed to be of order M. Thus, the net output of the system, $y(k)$, can be written as

$$y(k) = y_u(k) + \sum_{j=0}^{M-1} g_j \sum_{i=0}^{N-1} h_i(k) r(k-i-j), \tag{3}$$

where $y_u(k)$ is the response due to the effects of the primary disturbance alone. However, the order of convolution can be interchanged without changing the result, yielding

$$y(k) = y_u(k) + \sum_{i=0}^{N-1} h_i(k) \bar{r}(k-i), \tag{4}$$

where

$$\bar{r}(k-i) = \sum_{j=0}^{M-1} g_j r(k-i-j). \tag{5}$$

By rearranging the convolution, a signal $\bar{r}(k-i)$ is created, which is to be estimated by the filtered-reference operation. If the true impulse response of the plant $P_\phi(z)$ is estimated by a FIR or *Infinite Impulse Response* (IIR) filter, then an estimate of $\bar{r}(k)$ is given as $\hat{r}(k)$.

The problem of how best to adapt the filter coefficients $h_i(k)$ can now be addressed. One alternative is to adapt the coefficients in order to minimize a *cost function* quadratically dependent of the output, $J = E[y^2(k)]$. However, as for the LMS algorithm (Widrow and Stearns, 1985), the instantaneous value of $y^2(k)$ is used as an estimate of the expected value of J. A simple gradient descent algorithm is thus guaranteed to converge to the globally optimal solution of the problem of minimizing the cost function. Such an adaptive algorithm can be written as

$$h_i(k+1) = h_i(k) - \mu \frac{\partial J}{\partial h_i(k)}, \tag{6}$$

where μ is a convergence coefficient. Thus, from the definition of the cost function the derivative in Equation (6) is written as

$$\frac{\partial J}{\partial h_i(k)} = 2y(k) \frac{\partial y(k)}{\partial h_i(k)}. \tag{7}$$

From Equation (4) the derivative of $y(k)$ with respect to $h_i(k)$ is simply $\bar{r}(k-i)$. If one uses the estimated filtered reference signal $\hat{r}(k-i)$, the steepest descent algorithm required to adapt the coefficients of the digital controller, given by Equation (6), can thus be written as

$$h_i(k+1) = h_i(k) - \alpha y(k) \hat{r}(k-i), \tag{8}$$

where $\alpha = 2\mu$ is another convergence coefficient parameter that determines the speed and stability of adaptation. The convergence properties of the *filtered-reference* LMS algorithm are similar to those of the normal LMS algorithm, whose properties are described in detail for example by Widrow and Stearns (1985).

EXPERIMENTAL AND SIMULATION SETUP

The filtered-reference LMS algorithm was tested on a freely supported aluminum beam measuring $380 \times 15 \times 2\,\mathrm{mm}$, with two piezoelectric patches of material PXE-5 (Phillips Components, PLT 30/15/1-PX5-N) measuring $30 \times 15 \times 1\,\mathrm{mm}$ mounted each one $52\,\mathrm{mm}$ away from the free ends, and an ACLD patch, with a constrained layer made of the same piezoelectric material and a viscoelastic layer 3M ISD112, measuring $30 \times 15 \times 0.127\,\mathrm{mm}$, mounted on the same location but on the opposite side (Figure 2). The piezoelectric patches were bonded on the beam by means of a nonconductive cianoacrilate adhesive (Loctite 496) with high rigidity and the ACLD patch was bonded through the use of the adhesive of the viscoelastic layer.

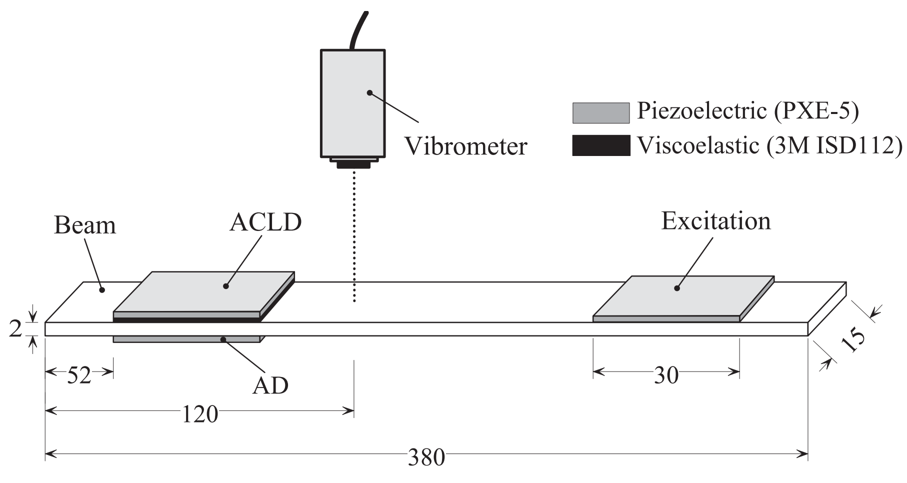

Figure 2: Freely supported test beam configuration (dimensions in mm) with the AD, ACLD and excitation patches.

The mechanical and electrical material properties of the aluminum, 3M ISD112 (3M, 1993) and PXE-5 are presented in Table 1 (see the standard IEEE, 1988, for further details about notation). The shear storage modulus and loss factor of the viscoelastic material at the 27 °C were defined by means of a three-series ADF model with parameters $G_0 = 0.5\,\mathrm{MPa}$, $\Delta = [0.743, 3.265, 43.284]$, $\Omega = [468.7, 4742.4, 71532.5]\,\mathrm{rad/s}$, which were determined by curve fitting experimental data in the frequency range $[20 - 5000]$ Hz (Trindade et al., 2000). Furthermore, the extensional storage modulus was obtained assuming a frequency independent Poisson's ratio equal to 0.45. For the simulation, the beam was discretized into 190 FEs and a coupled piezo-visco-elastic structural FE modal model with 20 elastic modes plus 3 rigid body modes was considered.

Table 1: Material properties of the aluminium, PXE-5 and 3M ISD112.

Aluminum		3M ISD 112		PXE-5			
E	$70\,\mathrm{GPa}$	G	–	c_{11}^E	$131.1\,\mathrm{GPa}$	d_{31}	$-215\times10^{-12}\,\mathrm{m\,V^{-1}}$
ν	0.3	ν	0.45	c_{12}^E	$7.984\,\mathrm{GPa}$	d_{33}	$500\times10^{-12}\,\mathrm{m\,V^{-1}}$
ρ	$2710\,\mathrm{kg\,m^{-3}}$	ρ	$1600\,\mathrm{kg\,m^{-3}}$	c_{13}^E	$8.439\,\mathrm{GPa}$	d_{15}	$515\times10^{-12}\,\mathrm{m\,V^{-1}}$
				c_{33}^E	$12.31\,\mathrm{GPa}$	$\varepsilon_{11}^T/\varepsilon_0$	1800
				c_{44}^E	$2.564\,\mathrm{GPa}$	$\varepsilon_{33}^T/\varepsilon_0$	2100
				c_{66}^E	$2.564\,\mathrm{GPa}$	ρ	$7800\,\mathrm{kg\,m^{-3}}$

A schematic of the experimental setup of the real-time feedforward controller is depicted in Figure 3. The velocity at one point of the beam was measured with a Doppler vibrometer transducer Polytec – OFV 303 with a laser vibrometer controller unit Polytec – OFV 3001. A Krohn-Hite dual-channel (model 3362) and single-channel (model 3550) low-pass filters with four-pole Butterworth characteristics with the cut-off frequency set to 1200 Hz were utilized as anti-aliasing and reconstruction filters. Furthermore, two power amplifiers were used, one LDS PA100E with a maximum output voltage of 20 V rms for the excitation, and another constructed in the lab with a maximum output voltage of 150 V for the control voltage. All real-time processing was carried out with a National Instruments Lab-PC+ DAQ board with 12-bit analog resolution, and the sampling rate was set to 3600 Hz both for simulation and real-time.

In the simulation and real-time implementation of the control system a *Single-Input Single-Output* (SISO) configuration, with the input being the velocity at one point of the beam and the output being the voltage applied into the piezoelectric constraining layer, for the ACLD configuration, or the piezoelectric patch, for the AD/PCLD configuration, is considered here. Assuming that the designer has access to a reference signal that is correlated with the disturbance, the filtered-reference LMS algorithm is employed to cancel the effects of the disturbance on the plant at the chosen output location. The simulation and all real-time processing were performed using *Matlab* and *Simulink* software, incorporating the *Real-Time Workshop* and the *Real-Time Windows Target*.

SYSTEM IDENTIFICATION

The first step in operating the control is to perform the off-line system identification. Thus, a band-limited random noise from 10 to 1200 Hz, which was generated with *Simulink*, was fed into the two piezoelectric patches of the AD and ACLD treatments and the FRFs between the

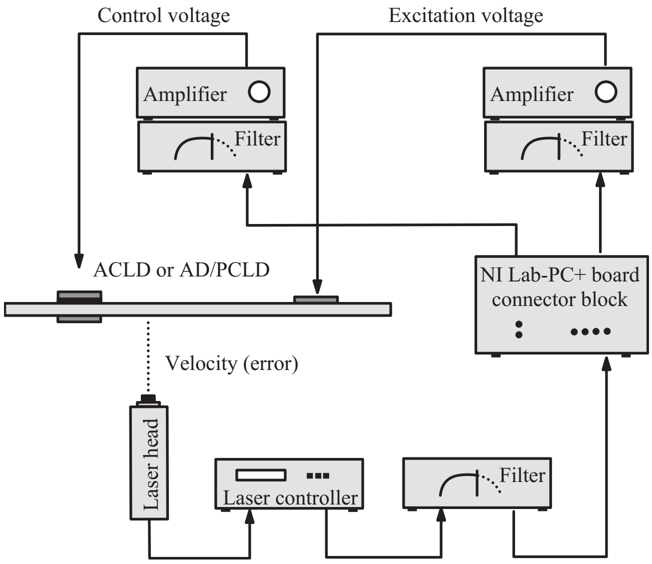

Figure 3: Schematic of the experimental setup of the realtime feedforward controller.

voltage applied and the measured velocity were estimated in *Matlab*. An IIR filter was used to identify the measured and simulated FRFs using the *Matlab* `invfreqz.m` routine and the identified coefficients were utilized as coefficients of the IIR discrete filters of the cancellation paths. The results are presented in Figure 4. Note that the identified FRFs of the discrete filters match the predicted and measured FRFs in magnitude and phase, with the exception of some parts of the phase of the measured FRFs, however presenting the same trend. The phase has been "unwrapped" so that the reader can see that the phase does not remain within a $[-90\,°, 90\,°]$ boundary and therefore the control loops are indeed nonminimum phase in the bandwidth of interest. The resulting nearly linear decreasing trend in the phase verified in the measured FRFs is due to a combination of the delays from the DSP system, analog filters, amplifiers, etc., and demonstrates the "non-causality" of the controller. Furthermore, it is also worthy to mention that the predicted FRFs match the measured ones for both AD and ACLD treatments, and that there is almost two orders of magnitude of difference between the actuating capacity of the AD and ACLD patches, where the viscoelastic layer decreases the transmissibility of efforts, mainly in the low stiffness range (low frequencies). That is evident in the measured FRF of the ACLD patch, were in the range from 0 to more or less $150\,\mathrm{Hz}$ (first mode) the measured results show a very low transmissibility.

RESULTS

All controlled measurements have been obtained with the adequate adaptation rates for the different damping treatments. A filtered-reference LMS algorithm with 50 adaptive coefficients was considered both in the simulation and real-time control. The time responses after convergence of the adaptive feedforward algorithm were saved and post-processed to obtain the measured and predicted FRFs of the open- and closed-loop control systems for the two damping configurations, namely AD/PCLD and ACLD, as presented in Figure 5. The open-loop FRFs show again a good match between the predicted and measured FRFs at all modes. It is shown that the closed-loop response with the AD/PCLD treatment is more effective than with the ACLD treatment. Both measured and predicted FRFs show that, and a similar trend in the

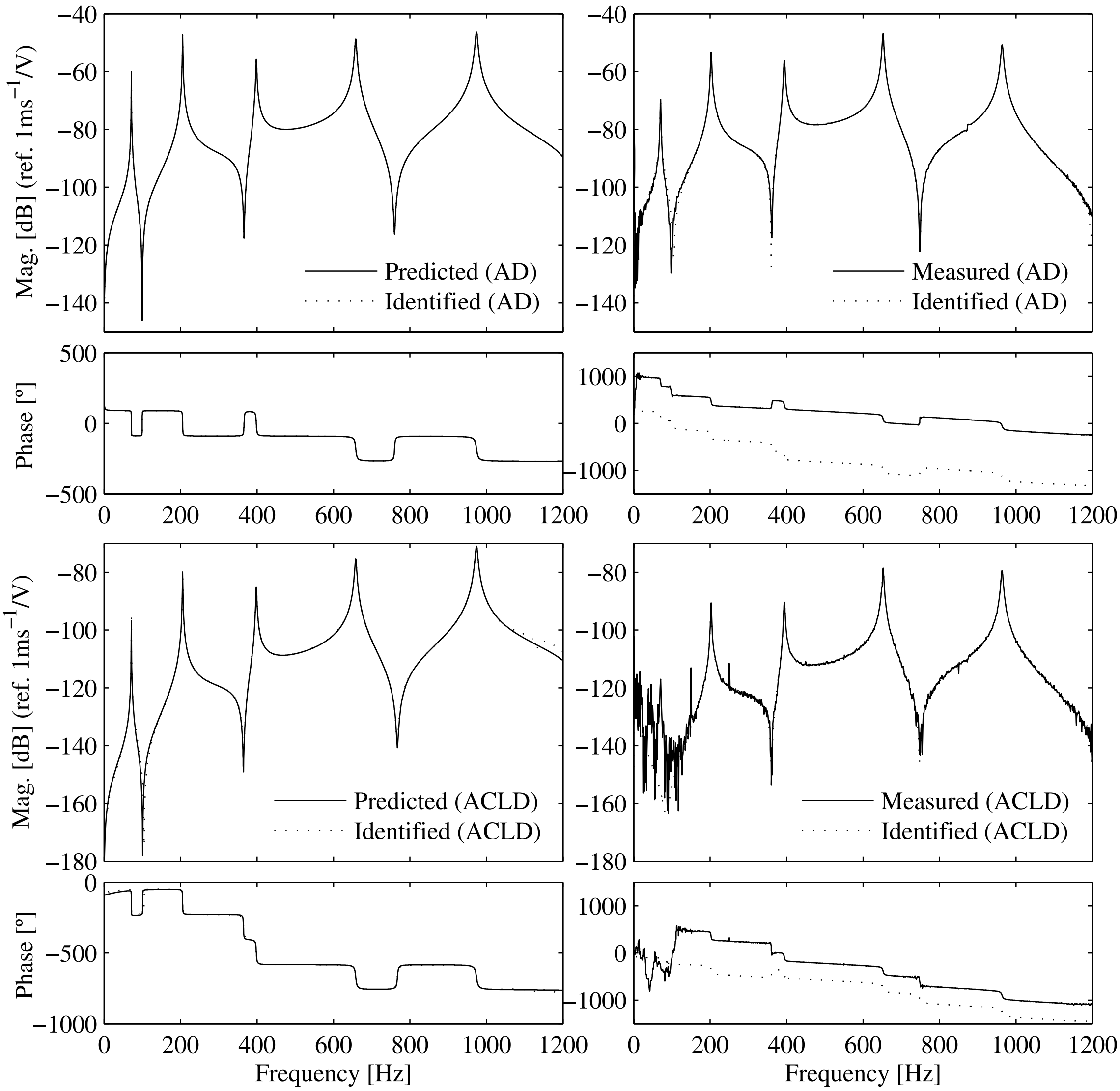

Figure 4: Comparison of the predicted and measured AD and ACLD cancellation path FRFs with the identified ones.

AD/PCLD results is achieved. However, the predicted results for the ACLD are too optimistic when compared with the measured ones, where the control was inefficient. That is justified by the fact that the control voltage of the simulation doesn't have any constraint and can increase indefinitely. However, in practice, the real-time controller was limited to a $150\,\mathrm{V}$ range output, and since ACLD requires higher voltages to have a similar actuating capacity to AD (recall Figure 4 where a two order magnitude difference between the actuating capabilities of AD and ACLD is shown), the real-time controller produces low damping. Of course, in order to be able to increase the excitation to actuating voltage ratio one can decrease the magnitude of the excitation. However, if the values are too low, the signal to noise ratio increases and the controller may become unstable. Other alternative to increase ACLD performance is try to implement a controller that can increase specifically the shearing of the viscoelastic layer, instead of trying to reduce the input power into the structure.

The measured and predicted ACLD to AD control voltage spectral density ratios are presented in Figure 6. It is shown that the predicted ratio is well above the measured one, showing that the real-time controller has not been able to feed the ACLD patch with the required control voltage to achieve a similar behavior to the simulated controller.

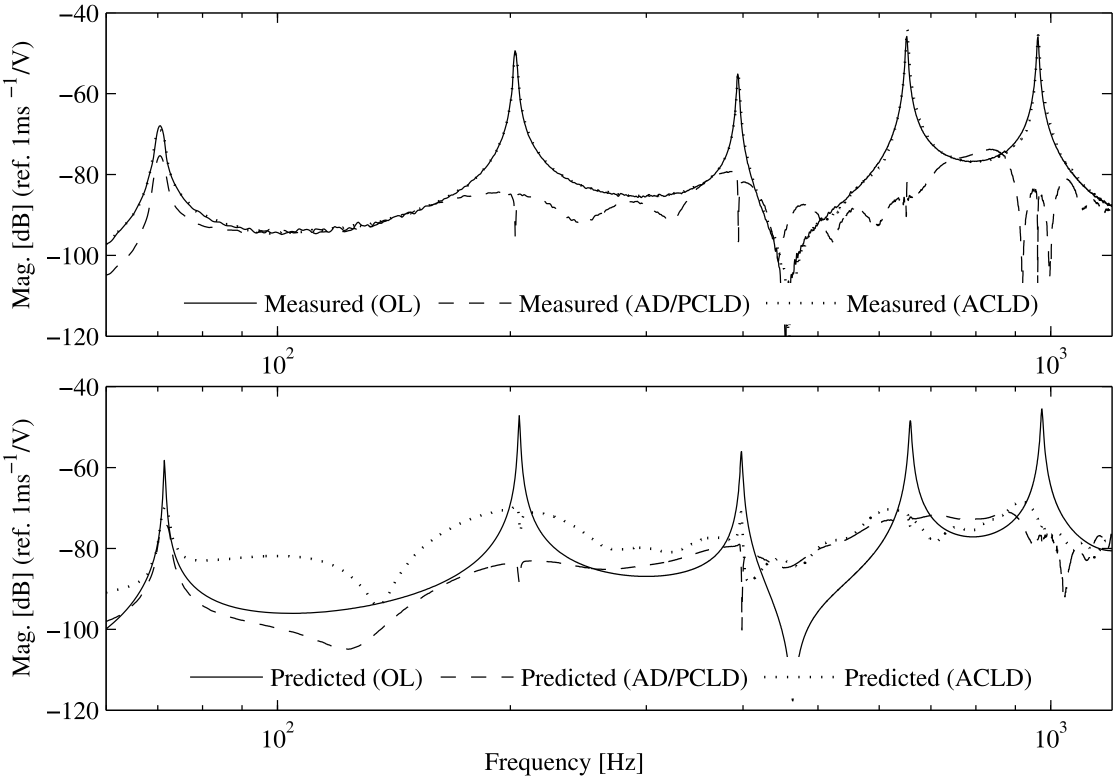

Figure 5: Measured and predicted open-loop (OL) and closed-loop FRFs of the SISO control system.

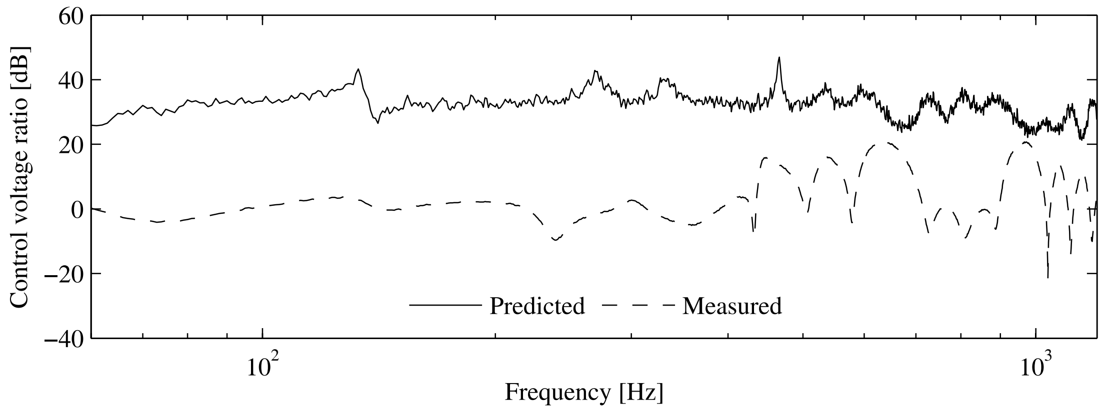

Figure 6: Measured and predicted ACLD to AD/PCLD control voltage spectral density ratios.

CONCLUSION

In this paper an adaptive feedforward controller using the filtered-reference LMS algorithm was utilized in the active control of vibrations of a beam excited by a piezoelectric patch and damped by *Active Damping* (AD) or an *Active Constrained Layer Damping* (ACLD) treatment. The two damping treatments were simulated and experimentally implemented in real-time with the purpose of comparing their damping performance. It was shown that the FE with the viscoelastic damping model utilized here has a very good accuracy and it is a reliable and useful tool for the controller and treatment design. In this specific application the AD/PCLD treatment is more efficient than the ACLD one requiring lower control voltages. The need of higher voltages of the ACLD treatment may constitute a problem in real-time applications due to the constraint of the saturating voltage of the piezoelectric patches or the limited output voltage of the amplifiers. However, one has to be cautious on generalizing the conclusions taken from this work, which may not hold in other cases such as when using feedback control, or when other performance indexes, or cost functions minimizing other parameters, are utilized. To conclude, it should be mentioned that since the transmissibility of efforts from the viscoelastic layer to the host structure is usually reduced, mainly at low frequencies, the performance of the ACLD treatment would only be improved if one designs a controller trying to optimize the damping due to the shearing of the viscoelastic layer, which in turn might require lower control voltages.

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