Vibration Control via Shunted Embedded Piezoelectric Fibers

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ABSTRACT

The scientific community has put significant efforts into the manufacturing of sensors and actuators made of piezoceramic fibers with interdigitated electrodes. These allow for increased conformability, integrability in laminate structures and offer high coupling factors. They are of particular interest for damping applications. This paper presents a comparison between piezoceramic monolithic actuators and Active Fiber Composites (AFCs) for shunt damping. For this purpose, the different actuators were bonded on aluminum cantilever plates, respectively embedded in a glass fiber composite cantilever plate. The vibration suppression was attained by converting the electric charge by means of the converse piezoelectric effect and dissipated through robust resonant shunt circuits. A new circuit topology was used, which enables efficient damping even with low piezoelectric capacitance. An integrated FE model was implemented for prediction of the natural frequencies, the optimum values for the electric components and the resulting damping performance. Patches working in the direct 3-3 mode show much better specific damping performance than the 3-1 actuated patch. The comparison between monolithic and AFC actuators shows that AFCs fulfill integrability and performance requirements for the planned damping applications.

1. INTRODUCTION

In order to damp low frequencies more effectively than with conventional passive damping materials, active vibration control¹⁻⁴ was introduced. However, traditional active vibration control techniques require a lot of electronic devices and power supply making this technology expensive. A new method of vibration control uses a passive electrical impedance that is attached across the terminals of a piezoelectric transducer. This is referred as Shunt Damping⁵⁻⁷. Among several electrical shunt networks, a resonant shunt consisting of an inductor and resistor ($R-L$ shunt⁵), achieves very good vibration suppression for one structure resonance. This shunt has been used to reduce vibration of rotor blades of helicopters⁸, turbo machine blades⁹ and for sport equipments like snowboards¹⁰ or golf-clubs¹¹. It was also successfully applied to increase the sound transmission loss of structures^{12,13} or to damp space structures like the solar array of the Hubble Space Telescope¹⁴ or aircrafts¹⁵. However, the $R-L$ shunt suffers from the drawback that its damping is very sensitive to parameter variations, such as temperature, structural load or piezoelectric parameters. This means that if the $R-L$ shunt is not optimally tuned due to environmental changes, its good damping performance will decrease significantly. This problem was addressed by making the shunt adaptive¹⁶⁻¹⁸, where on-line tuned $R-L$ shunts were proposed to keep the $R-L$ shunt tuned to the mechanical resonance.

The design of composite structures with embedded sensors and actuators requires adequate tools for predicting their static and dynamic behavior. An approach for modeling structures containing piezoelectric actuators using commercially available packages was presented by Reaves and Horta¹⁹. Côté et al.²⁰ proposed a simplified multilayer tri-dimensional model based on the analogy between thermal and piezoelectric strains.

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In the scope of this study the vibration suppression performance of different piezoelectric actuators with shunted $R - L$ circuits is compared. A fully integrated FE model predicts the dynamic response of open and shunt damped systems. Numerical results are validated with according experiments. A first attempt is made towards integration of Active Fiber Composites in composite structures for damping applications.

2. SHUNT CONTROL DAMPING

Figure 1. Adaptive $R - L$ shunted piezoelectric patch.

In this work, we investigate the damping performance of electrical $R - L$ networks shunted to different piezoelectric materials. As already explained, the $R - L$ shunt consists of a serial inductor with a resistor as shown in Figure 1 b) and acts similar like a mechanical damper. According to Hagood et al.²¹, optimal vibration suppression is achieved for

$$L^* = \frac{1}{\omega_n^2(1 + K_{ij}^2 C_p)} \text{ and } R^* = \frac{\sqrt{2} \frac{K_{ij}}{1 + K_{ij}^2}}{C_p \omega_n}, \quad (1)$$

where ω_n denotes the structure resonance frequency to be damped, K_{ij} the generalized piezoelectric coupling factor and C_p the piezoelectric capacitance. Since the tuning of L is very sensitive to C_p and to the structure resonance frequency ω_n that may vary in time, an online tuning of L is required like it is shown in Figure 1b). For this purpose, the Relative Phase Adaptation¹⁸ is applied, since it was shown that this adaptation technique converges faster than former adaptation laws and is easier to implement. The Relative Phase Adaptation of L is

$$\frac{dL(t)}{dt} = \beta (g_{LP}(t) * [v(t) \cdot I_z(t)]), \quad (2)$$

where $g_{LP}(t)$ represents the impulse response of a low-pass filter with a cut-off frequency below $2\omega_n$ (ω_n is the resonance frequency to be damped), β the tuning parameter, $v(t)$ the velocity on the structure where the piezoelectric patch is attached, I_z the current in the shunt circuit and $*$ denotes the time domain convolution operator. For more information the reader is referred to Niederberger et al.¹⁸.

3. EXPERIMENTAL SETUP

3.1. Materials and Set-up

Three different piezoelectric actuator models were investigated. The traditional strain actuator *QP25N* operates on the indirect piezoelectric 3-1 effect, while the new actuator *QP10Ni* takes advantage of the stronger direct 3-3 effect. Both devices were supplied by Mide Corp.*. They are schematically represented in Figure 3 a)

*Mide Technology Corp., 200 Boston Ave, Medford, MA 02155 USA, www.mide.com

Figure 2. One-side clamped plate

and b). As alternative actuator system, Active Fiber Composites²²⁻²⁵ (AFCs) were investigated. AFCs are composed of piezoelectric ceramic fibers and epoxy resin²⁶. Interdigitated electrodes (IDEs) are used for poling and to direct the electric field along the longitudinally oriented fibers (see Figure 3 c)). Just as the *QP10Ni* patches, AFCs achieve greater actuation energy density by exploiting the d_{33} effect along the fibers. The use of piezoceramic fibers maintains the majority of the stiffness and the bandwidth of pure piezoceramics, by simultaneously achieving high conformability and thus integrability in complex curved structures. Furthermore, they allow for anisotropic actuation. Above properties make AFCs interesting components for applications in different fields, such as shape and vibration control.

Figure 3. Investigated piezoelectric actuator models. a) *QP25N*, operating on the indirect 3-1 effect, b) *QP10Ni* and c) AFC, both operating on the stronger direct 3-3 effect.

The experiments were carried out on a one-side clamped aluminum plate as shown in Figure 2. The three different patches were bonded on the plate surface. Additionally, an AFC patch was embedded in the lay-up of a glass fiber reinforced composite (GFRP) plate with same dimensions. A cut-out in the third layer out of nine hosted the AFC. Optimum bonding and embedding location was determined by maximizing the deformation energy due to bending in the active patch area. The mode shape of the second bending mode was computed according to Young²⁷. The bending strain being proportional to the second derivative of the displacement κ

$$\varepsilon(z) = \varepsilon_0 + z \cdot \kappa = -z \cdot \frac{\partial^2 z}{\partial^2 x}, \quad (3)$$

the deformation energy U due to bending available to the piezoelectric patch can be formulated as

$$U = \int_{y_p - \frac{a}{2}}^{y_p + \frac{a}{2}} \frac{D}{2} \cdot (z \cdot \kappa)^2 dy, \quad (4)$$

where D is the flexural rigidity of the plate and a is the patch length.

$$D = \frac{E \cdot d^3}{12 \cdot (1 - \nu^2)}, \quad (5)$$

Patch	Actuation Mode	$R [\Omega]$	$L [H]$
QP25N	3-1	0.63k	2.118
QP10Ni	3-3	-8.5k	93.7
AFC	3-3	-14k	100

a)

b)

Figure 4. a) Values of the $R - L$ shunt for the different piezoelectric patches b) Synthetic Impedance Circuit.

with E and ν representing the material's Young modulus and Poisson ratio, respectively. d is the plate thickness. Setting the derivative $\partial U / \partial y$ to zero yields the optimum patch location. The parameters of the set-up are summarized in Figure 2 (right), whereas the mechanical properties for the aluminum alloy, the GFRP laminate and the piezoelectric patches can be found in Tables 2 - 4.

For the evaluation of the damping capability of the different piezoelectric patches, the transfer-function from a disturbance generated by an electromagnetic transducer to the velocity of the structure was measured. We define this transfer-function as

$$G_v(s) = \frac{v(s)}{U_{ex}(s)}, \quad (6)$$

where U_{ex} corresponds to the excitation voltage of the electromagnetic transducer and $v(s)$ the velocity. Additionally, the plate was excited by a sine-wave with a frequency equal to the resonance frequency of the 2nd bending mode. At time 1 s, this excitation was switched off, in order to compare the vibration decay for the open and the shunt damped system.

3.2. Implementation of the Shunt Control Circuit

As the piezoelectric capacitance of AFCs and QP10Ni patches is very small, the optimal inductance L^* of the $R - L$ shunt is very big (see Equation 1). Therefore, the conventional virtual inductor^{28,29} is not able to synthesize such a huge inductance with sufficient low serial resistance since it cannot drive the very small piezoelectric capacitance accurately enough. For example, this was observed in Sato et al.³⁰, where a vibration suppression of only 0.45 dB was obtained with a $R - L$ shunted piezoelectric fiber composite patch. In order to solve the problem of the implementation of the $R - L$ shunt, we propose the Synthetic Impedance Circuit³¹ shown in Figure 4b). The desired impedance transfer-function defined by $\frac{U_z(s)}{I_z(s)} = Z(s)$ is implemented by measuring the voltage $U_z(s)$ and controlling the terminal current $I_z(s)$. With this configuration, an arbitrary electrical network $Z(s)$ can be synthesized. Since the Digital Signal Processor (DSP), that controls the current source, is implemented with the XPC-Target System of MATLAB[†], the topology of the electrical shunt network and its parameters can easily and quickly be changed. The optimal parameters of the $R - L$ shunt for the different piezoelectric patches are summarized in Figure 4 a). Parasitic resistance in the synthetic impedance can explain the negative values of the resistors, since it is very difficult to drive the very low capacitance of the QP10NI and AFC patch.

4. FE ANALYSIS

In the scope of this study, the damping performance of the different patches was predicted by an integrated FE analysis. The new piezoelectric-circuit analysis capability of ANSYS 7.1 was taken advantage of for full harmonic analysis. This analysis can determine voltage and current distributions in an electric circuit with piezoelectric devices. SOLID 45, 8-node 3D structural elements were used to model the passive structure; SOLID 5, 8-node 3D coupled field elements were used for the piezoelectric material. The shunt circuit was implemented using

[†]The MathWorks, Inc., 3 Apple Hill Drive, Natick, MA 01760-2098 USA, www.mathworks.com

CIRCU 94 elements, which work with the large deflection and stress stiffening capabilities of the element SOLID 45. The layered version of the 8-node structural solid (SOLID46) was used to model the layered GFRP plate.

The actuators' capacitance was first predicted by a static analysis. Clamped conditions were applied to Representative Volume Elements (RVE) designed for each patch configuration. The charge on the electrode divided by the applied voltage yielded the capacitance of the RVE. Modal analysis was carried out in order to extract the resonance frequencies for open and short circuit regimes. Coupling coefficients were determined as

$$K_{ij} = \frac{\omega_{open}^2 - \omega_{short}^2}{\omega_{short}^2}, \quad (7)$$

where ω_{open} and ω_{short} are the resonance frequencies of the systems with open and short circuited patch, respectively. Optimum values for R^* and L^* were computed according to Equation 1, thus tuning the $R - L$ shunt to the 2nd bending mode. An harmonic analysis was then carried out for the addressed frequency spectrum. For the prediction of the damping capability, the transfer-function from the force generated by the electromagnetic transducer to the displacement of the structure was computed. The transfer function is defined as

$$G_{u_z}(s) = \frac{u_z(s)}{F(s)}. \quad (8)$$

If nonlinearities are neglected, the force developed by the electromagnetic transducer is proportional to the current in the coil, and thus to the integral of the excitation voltage, if the coil's resistance is very small. On the other hand, integration of the velocity yields the displacement.

$$F \propto I = \int v \, dt \quad (9)$$

$$u_z = \int v \, dt \quad (10)$$

Even though physically not perfect, the proposed analogy allowed for qualitative comparison between experimental and numerical results.

5. RESULTS AND DISCUSSION

5.1. Experimental Results

5.1.1. Vibration Suppression

In this experiment, the transfer-functions $G_v(s)$ from disturbance to velocity on the structure were measured. Curves measured for the open and the shunt damped systems are shown in Figure 5. The resonant $R - L$ shunt is tuned to the 2nd bending mode. One can see that the shunted QP25N, QP10Ni and AFC patches bonded on the aluminum cantilever plate achieve a vibration suppression of the 2nd bending mode by about 17–20 dB. The exact values are summarized in Table 1. The specific vibration suppression per unit volume of active piezoelectric material highlights the enhanced efficiency of systems working in 3-3 mode. The AFC embedded in the GFRP laminate achieves a lower vibration suppression of about 8 dB (the frequency range in Figure 5 d) was reduced in order to make the damping visible). In absolute magnitude, this is 2.82 times less efficient than the patch bonded on the surface. The reduced vibration suppression can be explained by looking at the distance z from the neutral line at which the patch is embedded in the lay-up, and thus at the strain experienced by the piezoelectric fibers. This is approximately 2.5 times lower than for the bonded patches. On the other hand, a slight decrease in vibration suppression performance is expected due to the higher critical damping ratio of GFRP compared with aluminum.

5.1.2. Damping

The damping of the 2nd bending mode for the open and shunt damped systems were measured by harmonically exciting the plate at the corresponding resonance frequency. After time 1 s, the excitation was switched off. The vibration decay for the three investigated aluminum plates is presented in Figure 6. The three systems show very similar behavior, with the AFC configuration being slightly less efficient.

Patch	Actuation Mode	Active Volume [mm ³]	Vibration Suppression [dB]	
			Measured	Predicted
QP25N	3-1	240.2	20.5	19.6
QP10Ni	3-3	120.1	17.5	23.5
AFC	3-3	109.6	17.0	20.9
Embedded AFC	3-3	109.6	8.0	10.4

Table 1. Damping performance of different piezoelectric patches. Measured and predicted vibration suppression values are compared.

Figure 5. Measured transfer-functions $G_v(s)$ with shunt (light) and without shunt (dark). a) QP25N bonded on aluminum plate, b) QP10Ni bonded on aluminum plate, c) AFC bonded on aluminum plate, d) AFC embedded in GFRP laminate.

5.2. Numerical Results

As often reported in previously published work, the modal analysis slightly overestimates the natural bending frequencies of the cantilever plate with bonded patch. Modeling perfect clamping conditions, the model is stiffer than the implemented experimental setup. Additional error sources can be found in the geometrical dimensions and in variations of the material properties.

Figure 6. Measured sine-decay with shunt (light) and without shunt (dark) for patches bonded on aluminum plate. a) *QP25N*, b) *QP10Ni*, c) *AFC*.

Figure 7. Predicted transfer-function $G_{uz}(s)$ with shunt (dotted line) and without shunt (straight line). a) *QP25N* bonded on aluminum plate, b) *QP10Ni* bonded on aluminum plate, c) *AFC* bonded on aluminum plate, d) *AFC* embedded in GFRP laminate.

As shown in Table 1, the vibration suppression was successfully predicted by the integrated FE model. The numerical results slightly overestimate the measured values for the *QP10Ni* and AFC patches, both working in the more efficient 3-3 mode. This is mainly due to the uniformly defined piezoelectric coefficients. Being computed for the region where the field is uniform, these are clearly exceeding the real values in the "dead zone" under the interdigitated electrode.

Figure 7 presents the predicted transfer-functions for the open and the shunt damped systems. The analogy to the measured transfer-functions is evident. Due to the neglected nonlinearities, however, predicted and experimental curves are not presented in the same graph. The straight lines represent the transfer-functions of the systems with no shunt damping. Optimum R^* and L^* parameters determined according to Equation 1 tuned the resonant $R - L$ shunt to the 2nd bending mode. The transfer-functions of the shunt damped systems are represented by dotted lines.

6. CONCLUSIONS

For the first time, damping was successfully implemented shunting an $R - L$ circuit on piezoelectric elements working in 3-3 mode. The need for very high inductance was overcome with a synthetic impedance (instead of the virtual impedance). The Relative Phase Adaption allowed for online tuning of the resonance frequency, thus making the system robust. The vibration suppression was predicted by analyzing the passive structure, the bonded active patch and the shunted circuit with an integrated FE model. The numerical simulations showed good agreement with the experimental results. *QP10Ni* and AFC patches, both working in the more efficient 3-3 mode, show much better specific damping performance than the 3-1 actuated *QP25N* patch. In addition, AFC patches exhibit low mass and anisotropic actuation. They have a great potential towards integration in complex, double curved structures thanks to their inherent conformability.

APPENDIX A. MATERIAL PROPERTIES

AlSi1MgMn		
Density	ρ [kg/m^3]	2700
Young's Modulus	E [GPa]	69
Poisson ratio	ν [-]	0.33

Table 2. Material properties for the AlSi1MgMn aluminum alloy.

Glass/Epoxy fabric prepreg		
Density	ρ [kg/m^3]	2000
Young's Modula	E_{11} [GPa]	26
	E_{22} [GPa]	26
	E_{33} [GPa]	10
Shear Modula	G_{12} [GPa]	3.53
	G_{23} [GPa]	3.7
	G_{13} [GPa]	3.7
Poisson ratios	ν_{12} [-]	0.112
	ν_{23} [-]	0.311
	ν_{13} [-]	0.311

Table 3. Material properties for the glass/epoxy fabric prepreg.

	QP25N	QP10Ni	AFC
e_{31} (C/m^2)	-5.35	-2.579	-0.25
e_{33} (C/m^2)	15.78	7.837	10.65
e_{15} (C/m^2)	12.3	6.101	0
ϵ_1^S/ϵ_0	916	916	916
ϵ_3^S/ϵ_0	830	830	830
c_{11}^E (GPa)	120	120	19.6
c_{12}^E (GPa)	75.1	75.1	8.7
c_{13}^E (GPa)	75.1	75.1	10.7
c_{22}^E (GPa)	120	120	25.4
c_{23}^E (GPa)	75.1	75.1	13
c_{33}^E (GPa)	111	111	48.6
c_{44}^E (GPa)	21	21	3.09
c_{66}^E (GPa)	21	21	0.26

Table 4. Effective material properties for the different patches.

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