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Application of a LIPCA for the Structural Vibration Suppression of an Aluminum Cantilever Beam with a Tip Mass

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Abstract

Use of bare PZT as an actuator in the field of active vibration suppression may cause some drawbacks such as critical breaks in the installation process, short circuits in the host material and low fatigue performance. To alleviate these problems, we developed a new actuator called a lightweight piezocomposite actuator (LIPCA). The LIPCA has five layers: three glass-epoxy layers, a carbon-epoxy layer and a PZT layer. We implemented a LIPCA as an actuator to suppress the vibration of an aluminum cantilever beam with a tip mass. For the control algorithm in our test, we used positive position feedback. The filter frequency for this type of feedback should be tuned to the frequency of the target mode. The first three experimental natural frequencies of the aluminum cantilever beam agree well with the results of finite element methods. The effectiveness of using a LIPCA as an actuator in active vibration suppression was investigated with respect to the time and frequency domains, and the experimental results show that LIPCAs can significantly reduce the amplitude of forced vibrations as well as the settling time of free vibrations.

Keywords: Lightweight piezocomposite actuator(LIPCA); Positive position feedback(PPF); Structural vibration control

1. Introduction

In recent years, an enormous amount of attention has been paid to the vibration suppression of flexible structures. When these structures are embedded with an actuator, a sensor and a control system, they are called smart structures. The actuator is typically made of a piezoelectric material, the most popular comercial example of which is lead zirconium titanate or PZT. A number of researchers have used bare PZT as an actuator to suppress vibration, as shown in the works of Fanson *et al.* (1990), Kwak *et al.* (2002) and Kermani *et al.* (2004), though this method may cause drawbacks such as critical breaks in the installation process, short circuits in the host material and low fatigue performance. Yoon *et al.* (2002) developed a new actuator called a lightweight piezocomposite actuator (LIPCA) that could alleviate such problems.

In controlling flexible structures, Kwak *et al.* (2002) and Baillargeon (2005) have successfully used the positive position feedback (PPF) algorithm as a controller. Fanson and Caughey (1990) originally developed the PPF control, the main advantage of which is that it can control the target mode without disturbing other modes. The frequency of the PPF control should be tuned to the frequency of the target mode, and finite element simulation can be used to effectively predict the natural frequency of the con-

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trolled system. Furthermore, because the PPF control is a form of strain-based sensing, a strain gage is a convenient sensor for this control.

Using the PPF control, we implemented a LIPCA as an actuator to suppress the vibration of an aluminum cantilever beam with a tip mass. Our goal was to experimentally demonstrate that a LIPCA can be used to suppress the vibration of a cantilever beam with a tip mass. We therefore used root locus analysis to determine an appropriate scalar gain. After obtaining the appropriate scalar gain from the simulation, we then applied it to the real system. We also performed a free vibration control and a forced vibration control. In short, our experiments demonstrate the effectiveness of using a LIPCA to actively suppress vibrations.

2. The LIPCA

The LIPCA is a kind of piezocomposite unimorph actuator comprised of five layers, as shown in Fig. 1. Table 1 lists the properties of these materials. After completing an autoclave bagging process, we vacuum bagged the stacked layers and cured them at an elevated temperature (177 °C). The LIPCA has some advantages over conventional unimorph actuators: for example, aside from being lightweight and easy to fabricate, it has flexibility in the installation process and good fatigue performance. The glass-epoxy layers in the LIPCA can also be used to conveniently isolate the high voltage PZT layer from the host structures.

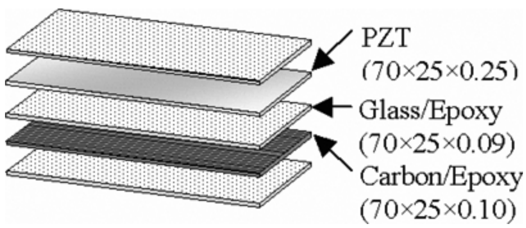

Fig. 1. The lay-up of the LIPCA-C2.(unit: mm)

Table 1. Properties of the constituent materials of LIPCA-C2.

Properties	Carbon-epoxy	Glass-epoxy	PZT 3032HD
E_1 (GPa)	231.2	21.7	62
E_2 (GPa)	7.2	21.7	62
G_{12} (GPa)	4.3	3.99	23.7
ν_{12}	0.29	0.13	0.31
d_{31} (10^{-12} m/V)	-	-	-320

Kim *et al.* (2005) reported that in order to obtain a good performance a variety of LIPCAs were developed with different materials, dimensions and stacking sequences. The latest and the best LIPCA is LIPCA-C2, which we used in this test.

3. PPF control

The PPF control is a strain-based sensing approach that is used to suppress vibrations, and the PPF algorithm has several advantages: it is insensitive to a spillover; it is convenient for strain sensing because it uses generalized displacements; and it can offer quick damping for a particular mode if the modal characteristics are well known. The PPF principle involves the direct feeding of the structural position coordinates to the compensator, resulting in the compensator position coordinates shown in Fig. 2.

The scalar system consists of the following two equations:

$$\begin{aligned} &\text{Structure}: \ \ddot{\eta} + 2\zeta\omega\dot{\eta} + \omega^2\eta = g\omega^2\xi \\ &\text{Compensator}: \ \ddot{\xi} + 2\zeta_f\omega_f\dot{\xi} + \omega_f^2\xi = \omega_f^2\eta, \end{aligned} \quad (1)$$

where $g > 0$ is the scalar amplifier gain, ω is the structural natural frequency, ω_f is the filter natural frequency, ζ is the structural damping ratio, and ζ_f is the filter damping ratio. By taking the Laplace transforms of these equations, we can find the transfer functions of the structure and compensator, which are given by

$$\begin{aligned} G(s) &= \frac{\omega^2}{s^2 + 2\zeta\omega s + \omega^2}, \\ H(s) &= \frac{\omega_f^2}{s^2 + 2\zeta_f\omega_f s + \omega_f^2} \end{aligned} \quad (2)$$

To implement the transfer function in the control system, we had to convert the compensator transfer function to a digital form. We used a bilinear transform to convert the analog expression to a digital equation.

By using the Laplace transfer function in Eq. (2), we enabled the closed loop control system of Fig. 2 to produce the root locus of Fig. 3. In Fig. 3, the boxes indicate the desired pole locations of the closed loop, whereas the X marks indicate the pole locations of the open loop. The scalar gain is zero in the open loop poles. As the scalar gain increases, the boxes (closed loop poles) move away from open loop poles,

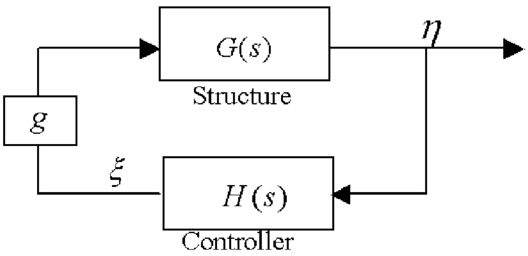

Fig. 2. Closed loop control block diagram.

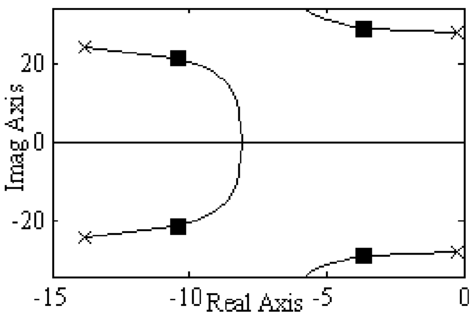

Fig. 3. Root Locus of closed loop system.

which in turn increases the damping of the compensator. However, for large damping values, the damped frequency can shift large amounts and, ultimately, the system can become unstable. It is important therefore to use just enough gain to move the structural poles away from open loop poles. Furthermore, one usually needs to use applied actuator voltage as low as possible to minimize energy consumption and avoid instability and domain switching problems while still guarantee the damping of the controlled system.

4. Test setup

The host structure in our test is an aluminum cantilever beam with dimensions of 275 mm × 25 mm × 0.75 mm. As shown in Fig. 4, the beam has a LIPCA and a strain gage, which are attached at the root as a collocated system; the beam also has an acrylic mass attached at the tip.

After the beam was shaken by sine wave excitation, we used a Bruel and Kjaer fast Fourier transform analyzer to determine the frequency response function, the results of which enabled us to examine the dynamic response and the closed loop performance. We used the LIPCA to suppress the vibration generated by the excitation. Because the LIPCA needs a large voltage to operate, we used a high-voltage amplifier to amplify the real-time active

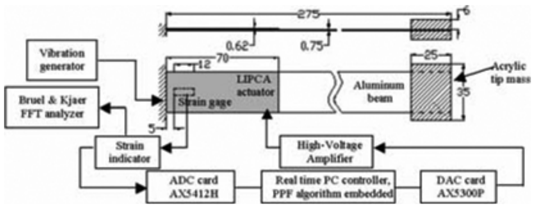

Fig. 4. Diagram of the experimental setup.

vibration control signal, which is produced by a PC controller embedded with the control algorithm. For the data acquisition system, we used the AX5412 and AX5300 Axiomtek digital-to-analog boards.

5. Results

5.1 The frequency response function and the mode shape of the cantilever beam

Figure 5 shows the frequency response function of the aluminum cantilever beam without the PPF control. The three peaks in the frequency range of 0 Hz to 120 Hz, which indicate the natural frequencies of the cantilever beam, represent three modes.

In Table 2, which compares the first three bending natural frequencies of the experiments with the finite element results, we can see that the experimental results agree well with the finite element results. The finite element model predicts the natural frequency of the cantilever beam very well. Figure 6 shows the finite element mode shape of the cantilever beam

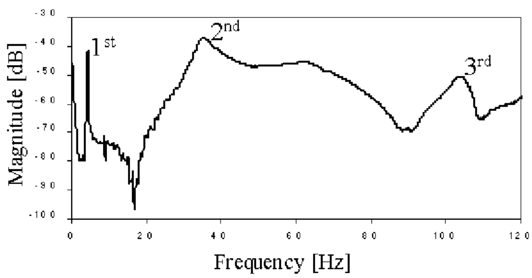

Fig. 5. Frequency response function without feedback control.

Table 2. Comparison of the first three bending natural frequencies.

Mode	Natural Frequency [Hz]		Difference [%]
	Experiment	Finite Element	
1	4.4	4.3	2.2
2	35.4	35.4	0.0
3	104.5	102.6	1.8

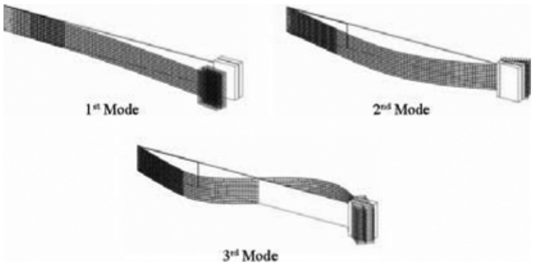

Fig. 6. The first three bending natural frequencies of the cantilever beam, as obtained from the finite element model.

corresponding to the natural frequencies of the first three bending modes listed in Table 2.

5.2 The transient and forced response of the system

The goal of the vibration control design is to obtain the maximum damping of a controlled structure. The maximum damping is usually obtained by applying a large gain to the system and, for a large gain, the actuator needs a high voltage. However, a high voltage may cause the system to become unstable and the fatigue performance of the actuator may deteriorate.

To obtain the optimum gain for the system, we conducted root locus analysis. We chose a scalar gain that was just high enough to ensure the damping of the vibration. When we set the scalar gain, g, to 1, the gain margin was 4.78. The gain margin represents the degree of change required in the scalar gain to make the system unstable. However, at g = 4.09, the poles of the control appear on the real axis. If the pole has no imaginary part, then, immediately after the feedback control is applied, either the system becomes overdamped or the beam fails to oscillate. The larger the gain, the better the controllability of the system.

Figure 7(a) shows the reduction of the fundamental mode peak in relation to the increasing scalar gain. When we set the scalar gain to 1, the peak was reduced from -38 dB to -52.5 dB, for a total reduction of 14.5 dB, which is significant enough to suppress the vibration of the fundamental mode.

To analyze the reduction in the settling time, we performed a free vibration control. The cantilever beam was excited with sine waves at the fundamental mode. Figure 7(b) shows the free response of the beam right after the disturbance was turned off. The settling time, which we defined as the time required to ensure that the amplitude of the tip beam is reduced to 2 percent of the initial displacement, was 23

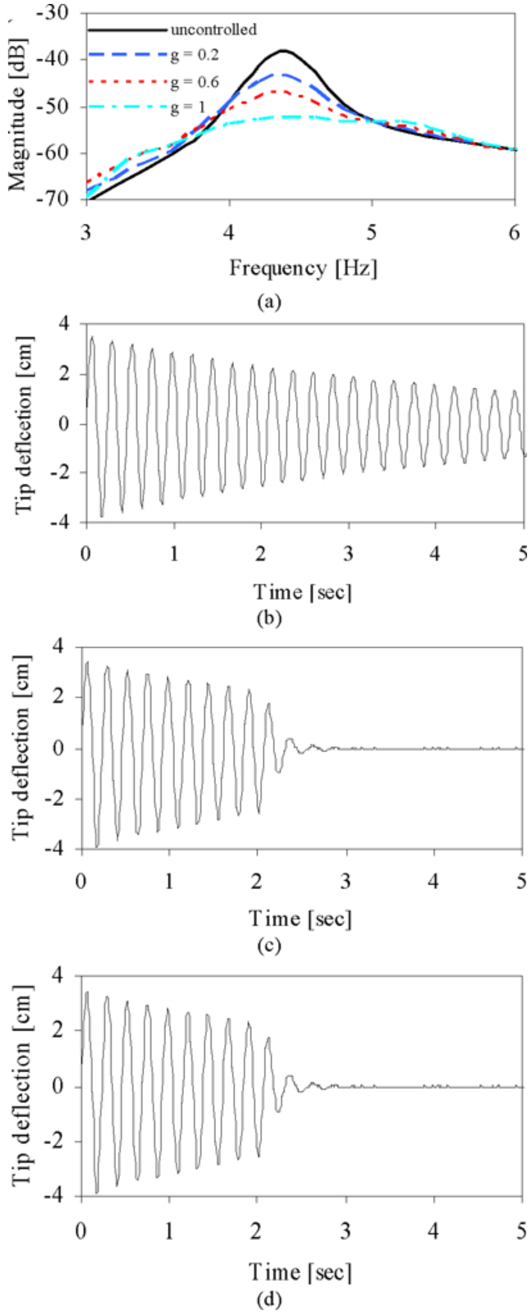

Fig. 7(a). Frequency response in relation to the varying scalar gain, (b) Free vibration response of the aluminum cantilever beam, (c) Free vibration response with the feedback control starting at 2 seconds, (d) Forced vibration response with the feedback control starting at 2 seconds.

seconds. Figure 7(c) shows how the cantilever beam with the PPF control responded when the beam was activated at a time of 2 seconds. With the same settling time criteria, the vibration of the cantilever beam settled one second after the PPF control was activated. This result means that the LIPCA reduces the settling time by 95.8 percent.

For the forced vibration control, we used a vibration generator to create harmonic excitation of the beam at the fundamental frequency. Figure 7(d) shows the response of the beam before and after we applied the PPF control to the cantilever beam. When no control is applied to the beam, the amplitude of the tip beam is 3.5 cm. However, when the PPF feedback control is applied to the system, the amplitude of the beam tip is reduced to 0.5 cm, representing an 85.7 percent reduction in the amplitude.

6. Conclusion

We investigated the use of a LIPCA in the active vibration suppression of an aluminum cantilever beam with a tip mass. For the control algorithm, we used PPF; and we used root locus analysis to select the control gain. We also used a finite element model to predict the natural frequencies of the cantilever beam. The effectiveness of the LIPCA was observed in the frequency response function. Furthermore, the experimental results show a significant reduction in the resonant peak of the fundamental mode. To investigate the effectiveness of the LIPCA in the time domain, we performed a forced vibration control and a free vibration control. The results show a significant reduction in the amplitude of the vibration in the forced vibration control and a significant reduction in the settling time in the free vibration control. In conclusion, we deduce that the LIPCA can be an effective actuator in active vibration suppression.

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