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Active Vibration Suppression of Sandwich Beams using Piezoelectric Shear Actuators

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Summary

This paper deals with the experimental and numerical assessment of the vibration suppression of smart structures using piezoelectric shear actuators. Experimental results are presented for an adaptive sandwich cantilever beam that consists of aluminum facings and a core made of two piezoelectric shear actuators and foam. The electric field is applied perpendicular to the poling direction of the piezoelectric actuators to cause transverse shear deformation of the sandwich beam. Active vibration suppression is achieved using positive position feedback. Experiments and numerical simulations show that the shear actuators can provide significant reduction in tip acceleration and settling time.

Introduction

Piezoelectric actuators used in adaptive structures are usually thin wafers which are poled in the thickness direction and bonded to the surfaces of the host structure. The application of an electric field in the thickness direction causes the lateral dimensions of the actuators to increase or decrease, thereby forcing the host structure to deform. The actuators are usually placed at the extreme thickness positions of a plate-like structure to achieve the most effective actuation. This subjects them to high longitudinal stresses and may lead to failure, especially when they are made of brittle piezoceramics. To alleviate these problems several researchers have investigated adaptive sandwich structures consisting of axially-poled piezoelectric actuators (e.g., see [1, 2]). The application of an electric field E_3 in the thickness direction of a piezoelectric actuator that is poled in the x_1 direction, will induce a transverse shear strain γ_{13} as shown in Fig. 1(a). The axially-poled piezoelectric shear actuator, when sandwiched between two elastic facing sheets, will cause transverse deflection of the sandwich structure. Since the shear actuators are placed at the midsurface of the beam, they are subjected to smaller longitudinal stresses than extension actuators. An exact analysis of the free vibration, forced vibration and active vibration suppression of simply

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supported adaptive composite beams and shells with embedded piezoelectric shear actuators was presented recently by Baillargeon and Vel [3, 4]. To date, there has been little experimental research on the effectiveness of the piezoelectric shear mechanism for active vibration suppression. In this paper, we demonstrate using experiments that piezoelectric shear actuators can be utilized for active damping of adaptive structures and compare the experimental results to finite element simulations. The sandwich beam consists of aluminum facings and a core composed of foam and two piezoceramic shear actuators. In the first part of the study, the frequency response function of the sandwich beam is obtained by using one of the shear actuators to excite the beam and the other shear actuator to suppress the vibrations. It is found that the experimentally obtained frequency response function compares well with the numerical results. In the second part of the study, the transient response of the adaptive beam is analyzed by using a proof-mass actuator for excitation. In this case, both piezoelectric shear actuators are used for active vibration suppression. The experiments demonstrate the effectiveness of piezoelectric shear actuators for active vibration suppression.

Experimental Setup

A sandwich cantilever beam consisting of aluminum facings and a core made of two piezoceramic shear actuators and foam is fabricated for the experimental studies. The configuration of the sandwich beam used in this study is depicted in Fig. 1(b). The facings of the sandwich beam are made of 6061-T6 aluminum. The core consists of two axially-poled PZT-5A shear actuators and lightweight foam (Baltek Airex R82.80). The beam is 0.305 m in length and 0.0142 m in total thickness. The principal material direction of the PZT-5A actuators is oriented along the longitudinal axis of the beam. The electric field is applied through the thickness of the piezoceramic actuators via electrodes at the interface between the actuator and aluminum facings. The two PZT-5A shear actuators are labeled as P_1 and P_2. In the first part of the study, the excitation actuator that is closest to the clamped end, denoted by P_1, is used as a disturbance actuator to excite the beam and P_2 is the control actuator for vibration suppression. Two sensors are attached to the cantilever beam. A dynamic strain sensor near the clamped edge is used to measure the deformation for the purpose of feedback control, and an accelerometer at the tip of the beam is used to quantify the effectiveness of the vibration suppression system. The outputs from both sensors are passed through a signal conditioner before acquisition of the respective signals. The signal from the accelerometer is acquired through SigLab, which

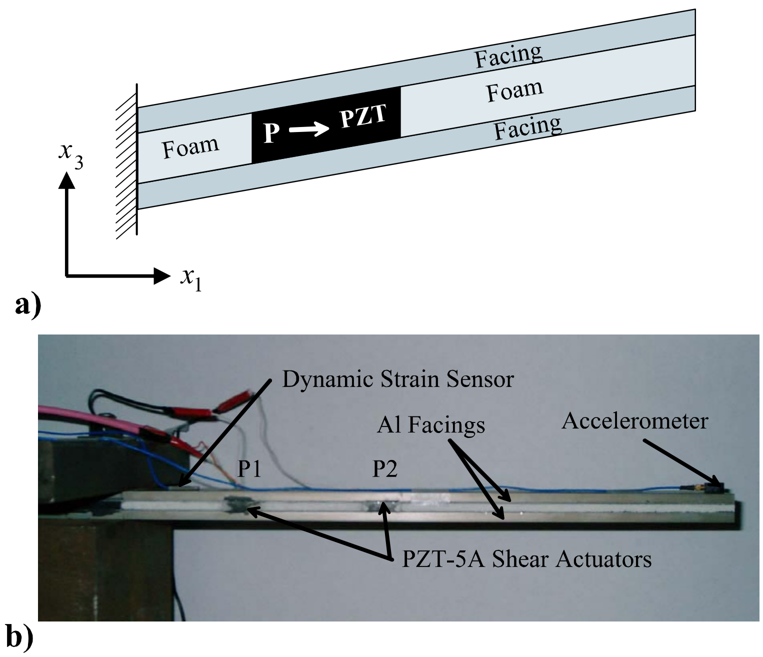

Figure 1: Configuration of adaptive beam consisting of shear actuators and foam core

is a dynamic signal and system analyzer that is accessed through a Matlab graphical user interface. The frequency response function of the system is determined using a chirp excitation to shear actuator P_1. In the second part of the study, a low-voltage piezoelectric stack and a steel block (mass 0.126 kg) are bonded to the tip of the cantilever beam, to form a proof-mass actuator that provides a repeatable harmonic disturbance to the beam. The application of a harmonic voltage to the piezoelectric stack, induces a harmonic strain in the thickness direction of the stack, resulting in a harmonic force at the tip of the beam due to the inertia of the steel block. The embedded piezoelectric shear actuators P_1 and P_2 are used to suppress the vibration caused by the proof-mass actuator.

Control System Design

Active vibration suppression is achieved using positive position feedback (PPF) [5]. The PPF controller is not sensitive to spillover and it is not destabilized by finite actuator dynamics. The PPF controller consists of a second order compensator which is forced by the displacement response of the structure. The output of the compensator, magnified by a gain, is fed back to the structure. The equations that describe the operation of the PPF compensator are

$$\begin{aligned} &\ddot{\xi} + 2\varsigma_n\omega_n\dot{\xi} + \omega_n^2\xi = G_p\omega_c^2\eta, \\ &\ddot{\eta} + 2\varsigma_c\omega_c\dot{\eta} + \omega_c^2\eta = \omega_c^2\xi, \end{aligned} \tag{1}$$

where ξ is the modal coordinate that describes the deformation of the structure, ω_n is the natural frequency of the structure, ς_n is the damping ratio of the structure, η is the compensator coordinate, ω_c is the compensator frequency, ς_c is the compensator damping coefficient and G_p is the feedback gain coefficient.

Finite Element Model of Adaptive Beam

The numerical model used in this study utilizes the commercial finite element analysis package ABAQUS/Standard 6.3-1 to analyze the active feedback control of a cantilever beam. A two-dimensional plane stress analysis of the adaptive beam is performed using 8-node biquadratic plane stress elements for the elastic and piezoelectric materials, respectively. Transient numerical simulations are performed using the finite element method by utilizing an implicit dynamic analysis subroutine. The proof-mass actuator is used to induce an excitation at the tip of the cantilever beam and the piezoelectric shear actuators P_1 and P_2 are used for active vibration suppression. The second order PPF compensator is modeled directly as second order spring-mass-damper system in ABAQUS.

Results and Discussion

The frequency response function of the cantilever beam without active control is compared with experimental results by applying a harmonic potential of constant amplitude to the piezoelectric actuator P_1. The ratio of the steady-state tip acceleration (in m/s^2) to the excitation voltage (in Volts) applied to P_1 is plotted as a function of the forcing frequency in Fig. 2(a). The experimentally obtained natural frequencies are $\omega_1 = 123.06$ Hz, $\omega_2 = 495.71$ Hz, $\omega_3 = 1098.28$ Hz, $\omega_4 = 1538.20$ Hz which compare well with the finite element natural frequencies of $\omega_1 = 126.85$ Hz, $\omega_2 = 495.12$ Hz, $\omega_3 = 1094.6$ Hz, $\omega_4 = 1558.3$ Hz.

The effect of using PPF control to actively suppress vibration of a cantilever beam is examined in the frequency domain by using shear actuator P_1 as

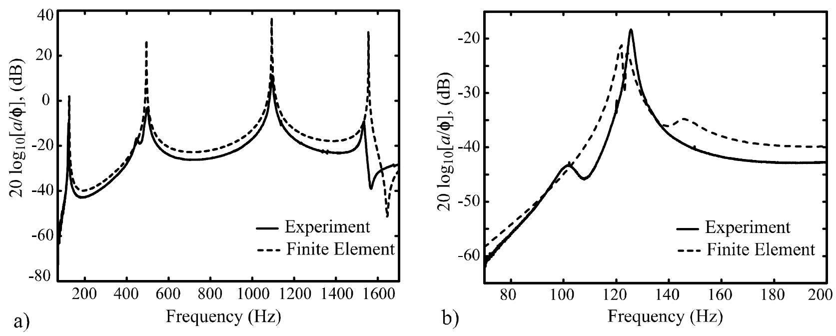

Figure 2: (a) Comparison of the experimental and finite element frequency response function without feedback control and (b) comparison of experimental and finite element frequency response functions using PPF control with $\omega_c = 1.1\omega_1$, $\varsigma_c = 0.05$ and $G_p = 17 \times 10^{-6}$ V-s^2.

the disturbance source and shear actuator P_2 for active vibration suppression. The strain, which is measured by the dynamic strain sensor, is chosen as the measure of the structural position for the PPF compensator. The experimental and finite element frequency response curves for $\omega_c = 1.1\omega_1$, $\varsigma_c = 0.05$ and $G_p = 17 \times 10^{-6}$ V-s^2 are compared in Fig. 2(b). The closed-loop resonant frequency from the numerical simulation is smaller than the experimentally obtained value. It is noted the finite peak in the frequency response is due to the PPF compensator since structural damping was not included in the finite element model.

The active feedback control of the cantilever beam in the time domain is investigated by attaching a proof-mass actuator, consisting of a low-voltage piezoelectric stack and steel mass, to the tip of the adaptive sandwich beam to provide repeatable tip excitation, and the two shear actuators are used for active vibration suppression. The addition of the proof-mass actuator causes the experimental fundamental frequency to drop to $\omega_1 = 75.66$ Hz. The PPF controller utilized for this portion of the study has compensator frequency $\omega_c = 1.3\omega_1$ and compensator damping coefficient $\varsigma_c = 0.3$. The scalar gains for both shear actuators P_1 and P_2 are $G_p = 17 \times 10^{-5}$ V-s^2. The beam is excited from rest by applying a harmonic potential at the fundamental frequency by the piezoelectric stack for 0.1 s, after which the excitation is switched off and the amplitude of vibration decays either due to natural damping or due to active vibration suppression using PPF control. A Rayleigh structural damping model is incorporated in the transient finite element analysis in order to obtain a realistic transient response for the cantilever beam.

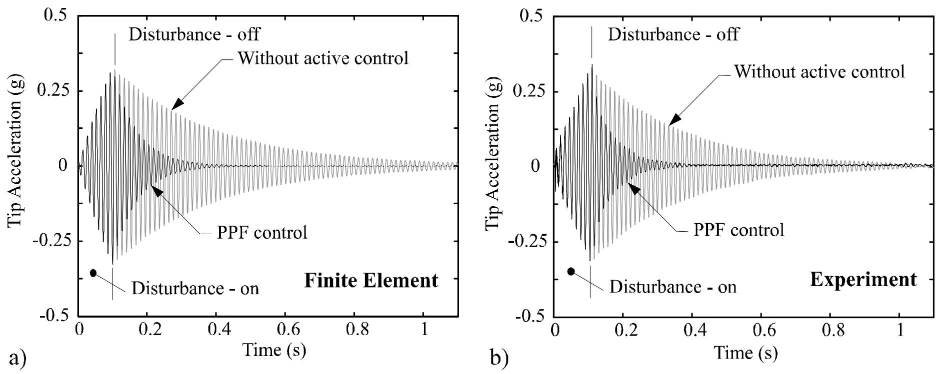

Figure 3: (a) Finite element and (b) experimental comparison of the transient response before and after the PPF controller is initiated.

The transient numerical simulations are depicted in Fig. 3(a) and the corresponding experimental results are shown in Fig. 3(b). It is observed that the transient numerical simulations compare very well with the experimental results. The settling time decreases from approximately 1.0 s to 0.267 s in the numerical simulations when the PPF controller is used. This constitutes a reduction of 73.3% in settling time. In comparison, the experimental settling time decreases from approximately 0.90 s to 0.272 s, resulting in a 69.8% reduction.

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