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Experimental study and numerical simulation of active vibration control of a highly flexible beam using piezoelectric intelligent material

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ABSTRACT

Active vibration control of a flexible beam with piezoelectric pieces on the surface is investigated experimentally using the independent modal space control method, which is able to control the first three modes of the beam independently. A comparison between the responses of the beam before and after control indicates that the modal damping of the flexible beam is greatly improved and the effects of vibration suppression are very remarkable. The dynamic equation of the beam is deduced by Hamilton's principles, and numerical simulation of the active vibration control of the first three modes of the beam is also conducted in this paper. The simulation results match the experimental results very well. Both the experimental and numerical results indicate that by using piezo-patches as actuators the independent mode control method is a very effective approach to realize vibration suppression, and has promising applications in the aerospace field.

1. Introduction

Many aircraft structures, such as the cabin of a space station, solar panels, large space-deployable reflectors, and precise antennae, are characteristically of large size, high flexibility, and low natural frequency. Spacecraft particularly tend to be impacted and disturbed by particulate flows, astro-winds, and other forces while traveling in space so that vibration is produced. As there is almost no damping in space, the vibration will continue for a long time. The strong vibration will affect the operating accuracy and stability of the structures, and long-term vibration may cause fatigue damage to the structures and reduce their service life; such vibration and its accompanying noise are very harmful to the health of people in space and worsen their living environment.

The active vibration control technique has become a research hotspot, due to its advantages of high adaptability and sound control effect. Using piezoelectric ceramics as intelligent materials to suppress the vibration of a flexible structure is an effective approach. Bailey and Hubbard [1] first performed a test of the active vibration control on a cantilever beam by pasting a whole piece of piezoelectric film on the beam. Their research started a new research field of active vibration control using piezo-patches as actuators. Crawley and Luis studied the interaction between the piezo-patches and the substructure and conducted an experiment of the active vibration control on the first mode of a cantilever beam with distributed piezoelectric actuators [4]. Then, various research results have been reported recently [2,3,5–10,12–15]. Baz and Poh adopted the mode control methods to control several vibration modes of a cantilever beam by using one piezoelectric actuator and obtained some control effect, but they didn't realize the simultaneously independent control of each vibration mode [2]. Song, Schmidt and Agrawal adopted Positive Position Feedback (PPF) and Strain Rate Feedback (SRF) to perform a test of the active vibration control of the first two modes of a cantilever beam with piezoelectric actuators and compared the effect of the two control algorithms [14]. Gaudenzi, Carbonaro and Benzi established a finite element model of a cantilever beam with piezoelectric sensors and actuators symmetrically bonded on the upper side and lower side, and conducted an experiment and simulation of active vibration control of the first two modes of the structure by the independent modal space control method [6]. However, most of the existing researches in the field of active vibration control focus on theoretical deduction and numerical simulation [3,5,7–10,12,13,15].

The structures of spacecraft cost extremely high which features light mass and high flexibility. When these structures are affected by external forces and disturbances, the effect of the high-order modal vibration components on the large flexible structures cannot

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be ignored. It requires a series of data processing procedures to realize dynamic control experimentally. These processes include data input, modal smoothing, controlled quantity calculation, data output and so on, which can lead to the non-correspondence of the point-in-time between the inputs and outputs. More modes needed to be simultaneously controlled means that the more control channels are required, which leads to increased computation complexity of the control algorithm and obvious phase shift between the input and output. In the experiments, when the calculated theoretical control amount is used as an input to control the vibration of the beam, the vibration status of the beam has changed and was different from the status when the theoretical data was input. Sometimes, using the theoretical control amount may cause excitation or other vibration modes. Thus, compared with the pure theoretical calculation, the effective active vibration controls on the relatively complex, highly flexible structures with high order vibration modes are difficult to be realized.

Currently, only the control of lower-order modes (i.e., the first and second-order modes) of the beam structure's has been reported in published literatures, in which the active control is experimentally realizable [1,2,4,6,14]. According to the retrieved results from the existing literatures, the realization of active vibration control of the third order modes of large flexible structures was seldom reported so far.

The independent modal method controls the maximum deformed positions of each order of mode of the large flexible structure at the same time. It will take a short time to suppress the vibration and get a good control effect, but too many channels need to be simultaneously controlled, which can cause large calculation amount and phase shift between the input and output. For this reason, practical implementation of this method is quite cumbersome. This research realizes the independent control of the first three modes of the large flexible cantilever beam by independent modal method. Meanwhile, Hamilton's principles are used to deduce the dynamic equation of a flexible beam with piezo-patches, and then the numerical simulation of active vibration control of the first three modes of the beam is performed. It shows the results of numerical simulation are coincident with the results of experiments. This research lays a foundation for the realization of the complex multi-channel active control of the higher-order large flexible beam structures or shell structures in the future.

2. Dynamic equation and independent mode control method

2.1. Dynamic equation

According to vibration theories, when a beam is subjected to the transversely dynamic loads, the bending vibration differential equation of it expressed with the displacement can be described in [16] as

$$E_\mathrm{b} I_\mathrm{b} \frac{\partial^4 w}{\partial x^4} + c\frac{\partial w}{\partial t} + \rho_\mathrm{b} A_\mathrm{b} \frac{\partial^2 w}{\partial t^2} = f \tag{1}$$

where E_b and I_b are, respectively, the modulus of the elasticity and the area moment of inertia of the cross-section; c is the damping coefficient of the structure; ρ_b and A_b are the density and the cross-sectional area of the beam; f is the transversely dynamic forces; w is the transverse displacement of the structure.

When the transverse vibration of the beam is controlled through the piezo-actuators, the transversely dynamic forces f on the right side of Eq. (1) are composed of two components, the transverse initial exciting forces and the control forces caused by the actuators, which can be expressed as

$$f = f_\mathrm{e} + f_\mathrm{v} \tag{2}$$

where f_e is the transverse initial exciting forces acting on the beam; f_v is the transverse control forces applied on the beam.

It is known according to the research results presented by Crawley and Luis in [4] that the force caused by the piezo-actuator bonded on the beam surface under the voltage can be equivalent to a bending moment upon the beam. Therefore, when there are many piezo-actuators dispersedly arranged on the beam, and the vibration of the beam is controlled through these actuators, the control forces caused by these piezo-actuators can be written as

$$f_\mathrm{v} = \frac{\partial^2}{\partial x^2} \sum_{i=1}^{n} k\varphi_i L_i(x) \tag{3}$$

In Eq. (3), k is a constant which relies on the Young's modulus and the size of the beam and the piezo-patches; φ_i is the voltage applied on the ith piece of piezo-patch; $L_i(x)$ is the local function of the piezo-patches, which can be expressed with Heaviside function $H(x)$ as

$$L_i(x) = H(x - x_{i1}) - H(x - x_{i2}) \tag{4}$$

where x_{i1} and x_{i2} are the coordinates at the left and the right end of the ith piece of piezo-patch.

Let's express the transverse displacement of the beam w by the first m-order modes, by substituting Eq. (3) into Eq. (2), then substituting the result into Eq. (1), in terms of the orthogonality of the normalized mode shape, after some simplifications, the dynamic equation of the system, when the flexible beam is only subjected to the control voltage $\boldsymbol{\varphi}$, can be expressed with the modal coordinate $\boldsymbol{\eta}$ as

$$\ddot{\boldsymbol{\eta}} + \boldsymbol{C}\dot{\boldsymbol{\eta}} + \boldsymbol{K}\boldsymbol{\eta} = \boldsymbol{K}_\psi \boldsymbol{\varphi} \tag{5}$$

where $\boldsymbol{\eta} = \{\eta_1, \eta_2 \ldots \eta_m\}^\mathrm{T}$, is the modal coordinate; $\boldsymbol{C}$ is the damping matrix; $\boldsymbol{K}$ is the modal stiffness matrix; $\boldsymbol{K}_\psi$ is the electromechanical coupling matrix, which relies on the material, the size, the mode shape of structure, and the location of the piezo-patches. Let

$$\boldsymbol{K}_\psi \boldsymbol{\varphi} = \boldsymbol{F}_\mathrm{c} = \{F_{\mathrm{c}1}, F_{\mathrm{c}2}, \cdots, F_{\mathrm{c}m}\} \tag{6}$$

In Eq. (6), $\boldsymbol{F}_\mathrm{c}$ is the modal control force vector. Then Eq. (5) can be written as

$$\ddot{\eta}_i + 2\omega_i \xi_i^* \dot{\eta}_i + \omega_i^2 \eta_i = F_{\mathrm{c}i} \quad (i = 1, 2, \cdots, m) \tag{7}$$

where η_i is the ith modal coordinate, ω_i is the ith natural frequency, ξ_i^* is the ith damping ratio, and $F_{\mathrm{c}i}$ is the ith modal control force.

2.2. Independent mode control method

It is known from Eq. (7) that all modes of the open-loop system are independent. Therefore, the vibration control of the whole structure can be converted into the control of each mode. Eq. (7) is the foundation of the independent mode control method. If speed negative feedback is adopted to control every mode, let the modal control force be

$$F_{\mathrm{c}i} = -g_i \dot{\eta}_i \tag{8}$$

where g_i is the ith-*order* modal velocity gain, then the equation of the closed-loop system after control is given as

$$\ddot{\eta}_i + \left(2\omega_i \xi_i^* + g_i\right)\dot{\eta}_i + \omega_i^2 \eta_i = 0 \tag{9}$$

As known from Eqs. (7) and (9), all modes of the structure are decoupled from each other before and after control. This is also called the independent modal space control method. Letting the damping ratio of the ith mode after control be ξ_i, then the following equation can be obtained from Eq. (9).

$$g_i = 2\left(\xi_i - \xi_i^*\right)\omega_i \tag{10}$$