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Reduction of radiated exterior noise from the flexible vibrating plate of a rectangular enclosure using multi-channel active control

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ABSTRACT

This study attempted to control the radiated exterior noise from a rectangular enclosure in which an internal plate vibrates by acoustic excitation and noise is thus radiated from that plate. Multi-channel active control was applied to reduce the vibration and external radiation of this enclosed plate. A piezoelectric ceramic was used as a distributed actuator for multiple mode control of the vibration and radiated noise in the acoustically excited plate. To maximize the effective control, an approach was proposed for attachment the piezoelectric actuator in the optimal location. The plate and internal acoustic space in the enclosure are coupled with each other. This will change dominant frequency characteristics of the plate and, thus, those of the externally radiated noise. Active noise control was accomplished using an accelerometer attached to the plate and a microphone placed adjacent to that plate as an error sensor under acoustic excitation of sine wave and white noise. It was found that the control of radiated external radiation noise requires a microphone as an error sensor, a sound pressure sensor due to vibration of the plate, differences in the dominant frequency of externally radiated noise, and complex vibration modes of the plate.

1. Introduction

Vibration reduction and acoustic radiation caused by vibration in an elastic structure has emerged as a very important concern in noise and vibration research. Existing passive reduction methods of increasing attenuation and adding mass and stiffness to a structure has weaknesses in terms of energy efficiency and installation space. Other methods of passively reducing radiated noise include the use of various sound-absorbing or soundproofing materials. These methods, however, generally have weaknesses in that noise reduction is difficult at low frequencies below 500 Hz. This is because low-frequency noise has a long wavelength, which requires the use of thick sound absorbing and soundproofing materials to eliminate. Thickening of the sound absorbing and soundproofing materials also has limitations because of the restrictions of small installation spaces [1]. Therefore, an alternate active control approach has been studied, which actively offsets the vibration and acoustic energy of a structure using additional energy [2,3].

Effective active control performance requires an appropriate control algorithm and an excitation system. For the algorithm, a digital signal processor (DSP) has been developed [4] along with a new signal processing technique. This makes possible real-time active control, a more advanced approach than previous computer simulation control. On the excitation system side, existing point-excited systems have drawbacks in that the support section is affected by the reaction force and that additional space is needed to install the excitation system. It is also difficult to generate simultaneous reduction effects in all modes using this type of excitation system [5,6]. For these reasons, piezoelectric materials [7], including piezoelectric ceramics such as Lead Zirconate Titanate and polymer-based piezoelectric films, have been studied as a more practical excitation system, since they are advantageous due to their distributed nature, price, efficiency, and volume, compared to conventional excitation systems [8,9].

Several studies of beam vibration control have been conducted in recent years. Baz and Poh [9] compared the vibration control performance of flexible beams according to the placement of the piezoelectric. Gibbs and Fuller [10] investigated active control of vibration power flow during a sinusoidal disturbance input using a piezoelectric element as an actuator and the least mean square (LMS) algorithm. Besides, Burdisso and Fuller [5] found that control

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effects decrease in the high-frequency band during active vibration control of beams using a shaker. Brennan et al. [6] conducted vibration control simulations of beams using a point-loaded actuator or piezoelectric. However, these studies focused on vibration control simulations or experiments using a point-excited actuator or piezoelectric. There have been few studies on the optimal placement to enable the actuator to exercise maximum control sensitivity. Therefore, it is necessary to select the optimal attachment placement of the piezoelectric to achieve maximum efficiency.

For vibration control of the plate, Guigou et al. [11] performed vibration control of a facing plate with 2 edges clamped and the other 2 edges free. Pan and Hansen [12] simulated active vibration control of a plate with 1 edge clamped, the opposite edge free, and the other two edges simply supported as a shaker. However, these studies did not deal with optimization of the actuator placement and their results differ from those using a type with 4 clamped edges, which actually often exists as a testing plate. The optimal position of the piezoelectric actuator for vibration control of a test plate with 4 clamped edges should thus be selected, followed by analysis of the vibration control performance accordingly.

For noise control of the plate, Fuller [13] optimally controlled the radiated noise of a plate using a point-loading actuator. Pan and Hansen [14,15] controlled the noise transferred in the enclosure through a plate using a point-loading actuator. Metcalf et al. [16] also controlled the radiated noise of a plate using a point-loaded actuator. However, these studies have a particular weakness, in that a point loaded actuator must be installed using an experimental process. This causes a spillover problem, leading to poor noise reduction performance. Keltie [17] manually controlled radiated noise in the medium-to-high frequency range above 500 Hz by attaching a rib to the plate. In that study, the control effect was found to be poor in the low frequency band of 500 Hz and below because of the manual control method used. Clark and Fuller [18] controlled radiated noise of a plate supported on all 4 edges using a piezoelectric element and a filtered-x LMS algorithm. Koshigoe et al. [19–21] subsequently performed acoustic transmission control into an enclosure through a plate from an outdoor noise source using a piezoelectric. Elliott [22] described several approaches for active control of structure-borne noise, including global control, local control, feedback control, and feed-forward control. Maillard and Fullerb [23] attempted to reduce the radiated noise of a plate, thus studying methods to control the radiated noise by attaching several accelerometers to the plate and estimating the far-field sound field. Charette et al. [24] decreased the radiated noise of a plate using PVDF as an error sensor and a piezoelectric as a shaker. However, there are also drawbacks to these studies, such as the number of accelerometers used or the designed shape of the PVDF. In order to compensate for these, the control system needs to be configured using two microphones as an error sensor.

This study attempted to carry out active vibration and noise control using a multi-channel filtered-x LMS algorithm by determining the optimal attachment position of the piezoelectric actuator to reduce vibration and externally radiated noise of a flexible plate in an acoustically-loaded enclosure. To that end, rather than using a point-loaded actuator, a piezoelectric ceramic was attached to the optimal position to maximize control efficiency for an enclosure with 4 clamped edges and thus reduces the vibration of the enclosed plate. A single-channel filtered-x LMS algorithm provided simultaneous control in several vibration modes in a 1-dimensional cantilever system, but showed poor control performance in a two-dimensional structure with its more complicated vibration mode. To overcome this disadvantage, active vibration control for the enclosed plate was improved by introducing a multi-channel filtered-x LMS algorithm. To select the optimal attachment position of the piezoelectric actuator on the plate, a control sensitivity equation was proposed. From the proposed control sensitivity equation, the optimal attachment placement that ensures maximum control sensitivity was determined. To prove the validity of the optimal attachment position, multi-channel active control was performed in the calculated optimal position. The dominant frequency characteristics of the plate in the enclosure tend to differ from those of the externally radiated noise due to coupling with the acoustic space. This study attempted to perform global control of radiated external noise by selecting and using an error sensor as an approach to control the radiated external noise in the coupled system.

2. Active vibration control of enclosed plate using multi-channel control

2.1. Multi-channel filtered-x LMS algorithm

For a control system using L error sensors and M shakers, the objective function can be expressed by the following mean-squared error.

$$J = \sum_{l=1}^{L} \bar{e}_l^2(n) \tag{1}$$

where $e_l(n)$ represents the signal of the l-th error sensor. This is denoted by $d_l(n)$ generated by the main noise source and the differences in sums of individual actuators.

$$e_l(n) = d_l(n) - \sum_{m=0}^{M}\sum_{j=0}^{J-1} C_{lmj} \sum_{i=0}^{I-1} W_{mi} X(n-i-j) \tag{2}$$

where C_{lmj} refers to the j-th finite impulse response filter coefficient for modeling a transfer function between the actuator m and the error sensor l.

Here, if the reference signal $X(n)$ has a correlation with $d_l(n)$, the objective function J can reduce the main noise source by g. Accordingly, the change in the adaptive convergence factor that minimizes in the objective function J using a steepest descent method can be found as follows:

$$\frac{\partial J}{\partial W_{mi}} = 2\sum_{l=1}^{L} e_l(n)\frac{\partial e_l(n)}{\partial W_{mi}} \tag{3}$$

Then, the weight-updating formula can be denoted by the following equation.

$$W_{mi}(n+1) = W_{mi}(n) - \mu^* \sum_{l=0}^{L} e_l(n) \sum_{j=0}^{J-1} C_{lmj} X(n-i-j) \tag{4}$$

In this study, a multi-channel filtered-x LMS algorithm was used as an error sensor and a shaker. In other words, a 2-algorithm combined form was used, as shown in Fig. 1 below.

2.2. Prediction of structure-borne noise of vibrating panels

Assume that the vibrating panel shown in Fig. 2 lies in the center of the face. Then, the pressure of the sound radiated by the vibration of the plate can be expressed by Rayleigh's integral equation.

$$p(x,y,z) = i\omega\rho/2\pi \iint_S w(x',y',0)\exp(-ik|r-r'|)/|r-r'|dx'dy' \tag{5}$$

The equation above is useful for predicting the field of the sound radiated under the given distribution of the mechanically excited vibration surface. When the boundary condition is under a clamped end, the vibration structural-radiated noise of the panel with piezoelectric materials can be rewritten as follows: