Active Vibration Control of Flexible Cantilever Beam Using Piezo Actuator and Filtered-X LMS Algorithm

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This paper presents the active vibration control of a flexible cantilever beam. The cantilever beam was excited by steady-state sinusoidal and white noise point forces. The vibrational control system was implemented using one piezo ceramic actuator bonded on the beam and the adaptive controller based on the Filtered-X LMS algorithm. Control results indicated that a considerable vibrational reduction could be achieved in a few seconds. Experimental results, demonstrate the feasibility of active vibration control of the flexible cantilever beam based on piezo ceramic actuator and the Filtered-X LMS algorithm.

Key Words: Active Vibration Control, Piezo Ceramic Actuator, Filtered-x LMS algorithm, Flexible Cantilever Beam.

1. Introduction

Structural vibration at the low frequency range has been a persistent problem in lightweight and flexible structures such as transportation devices and consumer products. Large space structures are especially very flexible since they are large in size and have structural light damping. Once these lightweight and flexible structures are excited by external forces, they vibrate continuously on their natural frequencies.

The conventional passive methods of increasing damping and adding mass or stiffness to these flexible structures may have disadvantages on the efficiency or cost. Therefore, active control methods have been investigated in the past decade as an alternative to the conventional passive techniques for reducing low frequency vibration. (Baz, 1988). To implement an efficient active vibration control system, an adequate actuator should be selected. Since the conventional point exciting actuator is not applicable to these lightweight structures devices based on piezoelectric materials such as PZT and PVDF are studied as practical alternatives (Mason, 1981). These piezoelectric transducers have several advantages from the viewpoint of weight, volume, shape and efficiency compared to the conventional ones. Furthermore, piezoelectric materials are bonded directly to the structures and serve as distributed internal moment generating actuators (Hong, 1993).

In recent years, the intelligent control methods by neural networks or fuzzy inference techniques have been studied to overcome the non-linear characteristics of structures. Digital control methods by the LMS algorithm with the DSP board were also widely studied (Shin, 1994 ; Kim, 1992 ; Vipperman, 1993 ; Shin, 1996). Gibbs and Fuller (1990) studied the active control of vibrational power flow for flexible beam by using PZT and LMS algorithm (Widrow, 1985) when the sinusoidal disturbance applied. Burdisso and Fuller (1992) performed a computer simulation and an experiment on the eigenproperties of controlled beam systems by using the Filtered-X LMS algorithm when the

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sinusoidal disturbance applied.

In this study, the Filtered-X LMS algorithm implemented on the DSP board were used as a real-time control strategy to reduce the vibration of the flexible beam, where the PZT actuater is bonded, when the sinusoidal and white noise disturbances are applied. Furthermore, vibrational intensity was measured to identify and visualize the vibrational power flow through the cantilever beam when the white noise is applied

2. Filtered-x LMS Algorithm

The signal measured by an error sensor comprises two parts, the primary signal, p_k, and the secondary source, s_k. If two signals have the same amplitude and opposite phases, the vibration at the error sensor position could disappear.

$$e_k = p_k + s_k \tag{1}$$

The secondary signal obtained at the error sensor position at time k does not coincide with the signal excited by the controller at time k. This is because, the control signal has a delay of n samples due to the finite distance between the actuator and the error sensor. Furthermore, the control signal is modified by the characteristic transfer function(A) of the actuator and the total error signal is also modified by the error sensor transfer function(M). Therefore, the practical error signal at time k is represented by Eq. (2).

$$e_k = (p_k + W_{k-n}^T X_{k-n} A) M \tag{2}$$

where, X represents the reference signal.

If the pure tone signals are considered, the secondary path transfer functions between the error microphone and the control actuator could be assumed as simple gains and phase shifts. However, in the case of broadband noise, they must be thought of finite impulse response filters, or vector quantities.

To simplify Eq. (2), the filtered primary source signal, G, and filtered reference signal, F, at time k are defined as follows :

$$G_k = p_k M, \ F_k = X_{k-n} A M \tag{3}$$

The estimated gradient based on the instantaneous squared error signal is

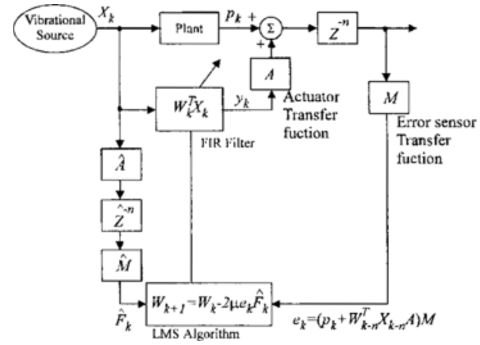


Fig. 1 Block diagram of the practical implementation of active vibration control.

$$\bar{\nabla}_k = \frac{e_k^2}{W} = 2 e_k F_k \tag{4}$$

Therefore, the modified LMS algorithm can be obtained using Eq. (5),

$$W_{k+1} = W_k - 2\mu e_k F_k \tag{5}$$

A block diagram of one possible implementation of the algorithm in Eq. (5) is shown in Fig. 1.

The reference signal, X, is delayed by an estimated value of n samples, then convoluted with estimated transfer functions.

$$\hat{F}_k = \hat{X}_{k-n} \hat{A} \hat{M} \tag{6}$$

Therefore, the practical implementation of the algorithm can be defined as follows :

$$\begin{aligned} W_{k+1} &= W_k - 2\mu e_k \hat{F}_k = W_k \\ &\quad - 2\mu \hat{F}_k (G_k + W_{k-n}^T F_k) \\ &= W_k - 2\mu(\hat{F}_k G_k + \hat{F}_k F_k^T W_{k-n}) \end{aligned} \tag{7}$$

3. Experimental Analysis and Discussion

3.1 Experimental set-up and procedure

Figure 2 represents the experimental set-up for the study. The aluminum cantilever beam had a length of 34cm, a width of 3cm and a thickness of 0.7mm, respectively. The longitudinal modulus of elasticity of the aluminum cantilever beam was 6.21×10[10] Pa. The disturbance was applied vertically to the free-end of the beam as a point force by using a shaker. Piezo ceramic was bonded to the fixed-end of the beam as an actuator and its dimension was 5cm×1.2cm×0.05cm.

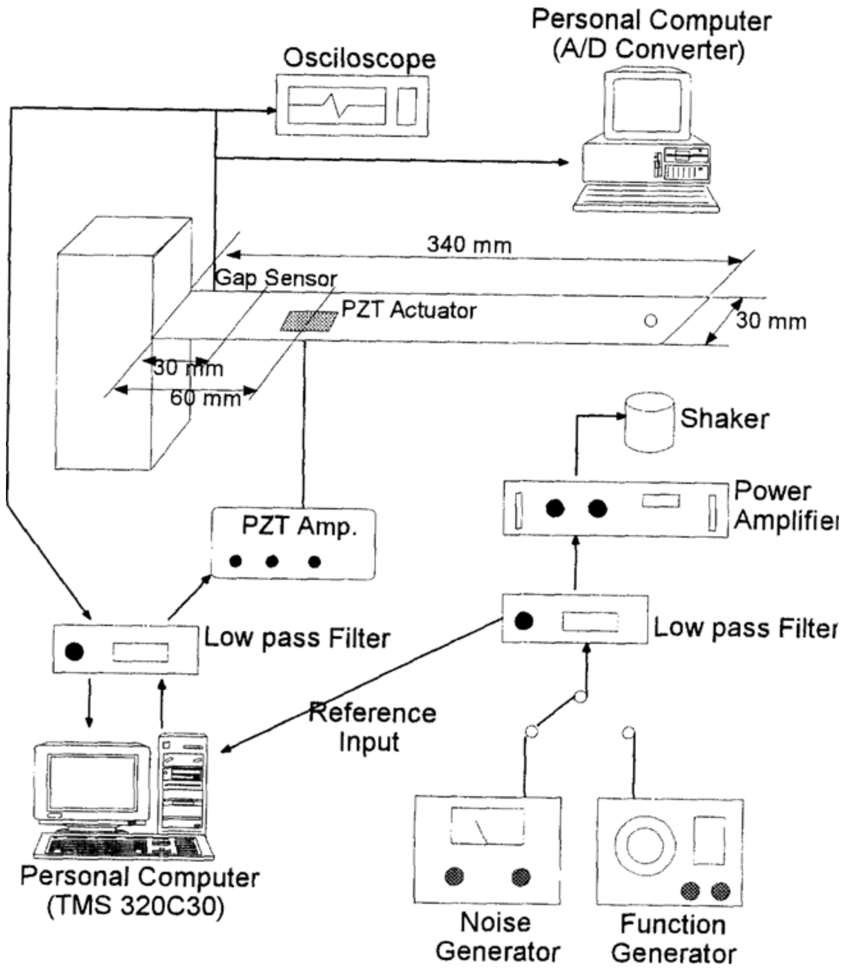


Fig. 2 Experimental apparatus for active vibration control.

The reference signal was generated by the function generator and the noise generator, respectively. Both of the reference and error signals were filtered through lowpass filters with the cut-off frequency of 300Hz. The control signal was amplified by a PZT Amp. (Max. output of ± 170 V) and the error signal was detected by a gap sensor. Oscilloscope(HP 54503A) and FFT(SA390) analyzer were used to monitor and analyze the desired signals. The main control devices were composed of DSP board(TMS320C30) and an A/D converter(NANOTECH) built in a 486 IBM computer.

The sampling rate of the DSP board was 1000Hz and the number of coefficients of the FIR filter for system modelling was 150. The control codes for DSP were written in assembly language and the interface programs for down-loading were written in C language.

The experimental procedure consisted of the following 5 steps.

1) Finding out the natural frequencies of the cantilever beam system by exciting it with a white noise signal.

2) Measuring the error signal at the specified position by exciting the beam system with sinusoidal signals which coincided with the obtained natural frequencies of the beam system in step 1.

3) Measuring the error signal at the specified position by exciting the beam system with a white noise signal.

4) Obtaining time series and frequency data from experiments in step 2 and 3.

5) Measuring the vibrational intensity at each point of the cantilever beam to identify the vibrational power flow.

3.2 Experimental results and discussion

Figure 3 shows the calculated transfer function of the cantilever beam system.

Figures 4 to 6 represent the frequency data and the time series of the error signal before and after the control when the beam was excited by natural frequencies of 24Hz, 66Hz and 140Hz, respectively.

The control results indicate that more than 20dB of vibrational reduction could be achieved over the considered frequency range when the sinusoidal disturbances were applied. It took about 10 seconds to control the vibration level to 90 percent of the residual error signal at the first natural frequency of the cantilever beam and it took about 2 seconds at other natural frequencies.

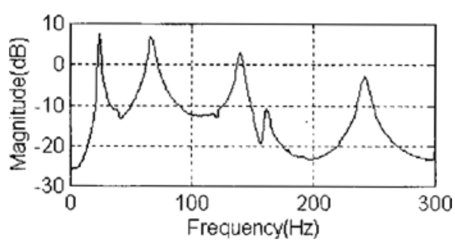


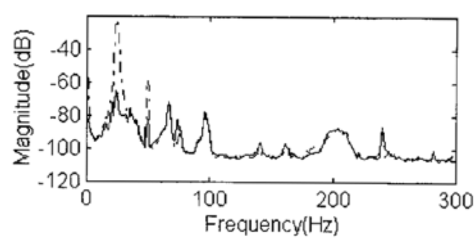
Fig. 3 Transfer function between shaker and gap sensor.

Figure 7 shows the frequency data and the time series of the error signal before and after the control when the beam was excited by a lowpass filtered white noise signal with a cutoff frequency of 300Hz.

The control result shows that the vibrational reduction of more than 7 dB could be achieved at the first natural frequency of the beam system and the vibration level of the beam decreased through the entire frequency range erenly.

However, Fig. 7 shows some shifting of peak frequencies after the control. It may be thought of as i) dislocation of the sensor/actuator system, ii) superposition of the exciting wave by the shaker and the deflecting wave by the fixed-end of the beam.

Finally, Fig. 8 shows the vibrational power flow on the cantilever beam when the beam was excited by the white noise signal. It can be seen that the vibrational power flow on the cantilever



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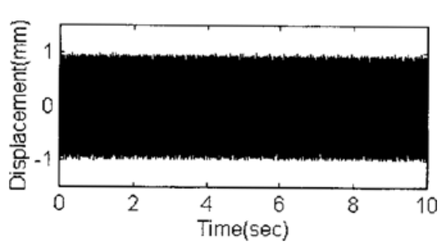
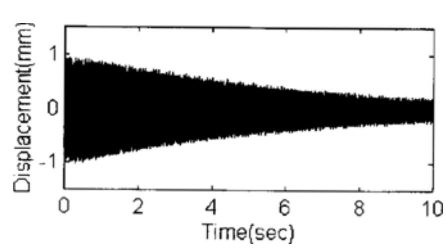
(a) Frequency analysis

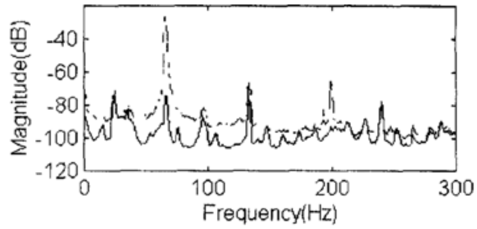


(b) Without AVC



(c) With AVC

Fig. 4 Results of excitation with the 1st natural frequency.



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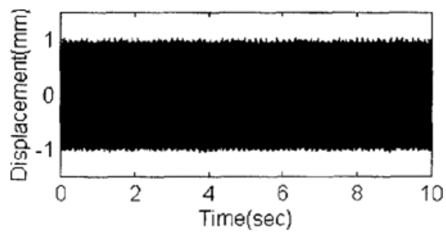
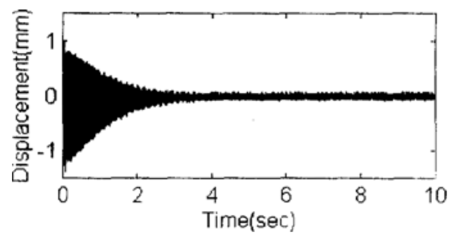
(a) Frequency analysis

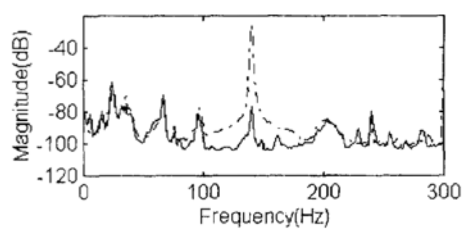


(b) Without AVC



(c) With AVC

Fig. 5 Results of excitation with the 2nd natural frequency.



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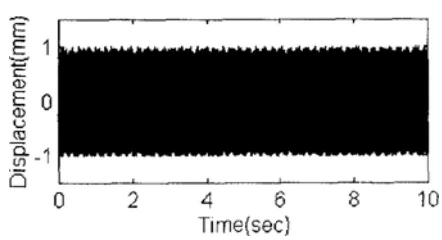
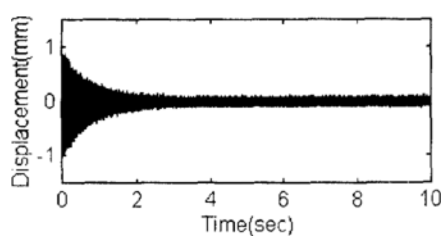
(a) Frequency analysis



(b) Without AVC



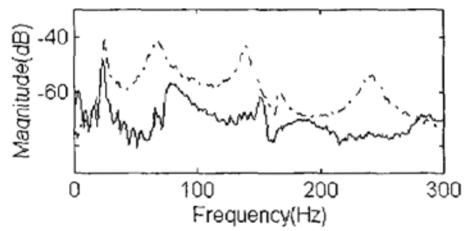
(c) With AVC

Fig. 6 Results of excitation with the 3rd natural frequency.

beam disappears practically after the control.

4. Conclusions

It was possible to control the cantilever beam system in real-time by using a PZT actuator and Filtered-X LMS control algorithm embodied in the DSP board when the arbitrary disturbances were applied. In the case of sinusoidal input, the levels of vibrational reduction to each exciting



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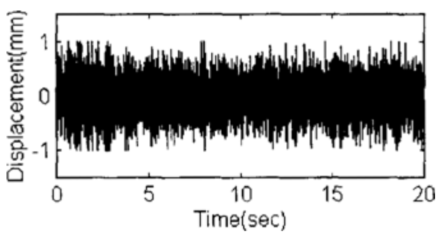
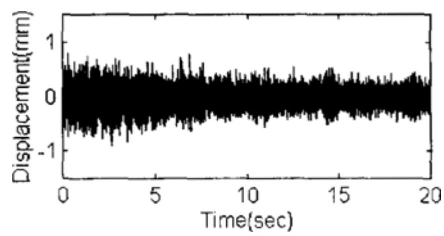
(a) Frequency analysis

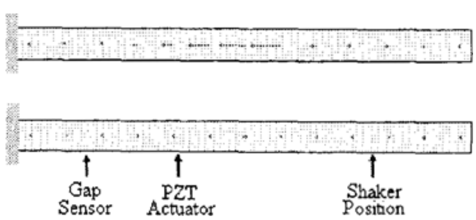


(b) Without AVC

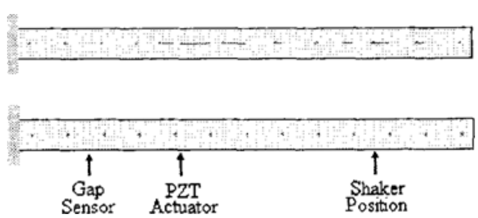


(c) With AVC

Fig. 7 Results of excitation white 300Hz lowpass filtered white noise.



(a) 1st mode(24Hz)



(b) 2nd mode(66Hz)

Fig. 8 Vibrational power flow on white noise excitation.

frequencies were remarkable. Also, in the case of white noise input, considerable vibrational reduction could be achieved at the first natural frequency of the beam and the whole frequency range considered. Furthermore, it could be identified that the energy flow on the cantilever beam disappeared after the control by the vibrational intensity method.

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