平板聲輻射之多點適應性主動控制

The multi-point active control on the acousticradiation from a plate

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摘要

本文是針對一四邊固定之矩形薄平板受外力激振時所產生的聲音輻射，進行兩點之適應性主動振動控制來抑制輻射聲功率。實驗採用方形及圓形陶瓷壓電片，以作為激振器以及控制器，乃是利用其具有質量輕且在不同頻率下皆有甚佳的電力-機械的線性反應之特性，且方形陶瓷壓電片可藉由理論之精確分析，而圓形陶瓷壓電片可產生圓形波，在任何頻率下皆可均勻的產生振動，使控制效果達到吾人所期望的效果。控制系統方面，使用 FIR 數位濾波器及 LMS 演算法作為控制系統核心，藉由增加控制點及適當的放大或縮小控制力及改變相位，使控制系統更加完備，達到無論在任何頻率下聲功率皆能大幅下降的效果。

由實驗結果可知，在共振頻率下（119，163，223Hz；148，187，255Hz）輻射聲功率能確實的降低 13dB 以上，其他非共振頻率也有 12dB 左右的控制效果，顯示有計畫的增加控制點，以及經由多次反覆實驗改變控制力之振幅及相位，能夠使適應性主動振動控制效果更趨完善。

關鍵詞:1.聲輻射　2.多點控制　3.適應性

Abstract

This study is mainly to study the active control on the acoustic radiation from a plate clamped at edges and excited by a force at resonant frequency.

Regarding the active control experiment involved, a PZT is used as the required actuator, and two another ones are used to be the controller. From dynamical analysis, the response from a rectangular PZT can be more accurately predicted. However, a circular PZT can uniformly cause vibration mode on the plate. Upon the above two situations, a better performance of active vibration control at low frequency can be achieved. Also, a FIR filter with LMS algorithm is chosen as the control system required. Either adopting more control points, or properly reducing

control force, or changing phase appropriately, might drive the plate to radiate much less acoustic energy.

The result obtained in this study reveals almost 13-decibels attenuation of acoustic radiation power at some resonant frequencies is achieved. Otherwise for some other non-resonant frequencies, quite the same attenuation of acoustic radiation is also obtained. As regards to the comparison with the result for the single-control point case, 8 decibels improvement of acoustic radiation power is approximately obtained.

Keywords: 1. Acoustic radiation. 2. Multi-point control. 3.Adaptive.

1. Introduction

Regarding the noise control techniques in acoustics applications, a passive and an active one are involved [1,2]. Usually, an active control method is used to reduce the acoustic field at low frequency, especially lower than 800 Hz. In this study, it deals with the active control on the acoustic radiation from a plate acted on by a force of frequency lower than 600 Hz.

Speaking to the topics as described, it is able to attenuate the acoustic radiation involved by using a PZT actuator to actively control the amplitude of vibration produced by an applied force [3,4]. From the past studies [3~9] involved in structural dynamics, two effective mechanisms, which are modal suppression and modal restructuring, are involved. The plate involved in this study is applied by a PZT actuator, and two another actuators are used as controller to reduce the amplitude of the highest radiation mode. Therefore, the second one is more effectively used.

Upon the above situation, this study is to concentrate on the multi-point control on the acoustic radiation from a plate, in which, three PZT are used as actuator and controller, a FIR filter with LMS algorithm is chosen as the required control system, and the adaptively active vibration control experiment at some frequencies is also performed. Its results show that almost 13 decibels attenuation of acoustic radiation power at some resonant frequencies is achieved. Furthermore, the comparison of the results between two cases for the multi-point control PZT and the single-point control PZT reveals 8 decibels improvement of the acoustic radiation power is obtained.

2. Theories and experiments

As shown in Fig.1, a rectangular plate is clamped at four edges and is applied by a force $P_e(x, y, t)$ of frequency ω. Modal analysis shows the plate dynamic response can be expressed in terms of the associated first p normal modes as:

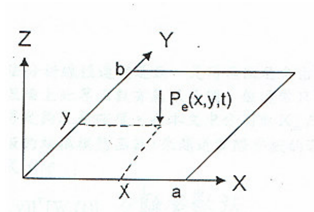

Fig.1 The coordinates system for a rectangular plate to be used

$$w(x,y,t) = [\Phi_p]^T [W_p] \tag{1}$$

In eqn(1), both of $[\Phi_p],[W_p]$ are p×1 column matrix with the elements defined by shape function and its relating response. Meanwhile from the theory of structural dynamics, the equation of motion for the plate dynamic response can be expressed in matrix form as [5,6]:

$$[M]\left[\ddot{W}_p(t)\right] + [K][W(t)] = [F(t)] \tag{2}$$

Where $[M]$ and $[K]$ are, respectively, the p×p diagonal mass and stiffness matrix, of those have the diagonal element having the form as that in Huang's study, and $[F(t)]$ is a p×1 row matrix with the element having the same form as that in Chiang's study [9]. Using eqn(2) can define the characteristic equation of plate as：

$$\left|[K] - \omega^2 [M]\right| = 0 \tag{3}$$

The roots of eqn(3) are the modal frequencies of plate. And the theory of structural dynamics tells us the modal contribution of acoustic radiation from a plate is much greater for lower one. Therefore, using PZT controller to actively control the dynamic response due to contribution from the lowest mode can reduce the overall acoustic radiation from a plate to a great extent.

As shown in Fig.2, a PZT actuator is affixed at some point of plate. When a harmonic driving electrical signal is applied to PZT actuator, the resulting tension and compression will cause the plate to be equivalently acted on by four equally concentrated bending moments at mid-point of four edges of PZT actuator. From some past studies, the radiation function of the first mode at lower frequency is almost a constant, which is proportional to the net volume velocity of the plate when acted on by a PZT actuator. Therefore, it is possible to actively attenuate the acoustic radiated power from the plate by means of the volume velocity cancellation method. Following Johnson and Elliott's study [7], it is possible to cause the radiated power from plate to be minimum when the bending moment produced by PZT actuator is

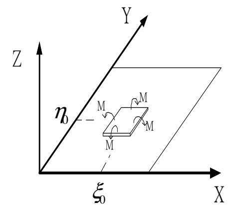

Fig.2 Equivalent bending moment system to a PZT actuator affixed on a plate

adjusted to the following value:

$$g^c = -\frac{[q][v^i]}{[q][v^c]} \tag{4}$$

All matrices involved in the above equation are defined in Johnson and Elliott's study [7] and not repeated here.

In Fig.3, the spherical coordinates system of a plate is shown. When this plate is in harmonic motion with displacement $w(x, y)e^{j\omega t}$, the net volume velocity from it is:

$$V_{total} = j\omega\int_0^{l_y}\int_0^{l_x} w(x, y)dxdy \tag{5}$$

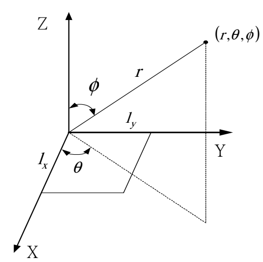

Fig.3 The far-field point representation in a spherical coordinates system

And the sound pressure at a far field point (r, ϑ, Φ) can be expressed in terms of Rayleigh integral as [5]:

$$p(r, \vartheta, \Phi) = \frac{j\omega\rho e^{-jkr}}{2\pi r}\int_0^{l_y}\int_0^{l_x} j\omega w(x, y)e^{j(k_x x + k_y y)}dxdy \tag{6}$$

where $k = \frac{\omega}{c}$ is the wave-number of the radiated sound wave, and $k_x = k\sin\vartheta\cos\Phi$, $k_y = k\sin\vartheta\sin\Phi$ are the associated components in x-, y-directions. When only the far field point on the vertical axis to the center of the plate is considered, then the contrast of eqn(6) with eqn(5) will reveal the radiated acoustic pressure at such far-field point is proportional to the net volume velocity produced by the plate. Therefore, we can place some microphones as acoustic sensor at the points as described, and then can combine some adaptive control system with the corresponding algorithm to actively control the net volume velocity produced from the plate. In this study, we use a FIR (finite impulse response) adaptive filter and a LMS (least mean square) algorithm for the adaptive control system involved, which is the same as that used in Chiang's study. Its theory in detail is not discussed here.

Following the theoretical consideration as described, we can perform the adaptively active control experiment involved in this study in the Acoustic Laboratory of the Department of Engineering Science and Ocean Engineering, National Taiwan University. Fig.4 shows the arrangement of the required experimental setup. The all devices involved in Fig. 4 are also the same as that in Chiang's study and is not discussed here. Following the experimental setup as shown in Fig.4, we choose three circular or three square PZT, one of them is used as an actuator and the other two are used as controller, for each-time experiment involved. Regarding the plate and the acoustic frequency used for the active vibration control experiment, the 120-centimter square glass plates of 5, 8 mini-meters in thickness, 6 resonant frequencies 119, 148, 163, 187, 223, 250 Hz, and 5 non-resonant frequencies 300, 350, 400, 450, 500 Hz are selected respectively.

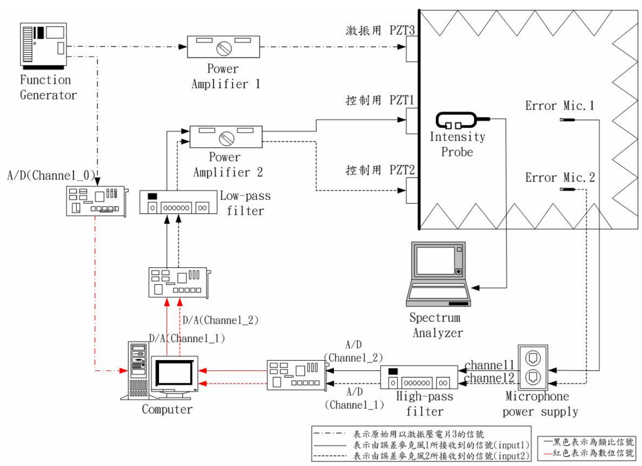

Fig.4 Arrangement of the setup for the active control experiment involved

3.Results and conclusions

Some experimental results expressed in acoustic intensity level for either two control points or one control point active vibration control by using either square or circular PZT as sensor are, respectively, listed in Table 1 to 6. Among them, they clearly reveal the active control effectiveness on the reduction of acoustic power radiated from the plate is mostly better for both one point and two points control at lower resonant frequency. Meanwhile, the control effectiveness by using two-point active control is also better than that for one-point control case for all frequencies to be measured. The difference of acoustic power reduction between them is mostly around 5 decibels. Regarding the control effectiveness for the use of the PZT type, a better result is obtained for the circular type PZT at non-resonant frequency and for the rectangular type PZT at resonant frequency.

Table.1　acoustic radiation intensity by two-point rectangular PZT active control (5mm glass)

Frequency (Hz)	119	163	223	300	350	400	450	500
Before control (dB)	58.4	60.5	60.4	63.1	65.7	70.8	66.2	66.7
After control (dB)	44.2	43.3	42.3	51.1	53.7	60.7	53.8	46.7
Difference (dB)	14.2	17.2	18.1	12.0	12.0	10.1	12.4	20.0

Table.2　acoustic radiation intensity by one-point rectangular PZT active control (5mm glass)

Frequency (Hz)	119	163	223	300	350	400	450	500
Before control (dB)	58.2	59.6	60.3	61.7	64.6	70.4	66.1	66.4
After control (dB)	48.0	49.6	46.8	54.7	57.5	64.1	56.8	51.8
Difference (dB)	10.2	10.0	13.5	7.0	7.1	6.3	9.3	14.6

Table.3　acoustic radiation intensity by two-point circular PZT active control (5mm glass)

Frequency (Hz)	119	163	223	300	350	400	450	500
Before control (dB)	61.0	61.0	61.3	61.5	61.4	61.0	58.7	68.6
After control(dB)	44.3	45.4	46.7	47.0	46.8	46.9	45.0	53.3
Difference (dB)	16.7	15.6	14.6	14.5	14.6	14.1	13.7	15.3

Table.4　acoustic radiation intensity by one-point circular PZT active control (5mm glass)

Frequency (Hz)	119	163	223	300	350	400	450	500
Before control (dB)	60.3	60.3	61.6	61.6	61.1	61.1	61.3	67.5
After Control(dB)	46.1	47.2	50.2	50.2	49.3	49.1	50.8	56.4
Difference (dB)	14.2	13.1	11.4	11.4	11.8	12.0	10.5	11.1

Table.5　acoustic radiation intensity by two-point circular PZT active control (8mm glass)

Frequency (Hz)	148	187	255	300	350	400	450	500
Before control (dB)	62.9	55.4	57.3	58.2	57.9	58.5	58.9	66.2
After control (dB)	42.5	39.2	42.1	42.1	45.0	46.0	45.6	56.1
Difference (dB)	20.4	16.2	15.2	16.1	12.9	12.5	13.3	10.1

Table.6　acoustic radiation intensity by one-point circular PZT active control (8mm glass)

Frequency (Hz)	148	187	255	300	350	400	450	500
Before control (dB)	61.9	55.5	57.9	53.2	57.6	59.6	59.7	66.4
After control (dB)	47.7	42.9	45.5	42.9	57.5	50.3	46.0	60.9
Difference (dB)	14.2	12.6	12.4	10.3	12.0	9.3	13.7	5.5

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