Active Vibration Control of a Simply Supported Plate with Piezoelectric Sensors and Actuators-II. Experiment

압전 센서와 액추에이터를 이용한 단순지지 평판의 능동 진동제어-II. 실험

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ABSTRACT

Undesirable vibration of a simply supported plate is controlled with piezoelectric sensors and actuators. Previous work of the author derived a theoretical equation to describe the vibration amplitude response of the plate, reflecting the combined effect of external driving forces and piezoelectric control moments. This study is aimed at verifying the validity of the solution through experiments for two types of external vibration exciters, i.e. a concentrated point exciter and a piezoelectric plate driver. Comparison between experimental and theoretical results shows a fairly good agreement with each other. Quantitative discrepancy between the two sets of data is discussed. The method investigated in this work is applicable to arbitrary external loading conditions and control algorithms.

요 약

압전 센서와 액추에이터를 이용한 단순지지 평판의 능동 진동제어에 관한 실험적 고찰을 하였다. 외력과 압전 제어력에 대한 평판의 진동 진폭반응을 고찰하여 구한 이론식의 타당성을 실험적으로 검증하였으며, 사용된 진동원으로는 집중 진동 여가기와 압전 구동기를 채택하였다. 실험치와 이론치 상호간에는 양호한 상관관계를 보였으며, 비교에서 나타나는 정략적인 오차의 원인을 분석하였다. 본 연구에서 고찰된 방법은 임의의 외력조건과 제어 알고리즘에 대해서 적용이 가능하다.

I. Introduction

Application of piezoelectric materials as sensors and actuators has become a growing source of interest in the active vibration control community. The piezoelectric materials have the advantages of light weight, good shapability, a wide frequency range, and a high electromechanical efficiency over conventional servo controller approaches. Much work has been done to apply this novel control element to one dimensional structures[1,2]. However, extension of the idea to higher order structures has been quite restricted [3]. Recently, Fuller[4] and Hubbard[5] conducted researches concerning two dimensional application of the piezoelectric material in active vibration control, even though their work is rather confined to a certain area of a whole control loop, i.e. transducer and actuator considerations. Hence there has been the need to connect the concept of piezoelectric sensors and actuators to dynamic properties of higher order structures

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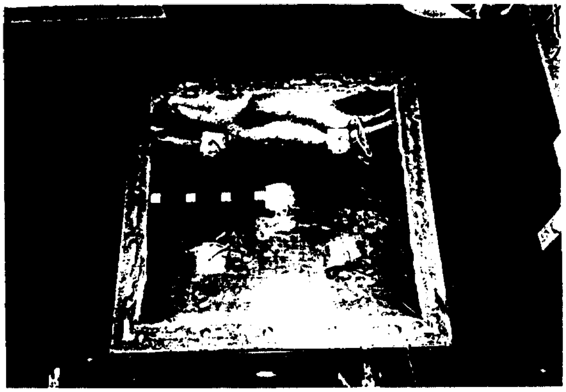
under active control scheme so that total vibration responses of the structures can be represented in a unified fashion. This sort of approach is tried to actively control sound radiation from a panel[6]. However, in the field of structural vibration control, no such attempt has been presented and this constitutes the main motivation to do a series of researches.

From this point of view, the previous work of the author[7] has achieved derivation of an approximate theoretical solution of the total vibration amplitude response of a simply supported plate to external forces. Numerical results showed the effectiveness of the approach. Detailed description of the theoretical equation will not be repeated in this paper. The present work is concerned with experimental demonstration of uses of the equation. With a simply supported plate setup, experiments are performed for two types of external force sources, a point vibration exciter and a piezoelectric driver. In this study, more favor is given to the piezoelectric plate driver. In addition to the fact that point force inputs can sometimes lead to control spillover, physical limitations are imposed due to the nature of the hardware necessary in implementing the control, i.e. shakers capable of providing the necessary driving force must be physically attached to the structure. Recent studies have recommended a piezoelectric driver as a promising driving input in order to overcome some of these disadvantages[4]. Induced structural vibration is controlled with piezoceramic sensor/actuator pairs under the scheme of direct feedback method as described in the Ref. 7. For the experimental results, no attempt is made to optimize the sensor/actuator pair shape or location to selectively excite modes. This is the target of researches ahead. The work presented here is an experimental extension of the theoretical investigation in controlling two dimensional vibration modes, that is verification of the validity of the theoretical derivation through comparison with experimental data.

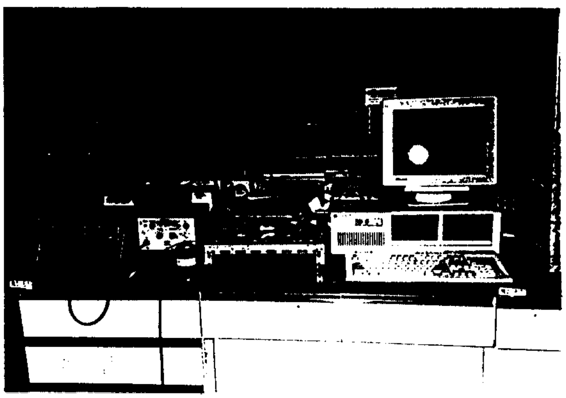
In Sec. 2, preparation of a simply supported plate is described. Ochs and Snowdon[8] achieved an almost perfect realization of the simply supported boundaries. But their process itself is a big job, and in this paper a simplified method is pursued to meet the boundary conditions. In Sec. 3, a modal analysis is performed to extract resonant frequencies of the plate. Once the natural frequency of each mode is determined, external forces are applied to selectively excite specific modes of our interest by means of the two types of exciters. Experimental data are compared with the results computed with the theoretical solution. Section 4 concerns the use of piezoelectric drivers, and Sec. 5 does that of a point vibration exciter. Discussion of the comparison is conducted in the final section.

II. Preparation of a Simply Supported Plate

The test plate, which is mounted in a rigid steel frame, is made of aluminium and measured 360 mm × 360 mm × 0.5 mm. Figure 1 illustrates the plate with piezoelectric sensor/actuator pairs on and experimental arrangement. Five piezoceramic, PZT, sensors and actuators are placed on the surface of the plate. The pair of one sensor and one actuator makes one control element. Each actuator consists of two PZT elements of dimensions 30 mm × 30 mm × 0.3 mm bonded symmetrically, front and back surfaces. The sensors are put on the front side only, and have dimensions of 15 mm × 5 mm × 0.3 mm. The sensors and actuators are bonded to the plate surface with Aremco Conducting Bond cured at 120℃. Material properties of the plate and PZT are the same as those in Ref. 7. Simply supported boundary conditions are accomplished by inserting edges of the plate to slots made in the steel frame and sealing the boundaries with a silicon rubber. Preparation of the edges is described in Fig.2-a. The slot in the rigid frame restricts the plate out of plane motion at the boundaries. If the insertion is too deep, the frame would prevent whole edge motion of the plate and the boundary conditions would change to those of

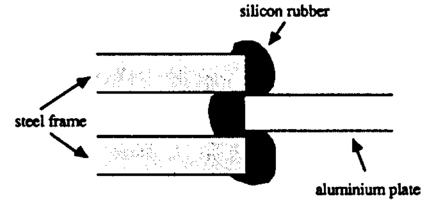


(a)simply supported plate used in the experiment

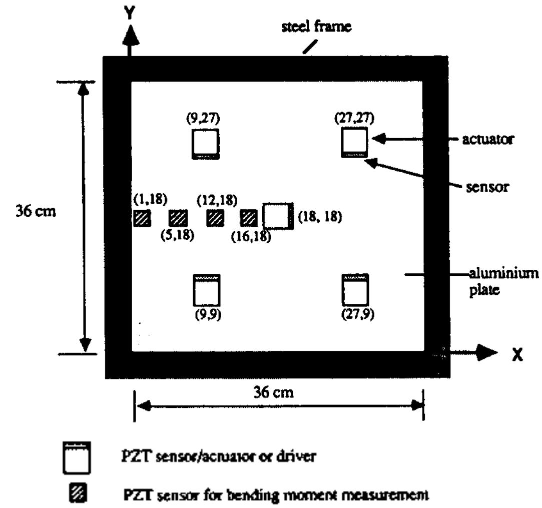


(b)experimental setup

Fig. 1. View of the experimental set up



(a)schematic view of the edge of the plate



(b)schematic upper view of the simply supported plate with piezoelectric sensors and actuators on. Numbers in the figure are (X, Y) coordinates denoting positions each element's center.

Fig. 2. Preparation of the simply supported plate

clamped edges. Thus, the insertion depth is kept as small as possible. Actually the edges are not inserted into the slot, but rather are tried to be in line with the boundaries of the frame, allowing the plate to bend relatively freely. The gap between the edges of the plate and the steel frame is sealed with flexible silicon rubber to supplement the holding effect. Therefore the condition of a negligent vertical displacement alogng the edges is to be achieved by the rigid steel frame, and the condition of a negligent bending moment along the edges is to be achieved by the silicon rubber. This sort of plate edge setup is different from that of Ochs and Snowdon[8]. To make sure that the boundary conditions are met correctly, a preliminary experiment is conducted and its result is compared with that of theoretical calculation. For a simply supported plate, vertical displacements and bending moments should be zero at the edges and should be the maximum at the center when the plate is loaded. Input load is applied to the plate by means of a point vibration exciter positioned at the center of the plate. Induced vertical displacement is measured with four mini-accelerometers (B&K 4375), and bending moment with four PZT vibration sensors. The PZT sensor responds to the bending moment and operates in the same principle as the sensor in the piezoelectric control element in Ref. 7. Each PZT sensor measures 10 mm $\times$ 10 mm $\times$ 0.3 mm. By symmetry, measurement along one center line in one half space is enough to check the satisfaction of the boundary conditions. Fig. 2-b illustrates the location of the sensors, where

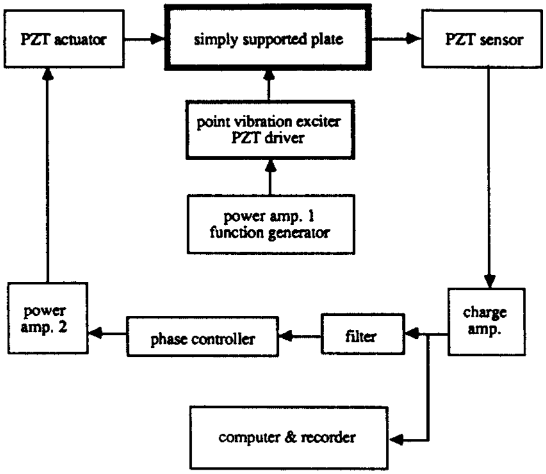
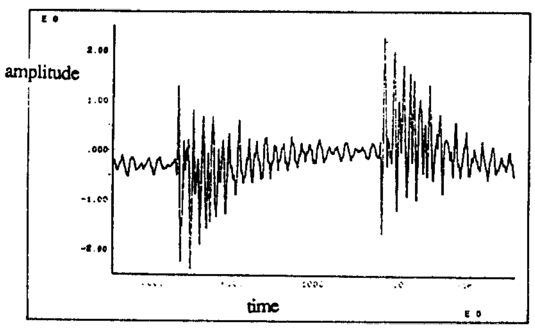
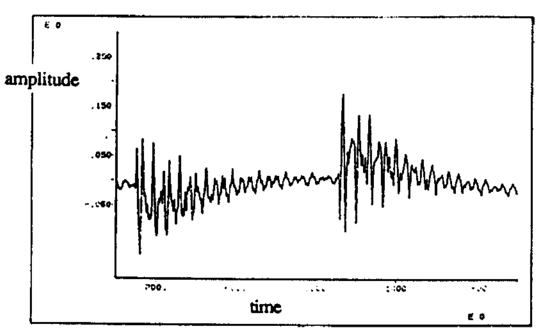


Fig. 3. Schematic diagram of measurement equipment setup



(a)measurement with the sensor at (16, 18)(near the center)



(b)measurement with the sensor at (2, 18)(near the edge)

Fig. 4. Measurement results of bending moment with pirzoelectric sensors

accelerometers replace the PZT sensors for measurement of a vertical displacement. Figure 3 describes measurement equipment setup. The control equipment consists of B&K precision conditioning amplifier 2650, Krohn-Hite dual filter 3202, B&K level/phase controller 5899, and B&K power amplifier 2713. The driving-side equipment consists of Philips function generator 5134, B&K vibration exciter 4810, and B&K power amplifier 2706. Figure 4 shows representative measurement results of the bending moment at two end points, i.e. near the edge and near the center. When the maximum amplitude of the two signals are compared with each other, bending moment ratio near the edge to that near the center is 0.06. If the simply supported boundary conditions are perfectly satisfied, corresponding theoretical value is 0.02. The difference between the two values is thought to be mainly due to the silicon rubber. Holding the bending moment to zero is restricted by the flexibility of the silicon rubber. The silicon compound is highly pliant, but it is not perfectly pliant and it causes some moment to occur along the boundaries. When the holding effect is focused on, the rubber should be inflexible, while the plate's free bending is focused on, it should be perfectly flexible. There must be some trade-off between the two targets. With the current silicon compound, more emphasis is given to the allowance of free bending because the role of edge holding is supposed to be taken by the steel frame. Similar data are obtained with the accelerometers. According to the results, holding the vertical displacement to zero is achieved fairly well. Measured values are summarized in Tables 1 and 2 with corresponding theoretical values. Quantitative comparison shows some difference even though not very much, but general trend agrees well with each other. Therefore, in conclusion, the current setup is considered to be suitable for representing a simply supported plate.

Table 1. Bending moment ration at various points to that near the center of a simply supported plate.

semsor position (X, Y coord.)	measured with PZT sensors	theoretical calculation
(1, 18)	1	1
(5, 18)	0.71	0.62
(12, 18)	0.29	0.20
(16, 18)	0.06	0.02

Table 2. Vertical displacement ration at various points to that near the center of a simply supported plate

semsor position (X, Y coord.)	measured with accelerometers	theoretical calculation
(1, 18)	1	1
(5, 18)	0.82	0.79
(12, 18)	0.47	0.44
(16, 18)	0.05	0.07

III. Modal Analysis

Before testing the validity of the theoretical derivation, a modal analysis is performed. A mode is not observable if the sensor is placed at the nodes of the mode in question. For one dimensional structure, the nodal points for the lower modes, which are the most likely to be in need of control, are well spaced, so that the estimation of the associated modal states does not present any special problem. On the other hand, for two and three dimensional structures, in which the modes are characterized by nodal lines and nodal surfaces, respectively, the problem of mode observability can be critical. For the same reason, care is paid for the location of the point load. As with the sensor, if the load is placed at the nodal points of a certain mode, the specific mode can not be generated. Hence both of the external load and vibration sensors should avoid the nodes of interesting modes. The previous theoretical work dealt with the lowest four different modes of a rectangular plate, (1, 1), (1, 2), (2, 2), and (1, 3). Thus the load position is determined as (7, 7) and the sensor as (27, 27) according to the coordinate in Fig. 2-b. Initial random signal is given

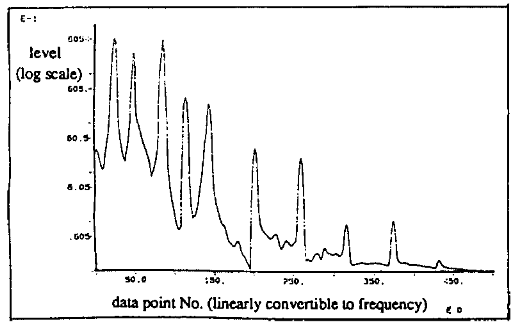


Fig. 5. Experimental modal analysis of the simply supported plate

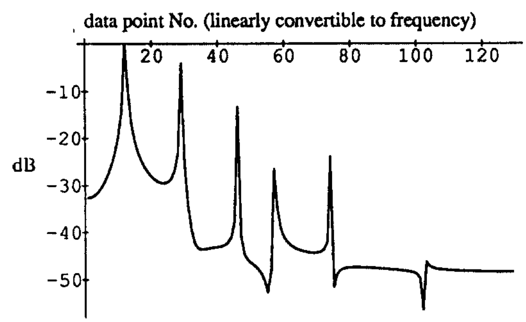


Fig. 6. Theoretical modal analysis of the simply supported plate

to the plate by a point exciter and induced vibration is detected by a piezoelectric sensor. The piezoelectric sensor operates in the same manner as that in Sec. 2, and has the dimension of 30 mm $\times$ 30 mm $\times$ 0.3 mm. Frequency of the source signal is set well below the (1, 1) resonant frequency, and is 5 Hz. Result is shown in Fig. 5. The figure clearly shows the resonant frequencies of each mode. For the same situation, theoretical calculation is conducted by including time harmonic terms in each model component in the derived equation and performing FFT. The theoretical result at the current exciter and sensor locations is shown in Fig. 6. Numerical values of the experimental and theoretical resonant frequencies of the lowest four modes are denoted in Table 3. Due to the assumption in the derivation like perfect linearization of modal shapes and experimental error factors like the imperfect realization of the boundary conditions, the two figures are not exactly the same. But the modal peaks generally well correspond to each other [9]. Therefore it can be said that the derivation can describe the modal distribution appropriately.

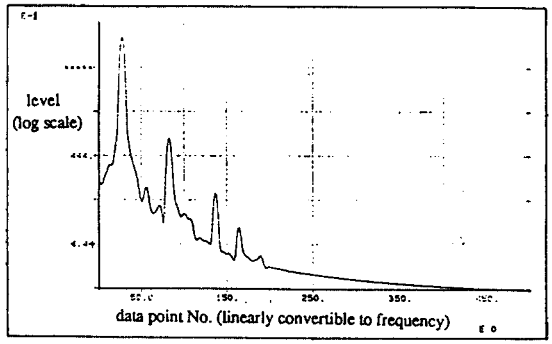
Table 3. Comparison of experimental and theoretical resonant frequencies of the lowest four modes of the simply supported plate

mode	experimental(Hz)	theoretical(Hz)
(1, 1)	24.4	22.5
(1, 2)	51.3	56.4
(2, 2)	87.8	90.1
(1, 3)	117.1	112.7

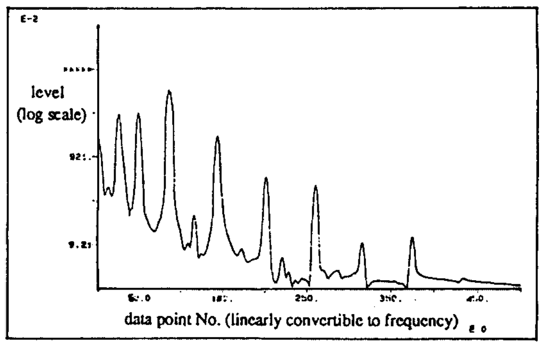
Ⅳ. Piezoelectric Plate Driver

In the previous work, piezoelectric plate drivers were employed as the source of external load to the plate. This situation is investigated through experiments. As shown in the Fig. 1, four piezoelectric sensor/actuator pairs are placed at the four rectangular points on the surface of the plate. One more sensor/actuator pair is positioned at the center of the plate. The PZT actuator element constituting the controller (sensor/actuator pair) is used as the driver. Instead of being connected to a feedback loop, the element is connected to a driving circuit, i.e, a signal generator and a power amplifier, and its role is changed. The driver at the center is used for generating odd-odd modes and the other four are for odd-even and even-even modes.

First, by the PZT plate driver at the center, (1, 1) mode is generated by tuning the external source signal to 24 Hz. Figure 7-a shows the experimental result. As expected, the (1, 1) mode is dominant. Other higher modes of vibration are generated, too. But their level is significantly lower than that of the first mode. Hence selective excitation of a particular mode with the PZT driver is considered successfully achieved. The same situation is simulated by means of the theoretical equation and its result is described in Fig. 8-a. Theoretical derivation did not include damping properties of the materials constituting



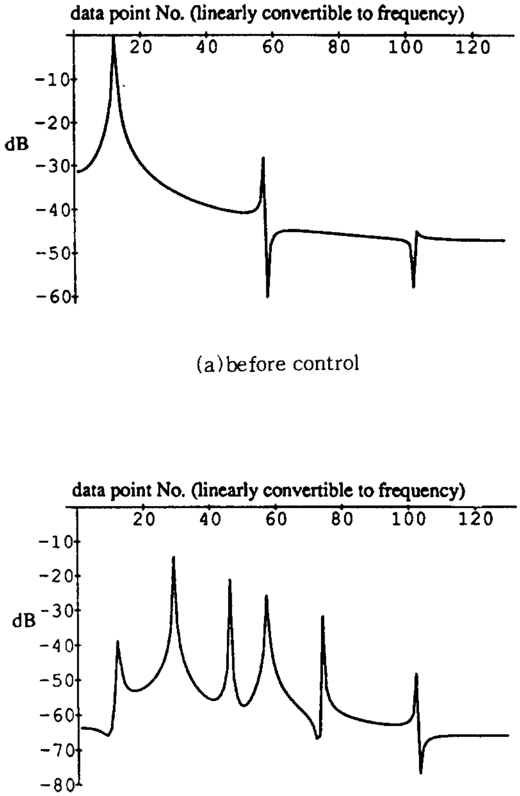
(a)before control



(b)after control

Fig. 7. Experimental vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 1) resonance mode

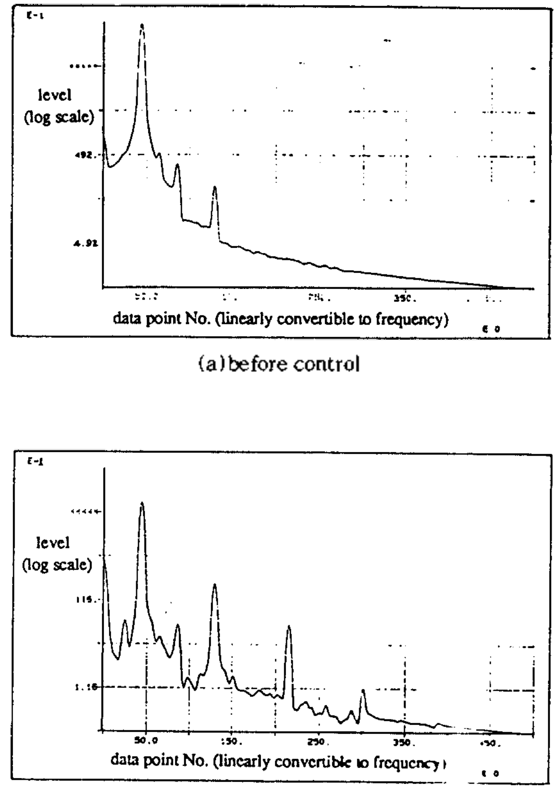
the plate. Thus if the experimental source frequency is set exactly equal to the theoretical resonant frequency, the peak of a tuned mode would go to infinity. Due to the difference in the experimental and theoretical natural frequencies as shown in the Table 3, the computed result illustrates a rather practical situation. Figure 8-a is quite similar to the experimental data. In the calculation, m and n are included up to 4. Now, the PZT driver induced vibration is controlled by a PZT sensor/actuator pair centered at (9, 9). In the control, direct feedback method is employed in favor of its simplicity in implementation. Control signal frequency applied to the actuator is filtered to be equal to the initial loading frequency. Experimental result is shown in Fig. 7-b. The position of the control element is not to be



(a) before control

(b) after control

Fig. 8. Theoretical vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 1) resonance mode



(a) before control

(b) after control

Fig. 9. Experimental vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 2) resonance mode

optimal for suppressing the (1, 1) mode of vibration. It is placed arbitrarily just to test the suitability of the analytic derivation. Of course, even at the current location, the PZT actuator can control the odd-odd modes of vibration, but while controlling, the actuator can impart energy to originally weak modes, that is odd-even and even-even modes. Figure 7-b confirms this expectation. While the (1, 1) mode decreases by 45 dB, the (1, 2) and (2, 2) modes increase their level by 24 dB and 12 dB, respectively. This situation is analyzed by the analytic solution and its result is shown in Fig. 8-b. The vertical dB scale is referenced to the maximum value in the Fig. 8-a. The result simulates the experimental data fairly well. (1, 1) mode decreases by 38 dB while (1, 2) mode and (2, 2) mode increase by 21 dB and 19 dB, respectively. There is some discrepancy in the quantitative increase and decrease of each mode's amplitude, but the general trend agrees with each other.

Secondly, the frequency of the external loading signal is tuned at the (1, 2) mode of vibration. The PZT driver at the center can not excite this mode because it is positioned at the nodal point of the mode. Hence, the PZT driver at (27, 27) is utilized to generate this mode. Control element, the PZT sensor /actuator pair, is still that at (9, 9). Figure 9 is the experimental result and Fig. 10 is the corresponding theoretical result. In general, they agree with each other. In a quantitative measure, (1, 2) mode decreases by 28 dB experimentally, and by 34 dB theoretically. In the calculation, nearby (2, 2) mode increases by 8

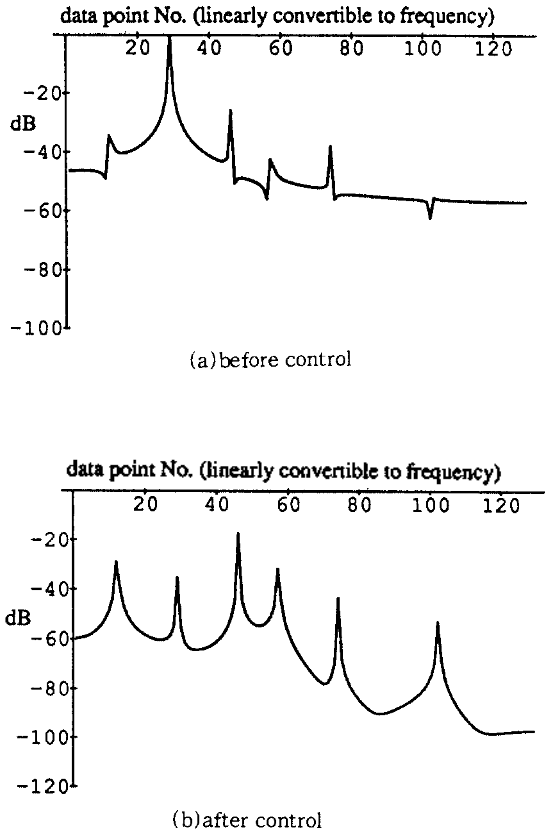


Fig. 10. Theoretical vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 2) resonance mode

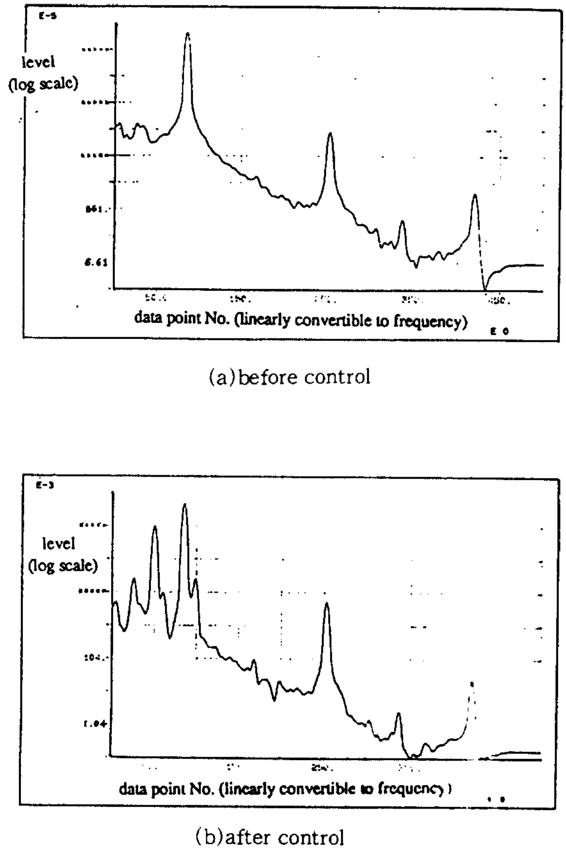


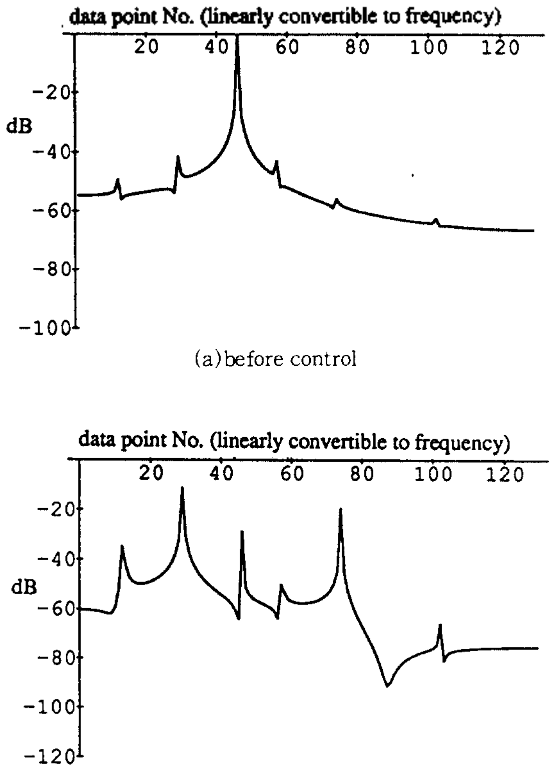
Fig. 11. Experimental vibration amplitude response of the plate with the PZT driver loading frequency tuned at (2, 2) resonance mode

dB, while the experimental data does not show that much increase.

Thirdly, with the same couple of PZT driver at (27, 27) and sensor /actuator pair at (9, 9), the source frequency is adjusted at the (2, 2) mode. Similar comparison can be made between experimental and theoretical results before and after control. Figures 11 and 12 are the results. Experimentally, (2, 2) mode decreases by 16 dB and (1, 2) mode increases by 51 dB. Theoretically, (2, 2) mode decreases by 28 dB and (1, 2) mode increases by 32 dB.

Finally, for the (1, 3) mode, the same procedure is taken and the results are displayed in Figs. 13 and 14. For excitation of this (1, 3) mode, the PZT driver is switched to that at the center. The driver at (27, 27) can generate (1, 3) mode, too. However, the driver at the center is exactly at the antinodal point of (1, 3) mode, and it is more efficient in the excitation of the mode. Experimentally, (1, 3) mode decreases by 28 dB and (1, 2) mode increases by 38 dB. Theoretically, they are 28 dB and 16 dB, respectively.

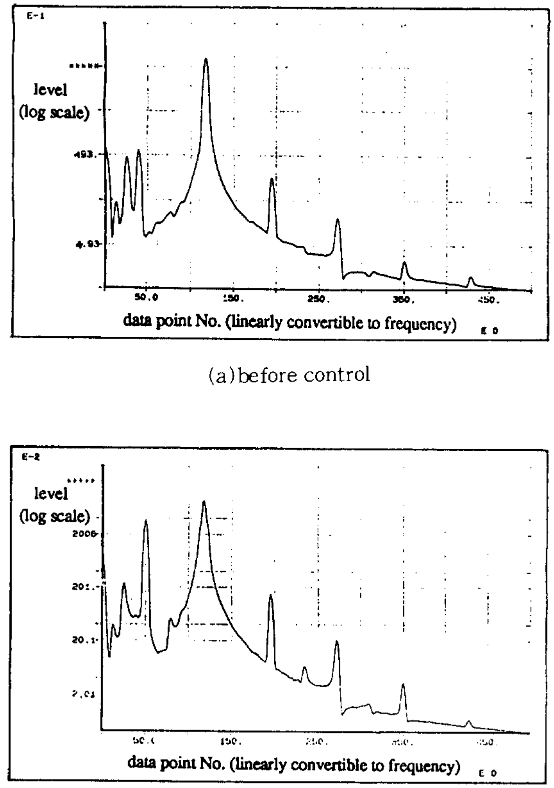
So far, active vibration control of a simply supported plate has been investigated experimentally and the data have been compared with corresponding theoretical values. Main purpose of this study is to check whether the approximate analytic solution derived in the author's previous work can describe two dimensional active vibration control phenomena appropriately or not. According to the comparison between the two sets of results, experimental and theoretical, the analytic equation can simulate the experiments



(a) before control

(b) after control

Fig. 12. Theoretical vibration amplitude response of the plate with the PZT driver loading frequency tuned at (2, 2) resonance mode



(a) before control

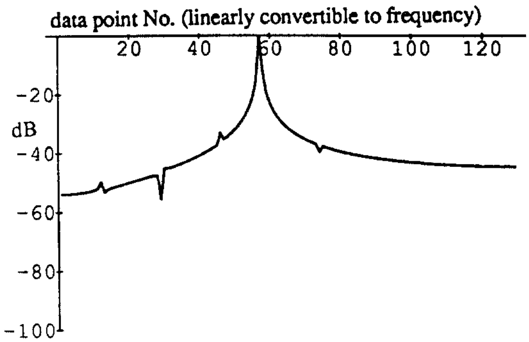
(b) after control

Fig. 13. Experimental vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 3) resonance mode

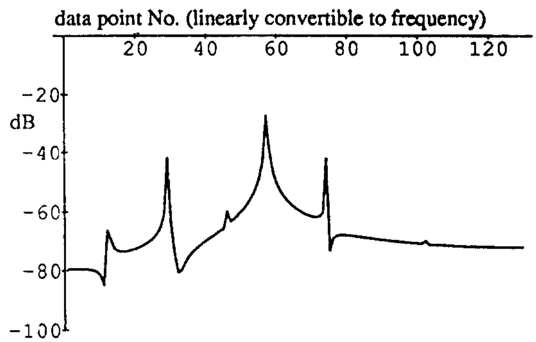
moderately well. In quantitative measure, there is some difference, which is mainly attributed to the factors explained in Sec. 2. Rigorous preparation of the simply supported boundaries of the plate will improve the closeness of the two sets of data. Besides, the calculation ignored dielectrical and mechanical damping properties of the materials constituting the plate, which is considered to be in charge of the discrepancy in increases or decreases of each mode's power level. This ignorance causes the theoretical power applied to the plate to differ from the experimental power. The equation derived in Ref. 7 is very general. Use of complex material constants in the calculation would enable the representation of lossiness of the plate[10], and would enhance the closeness of the two sets of data. In conclusion, for the case of piezoelectric drivers, the analytic solution can be utilized as a good simulation tool to describe the active vibration control mechanism of a simply supported plate.

V. Point Vibration Exciter

Similar experiment is performed with a point vibration exciter. Experiment setup is the same as Fig. 2, but the PZT plate driver is replaced with a vibration exciter. The exciting load is positioned at (27, 27). Induced vibration is controlled with the sensor/actuator pair at (9, 9). Center frequency of the external loading signal is set to the (2, 2) vibration mode of the plate and corresponding experimental result is shown in Fig. 15-a. As expected, (2, 2) mode has the highest peak and (1, 1) and higher modes show up, simultaneously. Theoretical result, Fig. 16-a,

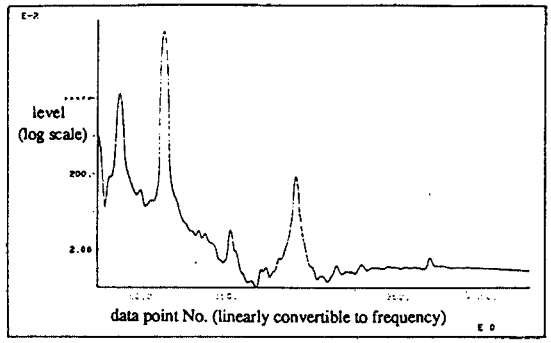


(a)before control

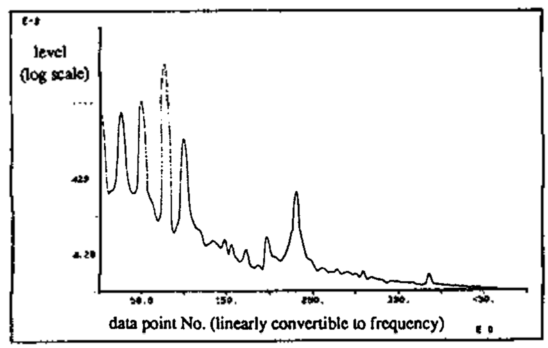


(b)after control

Fig. 14. Theoretical vibration amplitude response of the plate with the PZT driver loading frequency tuned at (1, 3) resonance mode



(a)before control



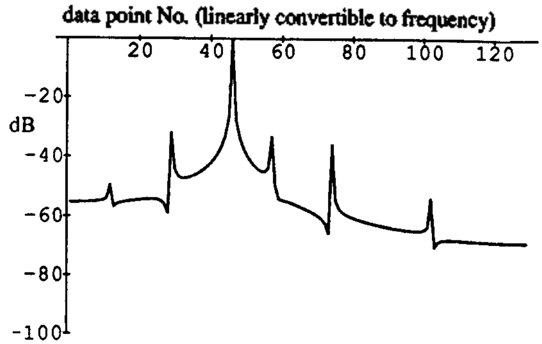
(b)after control

Fig. 15. Experimental vibration amplitude response of the plate with the point exciter loading frequency tuned at (2, 2) resonance mode

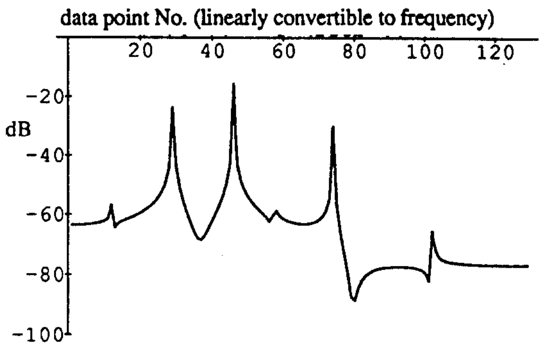
confirms this phenomenon. The PZT plate driver has finite planar dimensions and therefore has areal effect on the excitation of vibration. Since all the edges of the driver act in phase, control moment at each edge adds up or cancels out each other as far as the excitation of a particular mode is concerned. But the point load is applied to a single spot on the plate, i.e. no areal effect. In this sense, point load should be capable of exciting vibration modes more slectively. However, physically attaching the exciter to the plate and tightly holding the exciter and plate to prevent any harmful motion between themselves is a very difficult job. Hence, compared with Fig. 11-a, the point exciter is poorer in selective generation of a certain mode. Induced vibration is controlled in the same manner as before. Figure 15-b shows the experimental result and Fig. 16-b shows the corresponding theoretical result. They are consistent with each other reasonably well. Experimental decrease of (2, 2) mode is 14 dB and increase of (1, 2) mode is 32 dB. Theoretically, they are 17 dB and 19 dB, respectively. Therefore for the external point load, too, the theoretical solution derived in the author's previous work can describe the active vibration control mechanism appropriately.

Ⅵ. Conclusion

The purpose of this study is to experimentally verify the validity of the theoretical equation derived in the previous work describing the active vibration control of a simply supported plate with



(a) before control



(b) after control

Fig. 16. Theoretical vibration amplitude response of the plate with the point exciter loading frequency tuned at (2, 2) resonance mode

piezoelectric sensors and actuators. The analytic solution is general and it can be employed for arbitrary types of external load. In this work, measurements were performed for two types of vibration sources, a point load exciter and PZT plate drivers. Theoretical results agreed with the experimental data reasonably well. In quantitative measure, there is some difference, which is attributed to imperfect experimental realization of simply supported boundaries and theoretical idealization of the plate. But general trend of the two sets of data show fairly good agreement with each other. In conclusion, the analytic solution can be utilized as a good simulation tool to describe the active vibration control mechanism of a simply supported plate. The method presented in this paper is applicable to arbitrary control algorithm. Future work will be concerned with optimization of the sensors and actuators as well as employment of various control schemes.

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