Reduction of plate vibration and acoustic radiation using adaptively controlled boundaries

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Reduction of plate vibration and acoustic radiation at low frequencies using adaptively controlled boundaries is experimentally demonstrated. The control is based on changing the clamping pressure at the boundaries of the plate to continuously detune the plate resonance frequencies from the frequency of pure-tone excitation. Results are presented for the control of plate vibration with constant and varying frequencies of excitation and for the control of acoustic radiation with constant frequency excitation.

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INTRODUCTION

Recent approaches to controlling structural vibration and radiation at low frequencies, where conventional passive controls are ineffective, are based on active control schemes which use secondary sources to counteract the unwanted vibration. These methods of control are often implemented adaptively, such that they can automatically adapt to changes in the controlled system. The research presented in this paper approaches the control of structural noise and vibration adaptively, but without using secondary vibration sources. The control used continuously adapts to changes in the system so as to change the passive properties of the system to minimize vibration and/or acoustic radiation.

As shown by Joga Rao and Kantham,[1] Laura and Romanelli,[2] Laura *et al.*,[3] Laura *et al.*,[4] Filipich *et al.*,[5] Laura and Grossi,[6] Laura and Grossi,[7] Warburton and Edney,[8] and Hammouda,[9] the resonance frequencies of the lower-order modes of a plate are sensitive to boundary conditions. This sensitivity is exploited in a laboratory demonstration in which the clamping pressures at the plate boundaries are adaptively controlled to continuously detune the plate resonance frequencies from the frequency of excitation and thereby reduce the plate vibration and acoustic radiation.

The effects of the passive control of structural properties on the acoustic response of structures were considered by Liang *et al.*[10] and Saunders *et al.*[11] Both considered the effects of shape-memory alloys on changing frequencies and mode shapes of resonance of composite structures. Liang *et al.* used embedded shape-memory alloy fibers to control acoustic radiation and transmission of a composite panel. Shape-memory alloy fibers embedded in a composite beam were used by Saunders *et al.* to detune the resonance frequencies from the excitation frequency to reduce acoustic radiation. However, it is not always possible or cost effective to embed fibers in a structure to achieve acoustic control. Therefore, to remove the need to embed fibers in the structure, the effects of controls local to the boundaries of the structure are considered in this paper.

The experiment used to demonstrate the adaptive control is described in Sec. I. Section II outlines the control strategy, while results of both vibration and acoustic radiation control of the plate are given in Sec. III. Conclusions and recommendations for further research on controlling plate vibration and acoustic radiation are given in the last section.

I. DESCRIPTION OF THE EXPERIMENTS

A rectangular aluminum plate was mounted in the top of a rectangular steel box which contained hydraulic pistons at the four corners. The purpose of the pistons was to vary the clamping pressure at the edges of the plate. A schematic diagram of the system is given in Fig. 1. The dimensions of the plate were $0.013\times0.61\times0.91$ m ($\frac{1}{2}$ in.$\times$2 ft$\times$3 ft). The plate was mounted between two steel frames. The upper frame was bolted to the top of the box, and the lower frame rested on the four hydraulic pistons located under the four corners of the frame. The edges of the plate were fitted with aluminum strips milled to a sharp edge, which rested in grooves cut in the frames as shown in Fig. 2. These strips were used to reduce the damping of the plate by diminishing the amount of contract between the plate and the frames, thereby reducing the energy lost at the edges of the plate. The use of these strips resulted in the measured damping loss factors shown in Fig. 3.

The plate was excited by a Wilcoxon Research F4 shaker mounted directly to the plate. The shaker was also fitted with a Wilcoxon Research Z820 impedance head to provide input information for the control system. The hydraulic pistons, which provided pressures at the plate boundaries, were located inside of the box underneath the lower frame. The hydraulic pistons forced the lower frame up against the upper frame, which allowed for variable clamping pressures over the boundaries of the plate. This clamping pressure changed the boundary conditions to detune the frequencies of resonances of the plate from the excitation frequency.

A Vicker EHST-3 hydraulic pressure control value provided the interface between the electronic system and the hydraulics. This control value regulated the pump pressure from 90 to 900 psi in proportion to the input voltage from the electronic control system. A totally released state, which

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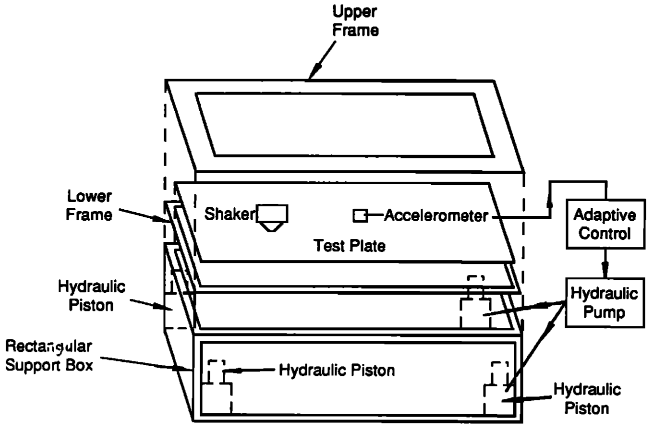

FIG. 1. Apparatus used in experiments to vary clamping pressures on boundaries of plate.

simulated free boundaries, was not directly obtainable using the electronic pressure control, so a 300-lb weight was attached to the lower frame to release the pressure at the edges of the plate when the hydraulic pressure was lowest (90 psi). Even with this weight, a pump pressure of 900 psi resulted in nearly clamped boundary conditions. For simplicity, the 90-psi condition will be referred to as "free" and the 900-psi condition will be referred to as "clamped," although neither is ideal.

The adaptive control was implemented with a Motorola DSP56000ADS digital signal processing board. The DSP56000 is a 56-bit general purpose digital signal processor which was interfaced with a personal computer. The assembly language code was written on the computer and then downloaded to the board. The adaptive control algorithm will be discussed in Sec. II.

The input analog signal from the control accelerometer or microphone was converted to a digital signal that the DSP board could use, and the digital output signal from the board was converted to an analog signal sent to the hydraulic pump. This was accomplished using an Ariel ADC56000 I/O board, which was designed to interface directly with the Motorola DSP56000. The ADC56000 allowed two channels of analog-to-digital (A/D) and digital-to-analog (D/A) conversions.

II. CONTROL STRATEGY

As the elastic constant of the translational edge restraint was varied from zero to infinity, Hammouda[9] showed that the boundary conditions changed from free to simply supported. Figure 4 illustrates the dependence of the frequencies of resonance on the translational restraint. The change in resonance frequency produced by changes in the boundaries has three distinct regions. In the first region, the resonance frequency corresponds to that for the free plate. The second region is characterized by a transition from the free-resonance frequency to the simply supported resonance frequency. Finally, the third regions shows that above a finite spring constant the plate responds as if it were simply supported. This shift in resonance frequency with changing boundaries was found to be most pronounced for the lower-order modes of the plate. Since the higher-order modes are

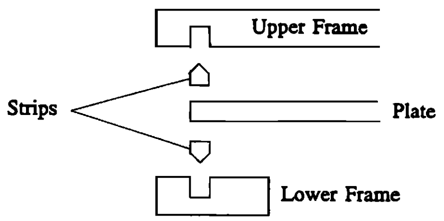

FIG. 2. Configuration at boundaries of plate.

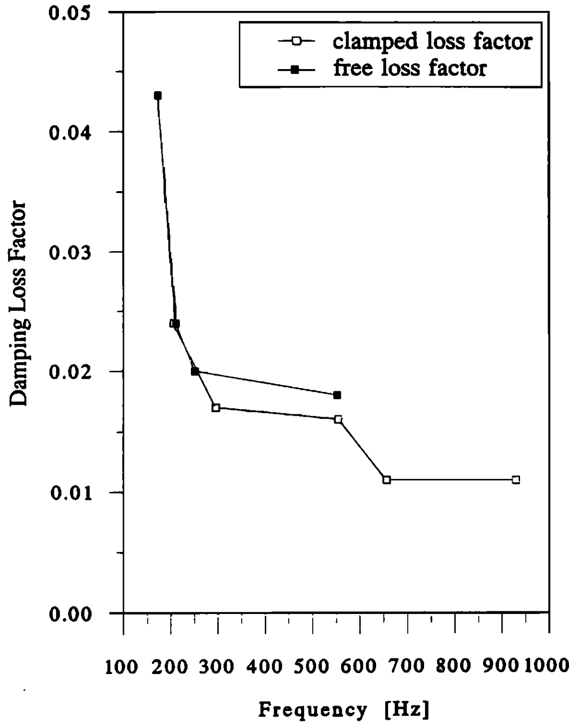

FIG. 3. Measured loss factors for the plate mounted in the experimental apparatus.

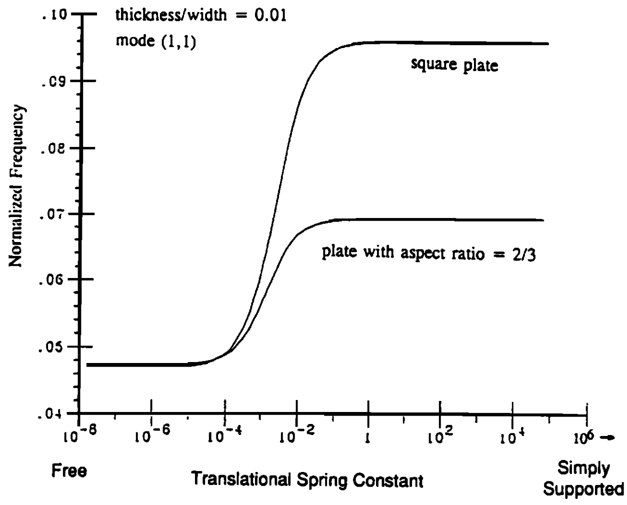

FIG. 4. Variation in frequencies of resonance for the lowest mode of a plate with increases in the translation restraint at two plate edges (from Ref. 9).

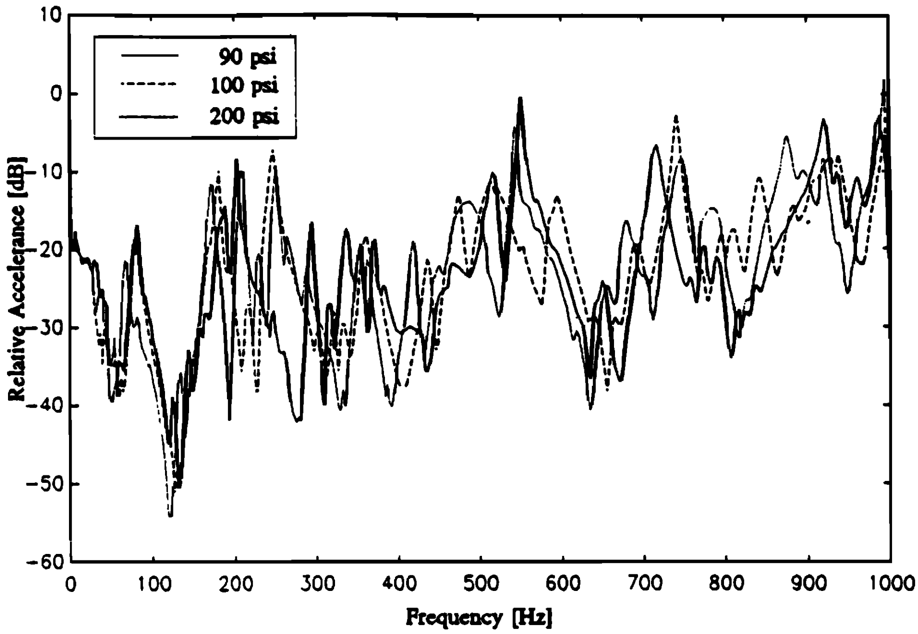

FIG. 5. Accelerance of the plate at lower clamping pressures.

not as sensitive to the boundary conditions, the boundaries have less influence on the higher frequencies of resonance.

To verify that shifts in the resonance frequencies occur in the experimental setup, the response of the plate was measured at various clamping pressures. To determine the uncontrolled response of the system, a swept sine drive force from 0 to 1000 Hz was applied by the shaker to the plate. The resulting acceleration of the plate was measured using the impedance head in the shaker. Figures 5 and 6 show the accelerance (acceleration normalized by force) of the plate for low and high pressures at the plate boundaries. These plots show the shift of the resonance frequencies as the boundary clamping pressure changes. It should be noted that there is very little difference among the three curves in Fig. 6. This indicates that the shift in the resonance frequencies was mostly complete when the clamping pressure reached 200 psi. Unlike the ideal conditions assumed in theoretical models, in-plane stresses may be induced by the edge pressures in the experiments,[12] which could have some impact on the results presented in Figs. 5 and 6.

The algorithm used to control plate vibration and acoustic radiation was based on a simple direct search strategy, similar to that used by Saunders *et al.*[11] The result of control at each time step was compared with previous results to determine the next trial solution. This is a simple approach to

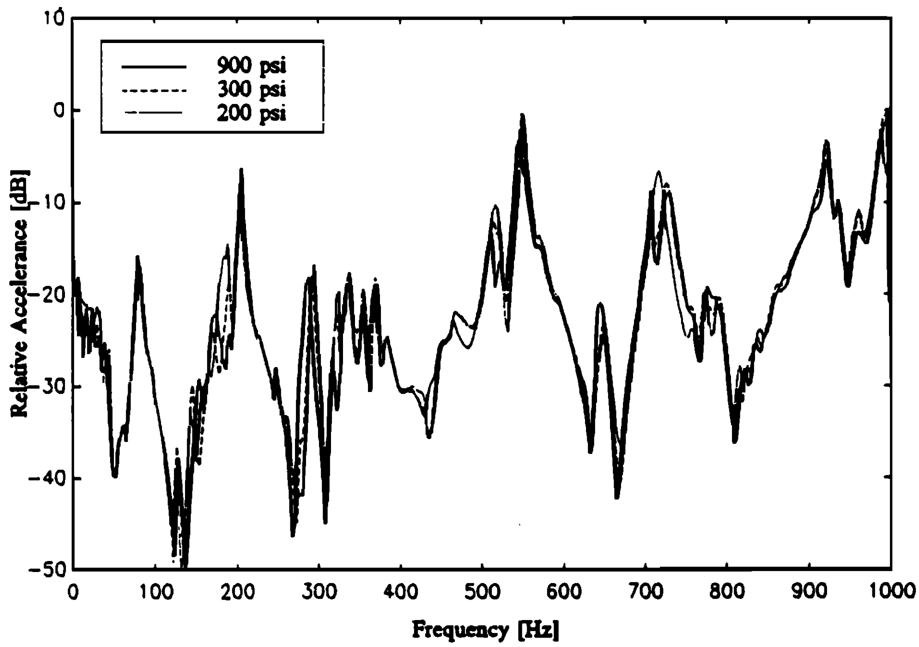

FIG. 6. Accelerance of the plate at higher clamping pressures.

adaptively controlling the plate, and is well suited for use with modern signal processing hardware.

For controlling the boundary conditions of the plate, the input data, which was either acceleration in the case of vibration control or acoustic pressure in the case of acoustic radiation control, was sampled at a rate of 1 kHz. The control output was sent directly to the hydraulic pressure control which produced a hydraulic control pressure proportional to the control signal voltage.

To determine if the control signal should be incremented or decremented, the algorithm averaged the input data over time and then compared it to the previous average. If the new average was less than the old average, the boundary clamping pressure had produced a decrease in the vibration level of the plate, which meant that control signal was changed in the "right" direction. If however, the new average was greater than the old average level, the change in clamping was in the "wrong" direction. The controller then changed the control signal in the opposite direction. By continuously taking averages and making corrections to the control signal, the algorithm was able to find the minimum vibration response or acoustic radiation of the plate at the error sensor. This process was continued so that a minimum response was maintained in the presence of changes in the drive frequency.

Three versions of this algorithm were implemented. The first version was implemented with a constant increment or step size. This version converged slowly, and once converged would oscillate about the final converged value. This behavior resulted since every change in the vibration response produced a change in the control signal. The second version of the algorithm took this into account by only incrementing or decrementing the control signal if the difference between averages was larger than a preset value determined by trial and error. This version of the algorithm worked better, but still converged slowly in some cases. The third and final version of the algorithm determined the step size by scaling it to the difference between consecutive averages. The absolute value of the difference between successive averages was multiplied by 14 (determined by trial and error) to determine the magnitude of the step size. This allowed for faster convergence by finding the minimum level more quickly, and was more stable.

III. EXPERIMENTAL RESULTS

A. Fixed frequency vibration control

To test the control system, several resonance frequencies for the lowest edge clamping pressure (90 psi) and for the highest edge clamping pressure (900 psi) were chosen. The plate was continuously excited at each of the selected resonance frequencies individually and the controller turned on. The level of the response was recorded before the controller was started and after the controller had converged, along with the final hydraulic control pressure. The reduction is reported in terms of accelerance, where normalizing the acceleration to the force removes the effects of the impedance of the source. Results of vibration control for some of the frequencies for plate resonances with the uncontrolled 90-psi edge pressure are given in Fig. 7. The reduction obtained

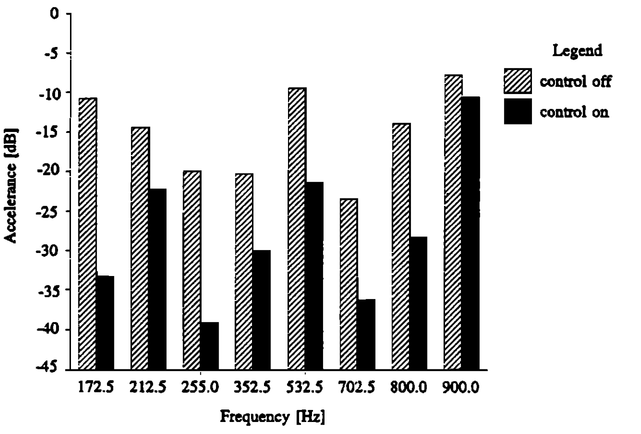

FIG. 7. Reductions in vibration achieved at the 90-psi plate resonances.

when the control was applied varied from 3 to 23 dB, with an average value of 13 dB. Similar results for some of the 900-psi resonance frequencies are shown in Fig. 8. Here, the reductions obtained varied from 0 to 15 dB, with an average value of 8 dB. These reductions occurred as the resonance frequency of the panel moved from the constant frequency of excitation, so that, after the controller converged, the excitation is at a nonresonance frequency. Ideally, the reduction should be close to the differences in a resonant peak and the closest antiresonance dip in the response curve, which will be lower for higher damping. The reductions shown in Figs. 7 and 8 were achieved in the presence of the relatively high damping shown in Fig. 3. With lower damping, greater reductions would be expected to occur.

The controller performed better for the free resonances at 90 psi than the clamped resonances at 900 psi. The final pressures produced with the controller on, along with the actual reduction obtained for each frequency, are listed in Tables I and II. The frequencies where less reduction occurred tended to be the frequencies where the resonance frequency spacing was small. The pressure needed to control the free-resonance frequencies was, in general, not much more than the original 90 psi. In other words, only small changes in the clamping pressures were required to control the free resonances since most of the plate's frequency shifts

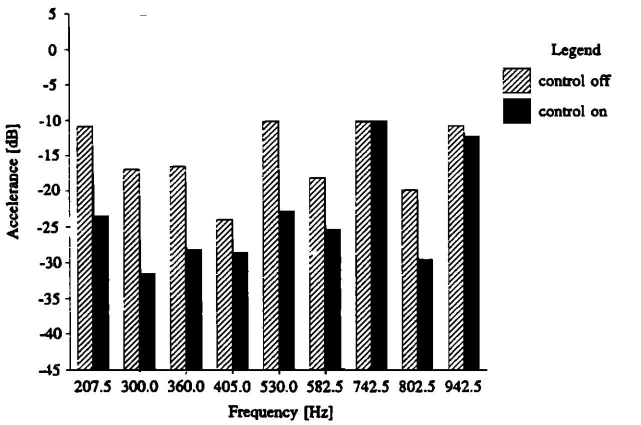

FIG. 8. Reductions in vibration achieved at the 900-psi plate resonances.

TABLE I. Vibration control results for 90-psi resonances.

Resonance frequency [Hz]	Final control pressure [psi]	Actual reduction [dB]
172.5	275	23
212.5	150	8
255	150	19
352.5	100	10
532.5	100	12
702.5	100	13
800	150	14
900	625	3

TABLE II. Vibration control results for 900-psi resonances.

Resonance frequency [dB]	Final control pressure [psi]	Actual reduction [dB]
207.5	100	13
300	150	15
360	150	12
405	100	5
530	90	13
583.5	200	7
742.5	900	0
802.5	90	10
942.5	300	2

were completed at pressures below 200 psi. For the clamped resonances, however, the controller had a more difficult time reaching the lower edge pressures needed to "unclamp" the plate and achieve the expected reduction. This was due to the inability of the hydraulic system to consistently release fully when driven electronically to achieve a free condition. This also lengthened the time required for the controller to converge for the clamped resonances as opposed to the free resonances, where the convergence was faster.

B. Variable frequency vibration control

In a number of applications, the structure to be controlled may be excited by a variable frequency source. To achieve control with a varying frequency excitation, the controller was run continuously while the excitation swept once from low to high frequency through a preset band about a resonance peak. Typical results for sweeping with and without control can be seen in Figs. 9 and 10 for free- and clamped plate resonances, respectively. Figure 9 shows control results in the 20-Hz band around the free-plate resonance at 250 Hz. Figure 10 shows results in the 25-Hz band around the clamped plate resonance at 334 Hz. An average reduction of about 15 dB was achieved in both cases. The measurements with the control on were taken after the control algorithm had converged, using a sweep rate of 0.25 Hz/s. The results obtained show that the controller can successfully adapt to control the system as the excitation frequency varies.

C. Acoustic radiation control

To control acoustic radiation, the error signal used by the controller was obtained from a microphone instead of an accelerometer. The microphone was suspended approximately 30 cm (1 ft) above the plate in an arbitrary off-center location. Figure 11 shows the results of controlling the free-plate acoustic radiation resonance at 724 Hz. In this case, a reduction of 17 dB was obtained. As shown in Fig. 11, peaks are introduced in the radiation from the controlled plate, which suggest that the controller is changing the modal distribution in the plate response, in addition to reducing the response at the drive frequency. Similarly, in Fig. 12, a reduction of 14 dB was obtained at the 484-Hz clamped plate resonance. In addition, the first harmonic at 968 Hz was reduced by 12 dB and the second harmonic at 1452 Hz was reduced by about 20 dB.

The results obtained show that acoustic radiation was controlled as successfully as vibration for single-frequency

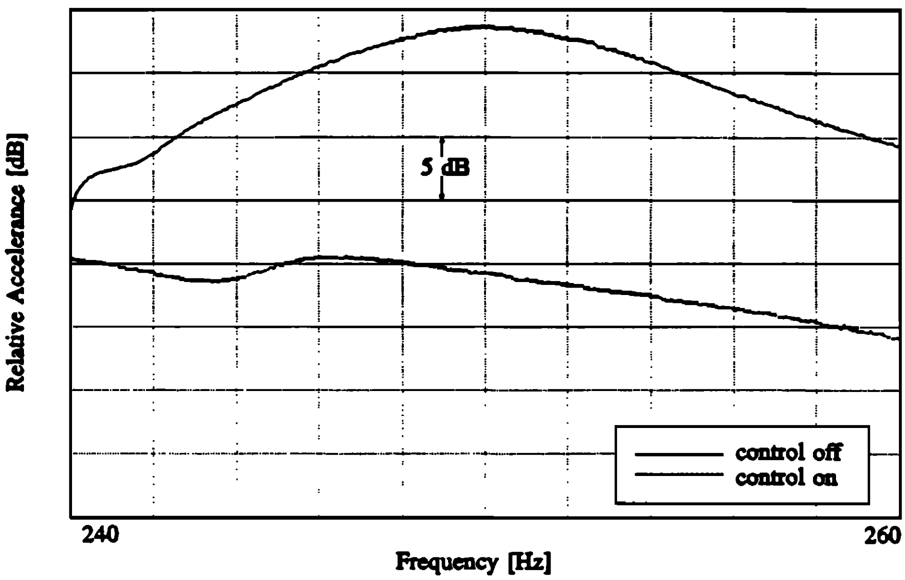

FIG. 9. Vibration control while sweeping through 250-Hz free resonance.

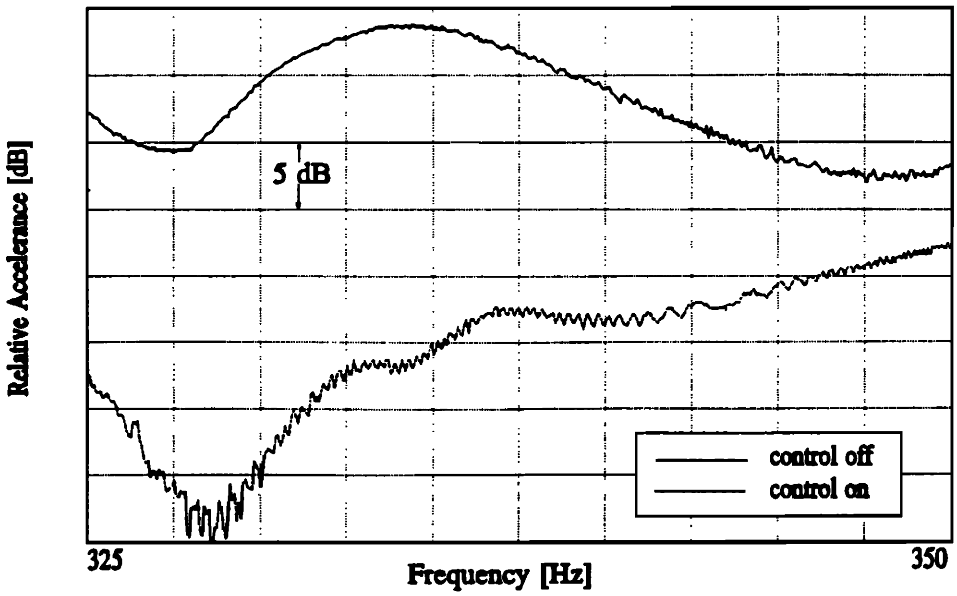

FIG. 10. Vibration control while sweeping through 334-Hz clamping resonance.

excitation. These results, however, are for control at a single-measurement location which does not guarantee global radiation control. Although changing the boundary conditions to detune the plate probably resulted in less radiation globally since the plate was being forced in a condition where it vibrated less efficiently, experimental confirmation is required.

IV. CONCLUSIONS AND RECOMMENDATIONS

It has been verified that controlling plate vibration and acoustic radiation by adaptively varying the boundary conditions is possible. The plate can be continuously detuned from resonance for both constant frequency and, more importantly, variable frequency excitation. This type of control was effective at low frequencies where traditional passive methods of noise control fail. Because the method depends on shifting resonance responses, it is likely not to be effective at higher frequencies where high modal overlap is usually prominent.

One of the main limitations encountered was the method of clamping the plate edges. Hydraulics were used purely for demonstration purposes. Noise from the pump prevented control of acoustic radiation with a variable frequency source. Clearly, improvements in the actuator at the plate boundaries are required before this technique can be implemented outside the laboratory. It may be possible to employ shape-memory alloy fibers at the edges of plates to achieve the desired control.

Improving the control algorithm should also be investigated further. For this research it was left as simple as possible since the objective was to demonstrate that boundary

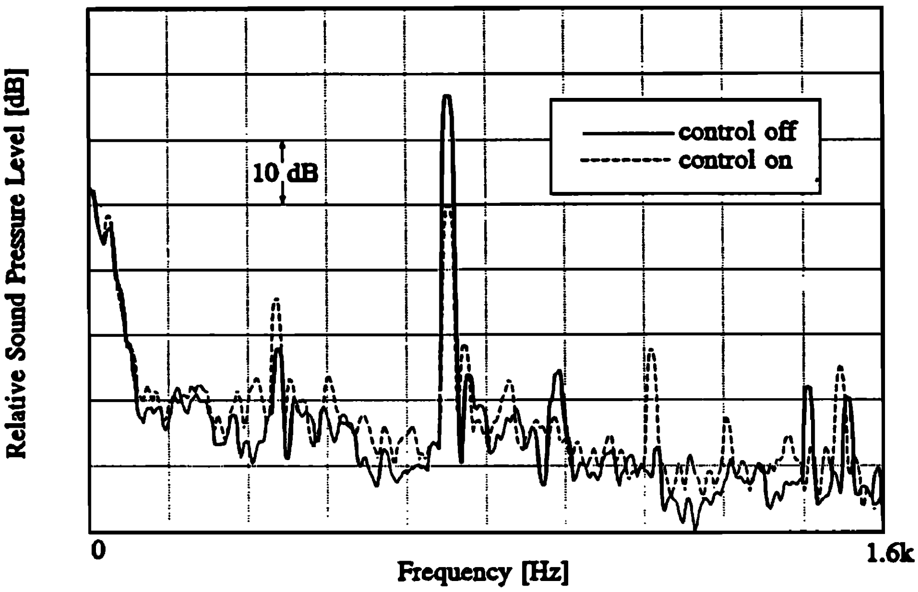

FIG. 11. Acoustic radiation control of 724-Hz free-plate resonance.

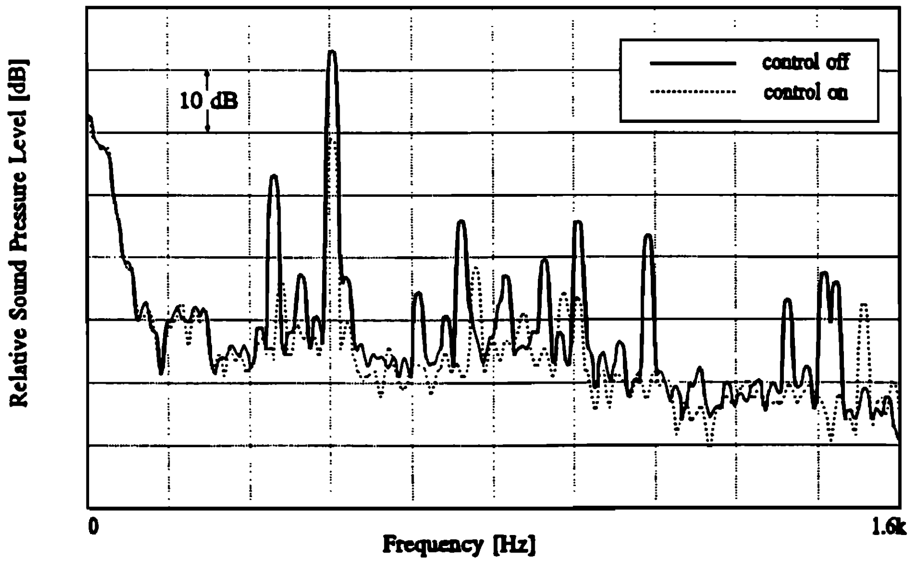

FIG. 12. Acoustic radiation control of 484-Hz clamped plate resonance.

control could be effective in reducing vibration and radiation from plates. Another possibility would be to minimize the acceleration at multiple points over the plate surface in an attempt to achieve more global reduction. As well, further study should be done with the current control method to determine global effects of controlling with only one error sensor.

Combining passive and adaptive controls would prove to be an effective means of noise control over wide frequency ranges. The technique presented in this paper would prove especially useful in situations where machinery attached to a resonant plate provides a variable frequency excitation to the plate.

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