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REDUCTION OF PLATE VIBRATION AND ACOUSTIC RADIATION VIA ADAPTIVELY CONTROLLED BOUNDARIES

by

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Chapter 1

INTRODUCTION

Unwanted noise and vibration in structures is becoming more of an issue as society becomes more technologically advanced. Controlling the noise or vibration in structures has typically been accomplished passively. Examples of passive methods of noise and vibration control are partitions, vibration isolation mounts, and free and constrained layer damping [1,2,3]. These types of control are more effective at high frequencies than low frequencies and they do not easily adapt to different operating conditions, thereby limiting their applicability. More recent approaches to controlling structural vibration are based on active control schemes which use a secondary vibration source to counteract the unwanted primary vibration. These methods of control are most effective at low frequencies and are often implemented adaptively, such that they can automatically adapt to changes in the controlled system. The research presented in this thesis approaches the control of structural noise and vibration adaptively, but without using a secondary vibration source. The control used continuously adapts to changes in the system so as to change the passive properties of the system.

This chapter gives a synopsis of the history of active and adaptive control, defines the problem approached in this thesis, and gives a brief overview of the remainder of the thesis.

1.1 History

The earliest work on active control was performed by Lueg [4] in the 1930's. He outlined the principle of canceling an unwanted noise signal in a duct using a second signal of opposite phase. The system used a microphone to detect the unwanted noise. This signal was then amplified and sent to a loudspeaker which was positioned at a location such that the sound from the loudspeaker was 180 degrees out of phase with the sound in the duct at that location. As a result, the sound wave produced by the loudspeaker canceled the unwanted sound downstream of the loudspeaker in the duct. However, this original system suffered from feedback problems, which led to oscillations in the system. In the 1950's, Olson [5, 6] developed a free-field electronic sound absorber based on the principle of feedback control. His system, however, would reduce the unwanted noise in the zone of the microphone only.

With the technological advances which began in the 1960's, interest in active noise control increased. The development of the field has been thoroughly chronicled [7, 8]. Much of the work done involved controlling the sound field in ducts. Early approaches to active noise control used preconfigured filters that fit the specific problem. With the advent of adaptive control, the filters were incorporated into the electronic system and constantly reconfigured based on the input data. Widrow *et al.* [9] developed the Least Mean Squares (LMS) algorithm, which is very practical for adaptive noise control. The applications which they investigated involved electrical noise, and include canceling 60 Hz interference in electrocardiography (ECG), canceling the donor ECG during heart transplants, and removing the maternal ECG for a fetal heart monitor, to name a few.

Active control using adaptive control algorithms has only recently been extended to structural vibration problems. One of the basic principles behind active control is that an arbitrary signal can be canceled when the time required to process and actuate the control signal is less than the time it takes the original signal to propagate through the medium past the control actuator. In general, the propagation time in structures is much shorter than the propagation time in airborne applications. Hence, there is less time to determine a control signal. Only recently has powerful signal processing hardware made active adaptive control of structures practical.

A single-degree-of-freedom spring-mass system has been studied by many as a simple test model for active control of structures. Lee and Sinha [10] have studied the design of an active vibration absorber. The absorber was designed to completely suppress periodic vibration at the resonance condition of the system, and performed better than an optimal passive absorber. Sommerfeldt [11] developed an adaptive control system for a vibration isolation mount. This system is capable of tracking system changes while supplying optimal control for periodic excitation and correlated broadband excitation at low frequencies.

Control of distributed-parameter systems, such as beams, has primarily been approached with either modal control [12, 13, 14, 15] or control of power flow [16, 17, 18]. Independent control of structural modes requires that the number of actuators should at least equal the number of modes to be controlled. Usually the lowest modes are controlled and control spillover to the higher modes may occur if the controller excites uncontrolled modes. To control power flow, a control force is similarly needed for each type of wave excited in the beam in order to achieve effective control. Active control has also been applied to structures such as gyroscopic spacecraft [19], high-speed rotors [20], and suspension systems [21].

Active control of plate vibration and acoustic radiation has previously been approached by two methods. The traditional approach to active noise control uses an array of acoustic control sources in the radiated field. An alternative approach uses active control forces applied directly to the radiating plate.

Deffayet and Nelson [22] have conducted a theoretical investigation to determine the minimum sound power output of a simply-supported plate when the radiation is controlled using secondary monopole acoustic sources. The results presented are applicable for low frequencies when the plate dimensions are less than the acoustic wavelength. They predicted that good global control could be achieved using this method, but that the number of secondary acoustic sources needed was related to the plate modal order. Consequently, for controlling higher-order plate modes many acoustic sources would be required.

Early work involving vibration control of plates using active point forces [23, 24, 25, 26] showed that substantial vibration reduction could be expected. More recently, Fuller [27] analyzed active control of acoustic radiation from vibrating plates by using oscillating point forces applied directly to the structure. The results showed that with one or two control forces global attenuation of broadband radiated sound could be achieved. These results were applicable for low to mid-range frequencies.

Fuller *et al.* [28] also performed an experimental comparison of the two methods of using acoustic sound sources and point force sources. They determined that each method worked by a different mechanism. Acoustic sources altered the radiation impedance seen by the plate, thus reducing the power radiated from the plate. Point force control worked by modal suppression, where the vibration response of the plate to all significant modes was reduced. They concluded that a single control force was more effective at achieving global noise control for pure-tone excitation than several acoustic sources.

Recent work at Penn State by Bischoff [29] involved a multichannel adaptive control scheme to attenuate the vibration of a mounted plate system. Inertial forces were used near the mounting points on the plate to attenuate the vibration transmission to the system foundation. He found the control scheme to be an excellent method of controlling low frequency vibration transmission.

All of these examples of noise control have been active, either adaptive or non-adaptive. Little work has been done using passive methods of noise control which can be controlled adaptively.

1.2 Problem Description

This thesis explores the possibility of adaptively controlling plate vibration and acoustic radiation by varying the boundary conditions of the plate. The resonance frequencies of the lower-order modes of a plate are sensitive to boundary conditions. The apparatus used to validate the control approach is able to approximate boundary conditions which vary from free to clamped by using hydraulic actuators located at the edges of a vibrating plate. The control algorithm implemented in this research varies the clamping pressure at the plate edges to take advantage of the sensitivity of the plate's resonance frequencies to the boundary conditions. This results in continuously detuning the plate resonance frequency from the frequency of excitation.

The plate is excited by an inertial force at one of its resonance frequencies and the resulting vibration or acoustic radiation attenuated with the adaptive control. Whether the frequency of excitation is a resonance frequency for a free or clamped plate, the controller varies the clamping pressure at the plate edges to minimize the plate vibration or acoustic radiation. This change in the boundary conditions effectively shifts the resonance frequency of the plate, thereby detuning the plate resonance from the frequency of excitation. The controller is also able to reduce vibration in the presence of variations in the frequency of excitation. This method can be applied in the control of noise in situations where a machine acts as a variable frequency source of pure-tone excitation to a plate that acts as a sounding board.

1.3 Thesis Overview

In Chapter 2, the apparatus used to obtain experimental results is described. Chapter 3 describes the basis of the control approach and formulates the strategy used to provide control of the plate vibration and radiation. Results of both vibration and acoustic radiation control of the plate are given in Chapter 4. Conclusions and recommendations for further research on controlling plate vibration and acoustic radiation are given in the last chapter.

Chapter 2

EXPERIMENTAL APPARATUS

The apparatus used to obtain experimental results consists of a mechanical plate and mounting system controlled by an electronic system. This chapter describes both systems.

2.1 Mechanical System

The mechanical system consists of a rectangular aluminum plate, mounted in a frame which contains hydraulic pistons used to vary the clamping pressure at the edges of the plate. A schematic diagram of the system is given in Figure 2.1. The dimensions of the plate are 0.013 m x 0.61 m x 0.91 m (1/2 inch x 2 feet x 3 feet). The plate is mounted between two steel frames in a steel box. The edges of the plate are fitted with aluminum strips milled to a sharp edge, which rest in grooves cut in the frames. A detailed diagram is given in Figure 2.2. These strips are used to reduce the damping of the plate by diminishing the amount of contact between the plate and the frames, thereby reducing the energy lost at the edges of the plate. The use of these strips results in an average measured damping loss factor of 0.02 for the system, as shown in Figure 2.3.

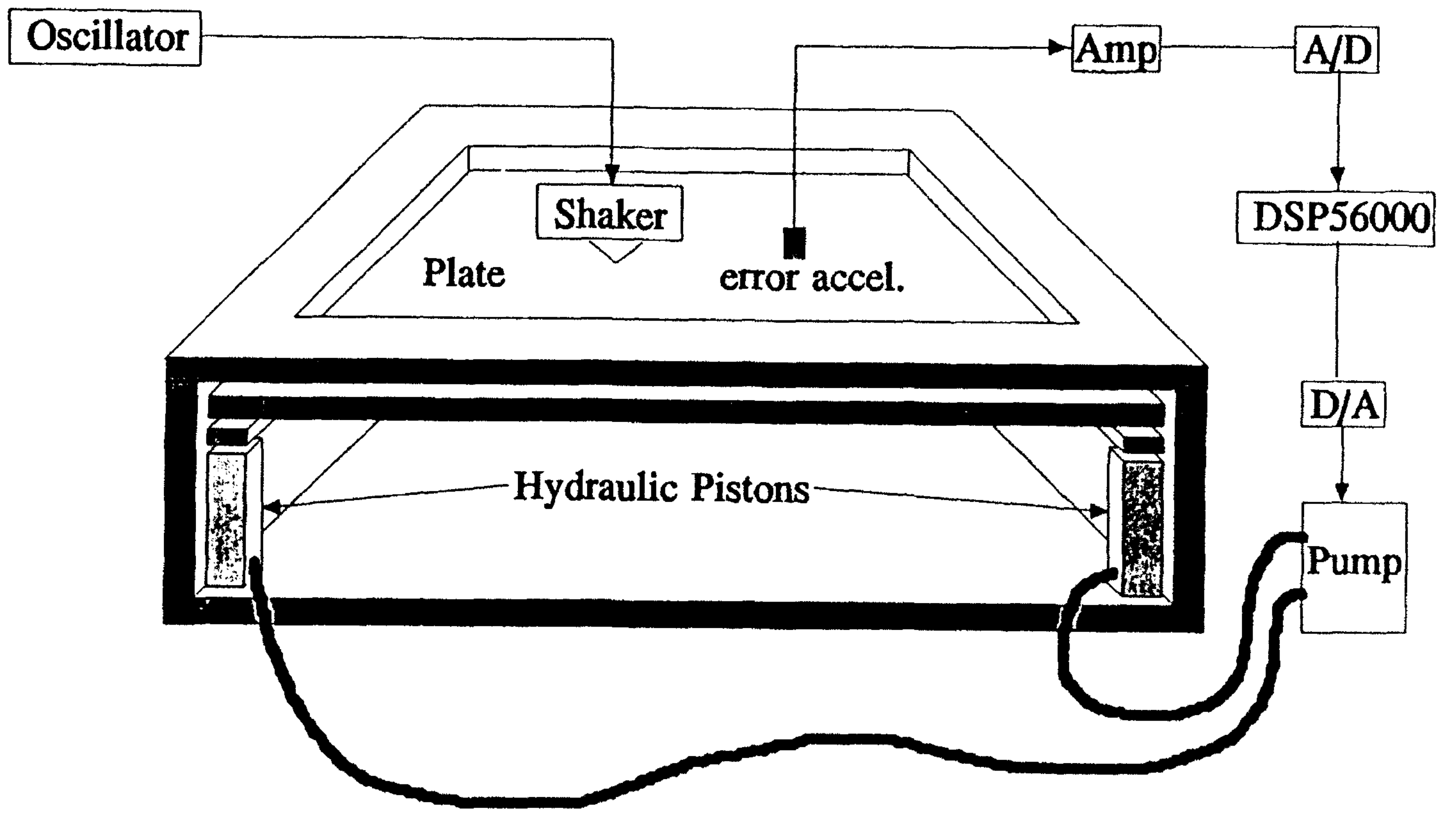

Figure 2.1: Experimental Apparatus

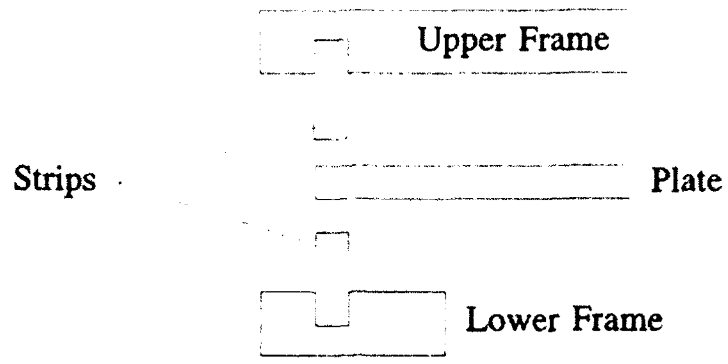

Figure 2.2: Strip Configuration

The plate is excited by a Wilcoxon Research F4 shaker. This shaker has a resonance frequency at 30 Hz and a usable frequency range of 25 to 40,000 Hz. The shaker is mounted directly on the plate in the center to provide an inertial force to excite the plate. The shaker is fitted with a Wilcoxon Research Z820 impedance head to provide input information for the control system.

The hydraulic pistons, which provide a clamping force used for control, are located inside of the box underneath the lower frame. One piston is located at each corner of the frame for a total of four pistons. By controlling the pistons, the upper and lower frames are capable of exerting a variable clamping force uniformly on the boundaries of the plate. This clamping approximates the different boundary conditions needed to vary the frequencies of resonances of the plate to validate the control approach.

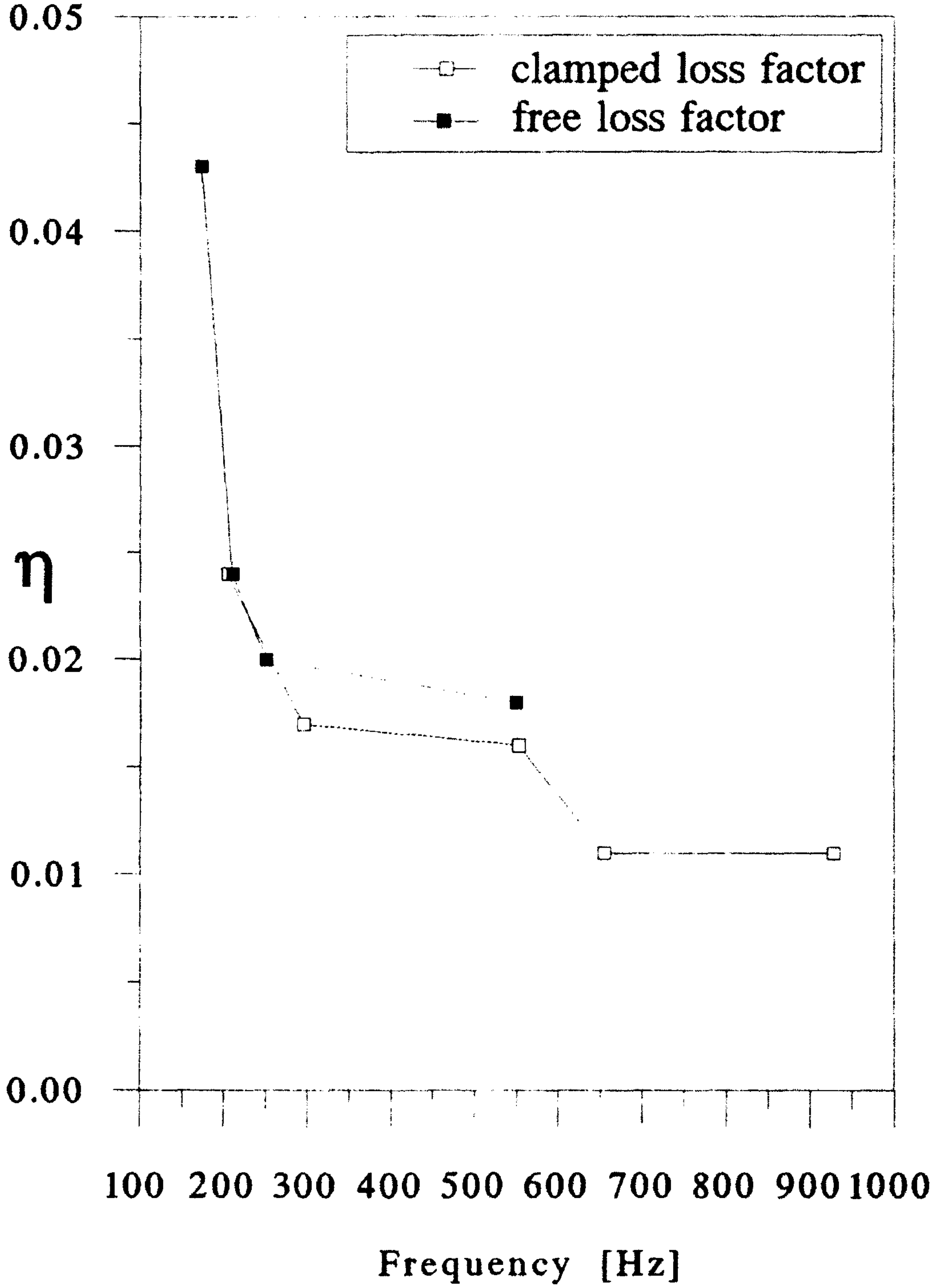

Figure 2.3: Measured Loss Factors

A Vickers EHST-3 hydraulic pressure control provides the interface between the electronic control system and the hydraulics. This control valve regulates the pump pressure in proportion to input voltage. The input signal range is rated at 0-10 volts DC, which is easily provided by the electronic control system. This range of input voltages results in pump pressures of approximately 90-900 psi. A totally released state, or 0 psi, is not directly obtainable using the electronic pressure control, so a 300-pound counteractive weight is attached to the lower frame to provide approximately 0 psi at the edges of the plate with the lowest hydraulic pressure. With the counteractive weight, a pump pressure of 90 psi results in nearly free boundary conditions, while a pump pressure of 900 psi results 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 both conditions are approximations.

2.2 Electronic System

The adaptive control is implemented with the use of a Motorola DSP56000ADS digital signal processing board. The DSP56000 is a 56-bit general purpose digital signal processor. The processor features a parallel internal architecture, which makes it possible to move one set of data while simultaneously processing another. The 62 instruction mnemonics include a DO loop and a repeat instruction which simplify programming. The DSP56000 is interfaced with an IBM-AT computer. The assembly language code is written on the computer and then downloaded to the board. An adaptive control algorithm is implemented on the board and will be discussed in detail in Chapter 3.

The input analog signal from the control accelerometer or microphone is converted to a digital signal that the DSP board can use, and the digital output signal from the board is converted to an analog signal to send to the hydraulic pump. This is accomplished using an Ariel ADC56000 I/O board. This 16-bit converter is designed to interface directly with the Motorola DSP56000. The ADC56000 plugs into the DSP56000 and allows two channels each of analog to digital (A/D) and digital to analog (D/A) conversion.

Additional electronic hardware included Ithaco amplifiers for the output acceleration and force signals, an Adcom amplifier to power the shaker, and various DC power supplies for the hydraulic control valve.

Chapter 3

CONTROL STRATEGY

Most approaches to adaptive control of structures have been active. Active control requires that the control signal be determined before the original signal propagates through the medium past the control actuator. Since the control strategy implemented in this thesis is not time dependent, it is considered "passive" adaptive. The control algorithm adapts to system changes, but does not require providing control at the excitation frequency, since counteracting the primary vibration is not the mechanism used to control the system. This chapter details the working principle behind the control approach, and explains the control algorithm.

3.1 Working Principle

In 1988, Hammouda [30] investigated the effects of boundary conditions on the resonant response of finite plates. He showed that the resonance frequencies and the associated response amplitudes were dependent on the boundary conditions of the plate. The algorithm implemented in this thesis uses this property to continuously detune a plate by adaptively varying the boundary conditions.

The boundary conditions of a plate were modelled by Hammouda with three variable strength springs. This model is shown in Figure 3.1. Here K_1 was a linear spring constant, and both K_2 and K_3 were torsional spring constants. By varying these spring constants, different boundary conditions could be approximated. A clamped boundary was obtained when all three spring constants approached infinity, and free conditions were reached when all three were set to zero. Simply-supported conditions were obtained when both K_1 and K_2 went to infinity while K_3 was set to zero.

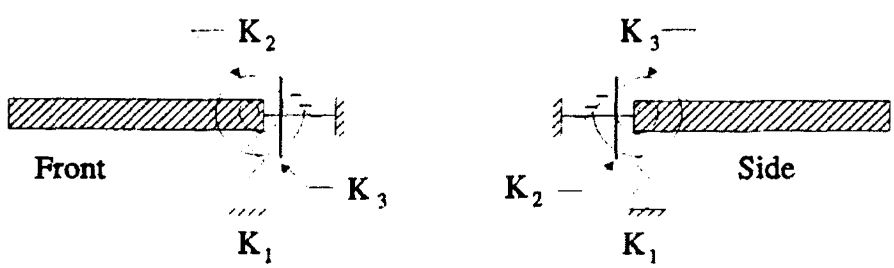

Figure 3.1: Boundary Condition Model

As K_1 was varied from zero to infinity, the boundary conditions changed from free to simply-supported. Figure 3.2 shows the response of a plate to variations in spring constant K_1. The change in resonance frequency produced by changes in the boundaries was seen to have three distinct regions. In the first area, the resonance frequency was that for the free plate. The second zone showed the transition from the free resonance frequency to the simply-supported resonance frequency. Finally, the third region showed that above some finite spring constant the plate responded as if it were simply-supported.

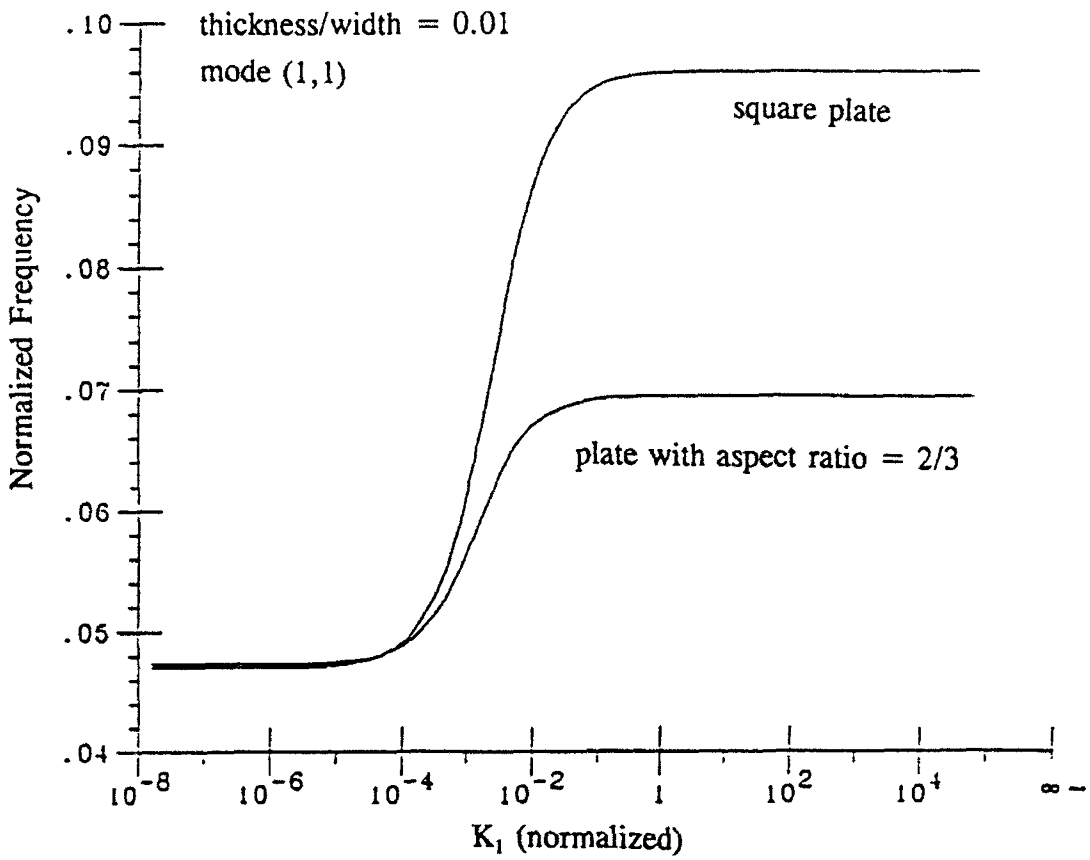

Figure 3.2: Resonant Response of Plate to Variations in K_1.

(Source: Hammouda)

This shift in resonance frequency with changing boundaries was found to be most pronounced for the lower-order modes of a plate. The higher-order modes are not as sensitive to the boundary conditions. At the higher frequencies, more wavelengths exist across the plate. Since the effect of the boundary conditions are proportional to the wavelength, the boundary conditions affect a smaller percentage of the plate area at the higher frequencies than at the lower frequencies. Therefore, the boundaries have less influence on the higher frequencies of resonance.

To verify that this shift occurs 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 input from 0 to 1000 Hz is applied to the shaker to produce a force excitation. The resulting acceleration is measured using the impedance head of the shaker. Figures 3.3 and 3.4 show the accelerance (acceleration normalized by force) of the plate. Each curve is the result of applying a different clamping pressure at the plate boundaries. The lowest pressure, 90 psi, corresponds to free boundary conditions. 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 between the three curves in Figure 3.4. This indicates that the shift in the resonance frequencies is mostly complete when the clamping pressure reaches 200 psi.

3.2 Control Algorithm

The algorithm used to control plate vibration and acoustic radiation in this thesis is based on a simple direct search strategy. "Direct search," as described by Hooke and Jeeves [31], involves comparing trial solutions of the problem to the "best" solution obtained from previous trials. The result of each comparison is used to determine the next trial solution. This is a simple approach to adaptively controlling the plate, and is well suited for use with modern signal processing hardware.

For controlling the boundary conditions of the plate, this strategy is implemented as shown in the flow chart in Figure 3.5. The input data (a_n) is sampled at a rate of

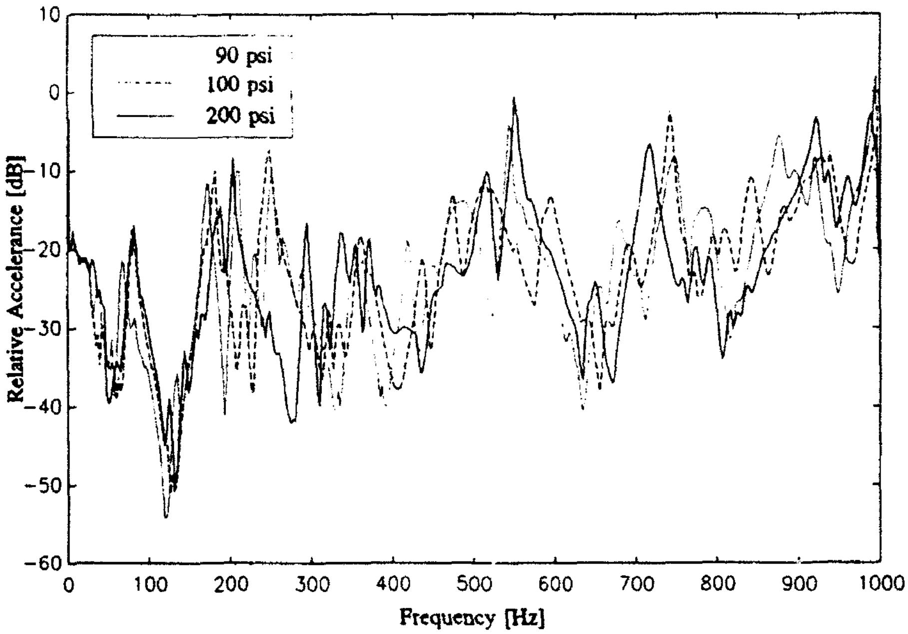

Figure 3.3: Accelerance of Plate at Lower Clamping Pressures

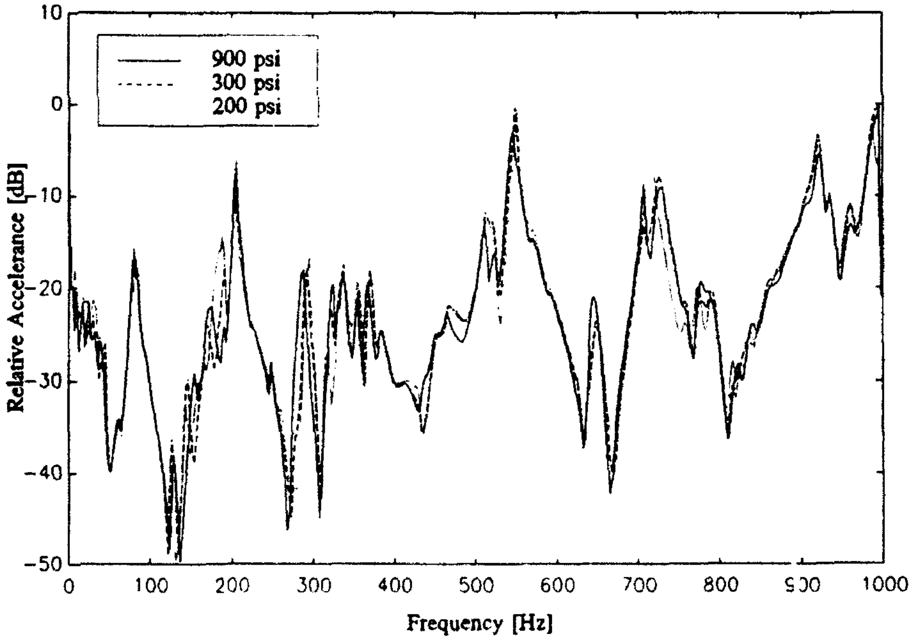

Figure 3.4: Accelerance of Plate at Higher Clamping Pressures

1 kHz, and is either acceleration in the case of vibration control, or acoustic pressure in the case of acoustic radiation control. The control signal output is sent directly to the hydraulic pressure control which produces a hydraulic control pressure proportional to the control signal voltage.

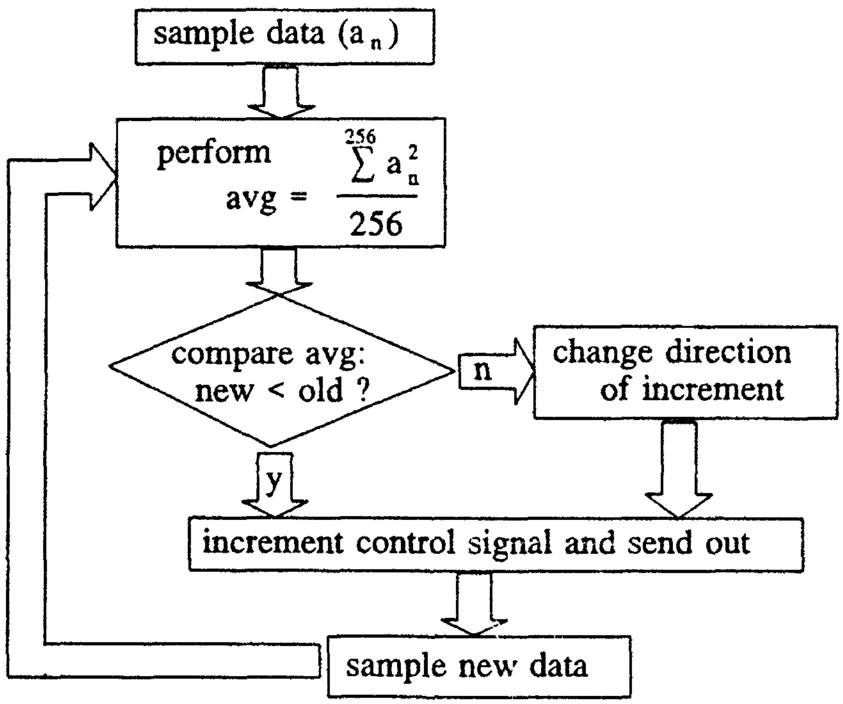

Figure 3.5: Control Algorithm Flow Chart

For the direct search strategy, this means that the trial solutions are the changes in the control signal used to change the boundary clamping. The solutions are compared by comparing the corresponding average vibration or radiation levels which they produce. The result of the comparison determines if the control signal has been changed to increase or decrease the plate response. This information is then used to determine the next control signal value.

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

Three versions of this algorithm were implemented. The first version was implemented as outlined in the flow chart with a constant increment, or step size. This version converged slowly, and once converged it would constantly 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 the control signal if the difference between averages was larger than a preset value determined by trial and error (0.66 in this case). 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 fourteen (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.

Chapter 4

EXPERIMENTAL RESULTS

This chapter presents experimental results for control of both vibration and acoustic radiation of the plate system described in Chapter 2. Vibration control was attempted for both single frequency and variable frequency excitation. Acoustic radiation control was undertaken only for single frequency excitation.

4.1 Vibration Control

Vibration control is achieved by seeking the minimum acceleration of the plate at a point. The controller uses the acceleration data to vary the boundary conditions as described in Chapter 3 to attempt to minimize the vibration level of the plate. The control is based on acceleration, not accelerance, because the levels of the forces exciting a panel are rarely known and the acceleration is correlated better with radiation than accelerance. Therefore, expected changes in the plate response with control is reflected in changes in the acceleration levels in response to changes in edge clamping pressure. An example of acceleration levels measured with different clamping pressures is presented in Figures 4.1 and 4.2. It should be noted that the acceleration does not change much above 200 psi. The expected change in the acceleration level produced

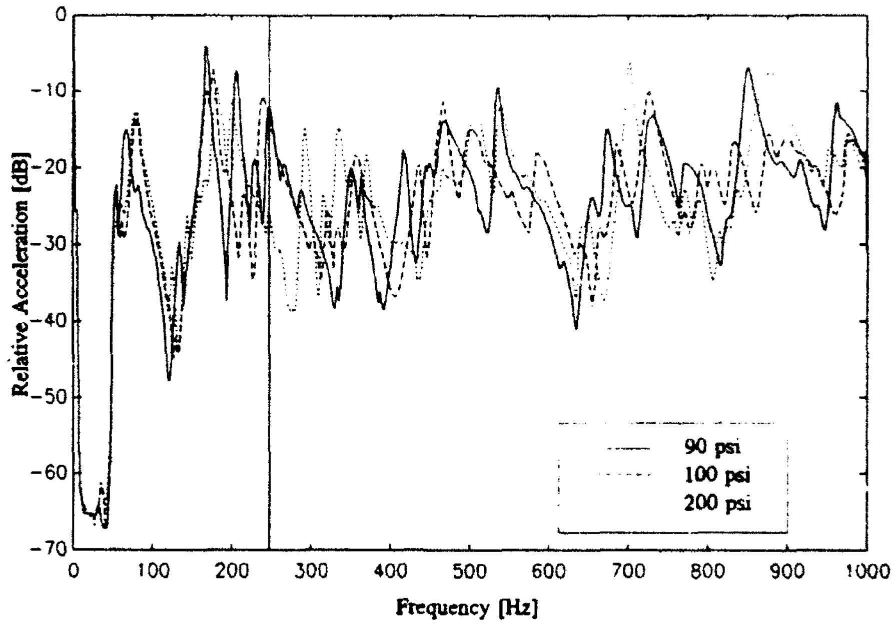

Figure 4.1: Acceleration at Low Clamping Pressures

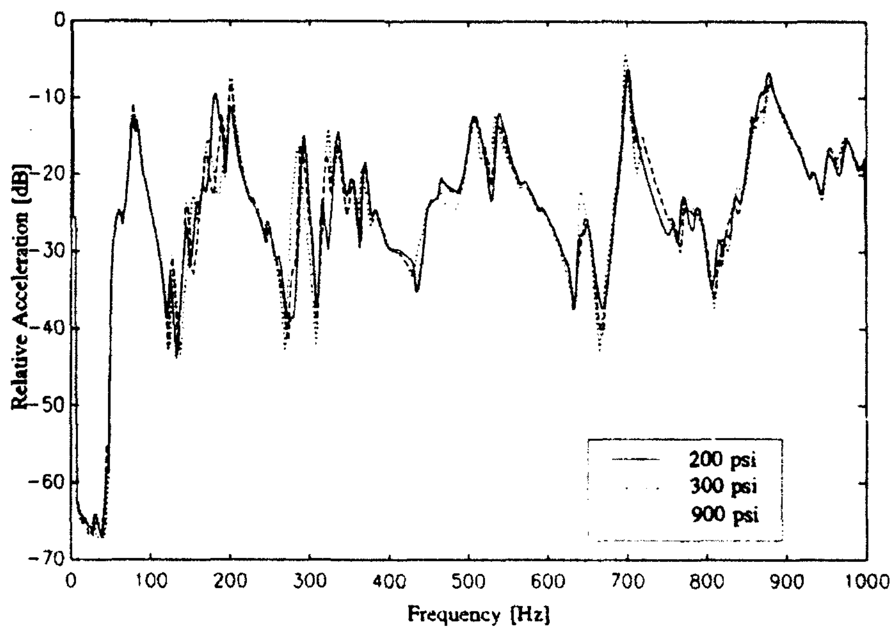

Figure 4.2: Acceleration at High Clamping Pressures

by the adaptive control at the 250 Hz, 90 psi resonance frequency is illustrated. The expected change is the difference between the level at resonance without control and the minimum of the levels obtained by changing the clamping pressure. In this case approximately 15 dB reduction when the clamping pressure changes from 90 psi to 200 psi.

4.1.1 Single Frequency Vibration Control

To test the control system, various resonance frequencies are chosen at 90 psi and 900 psi. The plate is continuously excited at each frequency individually and the controller turned on. The level of the response is recorded before the controller is started and after the controller has 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 90 psi plate resonance frequencies are given in Figure 4.3. The reduction obtained varies from 3 dB to 23 dB, with an average value of 13 dB. Similar results for some of the 900 psi resonance frequencies are shown in Figure 4.4. Here, the reduction obtained varies from 0 dB to 15 dB, with an average value of 8 dB. These reductions were achieved in the presence of the relatively high damping shown in Figure 2.3. 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 control pressures along with the actual reduction

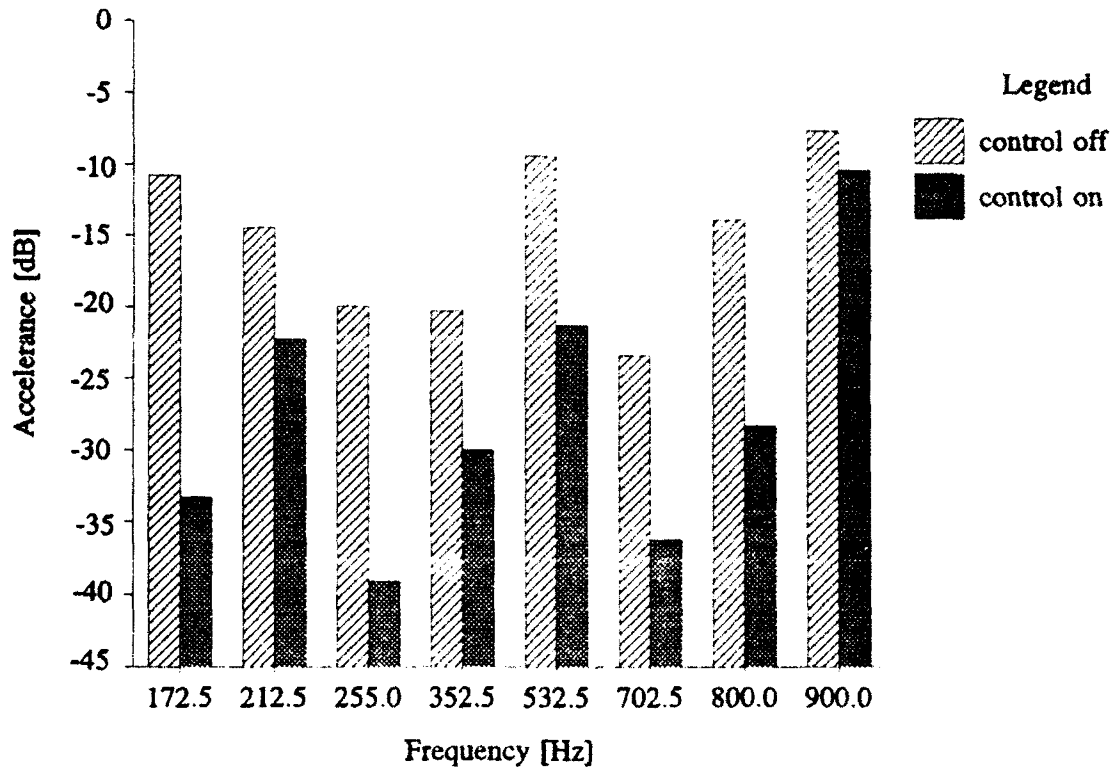

Figure 4.3: Vibration Control of 90 psi Plate Resonances

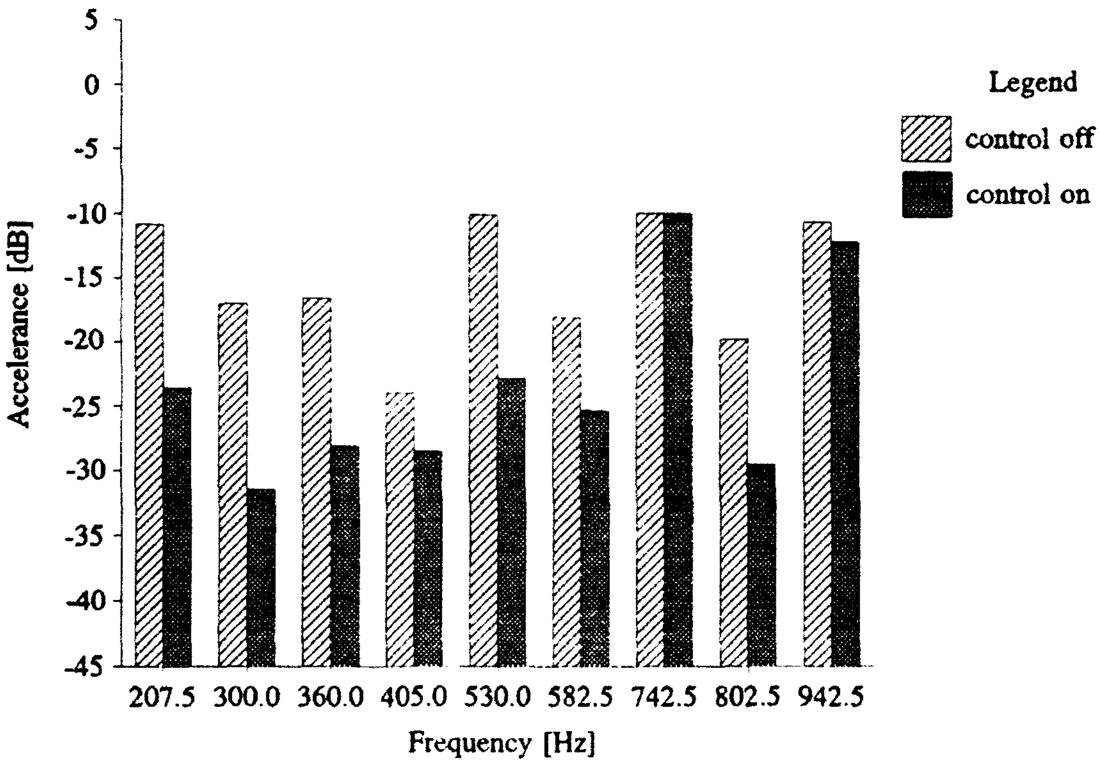

Figure 4.4: Vibration Control of 900 psi Plate Resonances

obtained for each frequency are listed in Tables 4.1 and 4.2. The frequencies where less reduction occurs tend to be the frequencies where the peaks overlap for different pressures. The pressure needed to control the free resonance frequencies is, in general, not much more than the original 90 psi. In other words, minimal clamping pressure is enough to control the free resonances since most of the plate's frequency shift is complete by 200 psi. For the clamped resonances however, the controller had a hard time reaching the 90 psi conditions needed to "unclamp" the plate and achieve the expected reduction. This is 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.

Table 4.1: Vibration Control Results for 90 psi Resonances

Resonance Frequency [Hz]	Final Control Pressure [psi]	Actual Reduction [dB]
172	275	23
212	150	8
255	150	19
352	100	10
532	100	12
702	100	13
800	150	14
900	625	3

Table 4.2: Vibration Control Results for 900 psi Resonances

Resonance Frequency [Hz]	Final Control Pressure [psi]	Actual Reduction [dB]
208	100	13
300	150	15
360	150	12
405	100	5
530	90	13
582	200	7
742	900	0
802	90	10
942	300	2

4.1.2 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 is run continuously while the excitation sweeps 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 Figures 4.5 and 4.6 for free and clamped plate resonances respectively. Figure 4.5 shows the results of control in the 20 Hz band around the free plate resonance at 250 Hz. Figure 4.6 shows results in the 25 Hz band

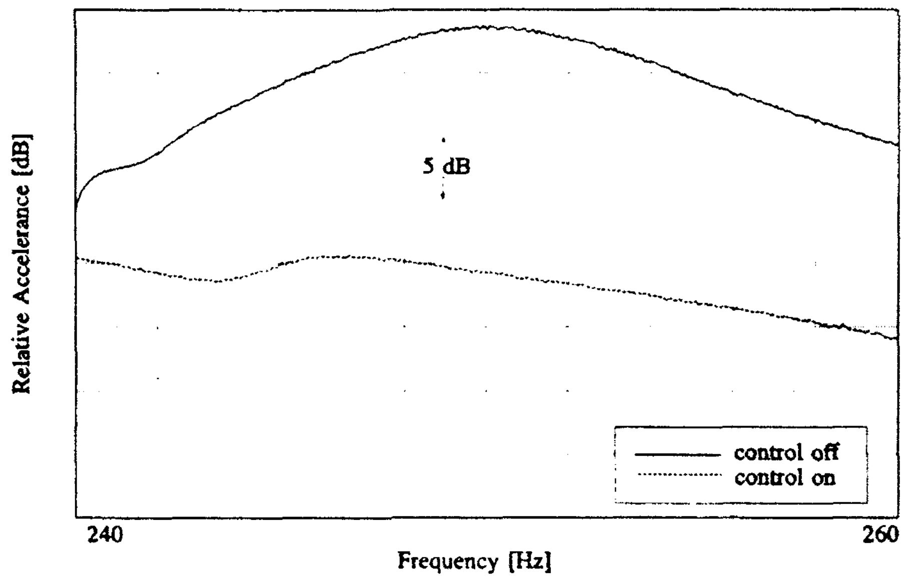

Figure 4.5: Vibration Control While Sweeping Through 250 Hz Free Resonance

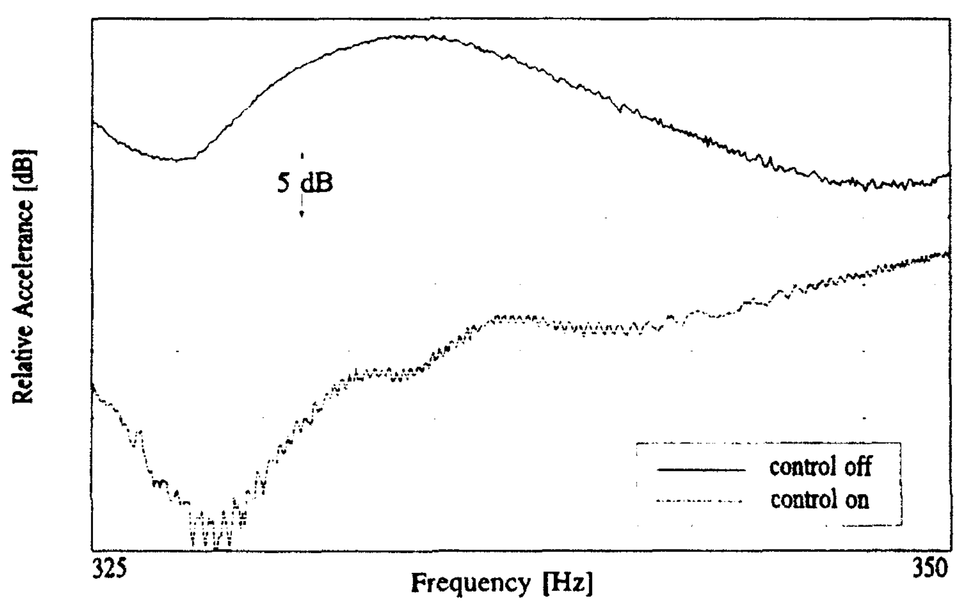

Figure 4.6: Vibration Control While Sweeping Through 334 Hz Clamped Resonance

around the clamped plate resonance at 334 Hz. An average reduction of about 15 dB is achieved in both cases shown. The measurements with the control on are taken after the control algorithm has converged using a sweep rate of 0.25 Hz/sec. The results obtained show that the controller can successfully adapt to control the system as the excitation frequency varies.

To determine how fast the controller can track changes in the excitation frequency, different sweep rates are tried. The controller is left running continuously while the frequency is constantly swept back and forth through a preset band. The sweep time (the same for sweeping either up or down) is reduced in steps to determine the limits of the controller.

For the 250 Hz resonance at 90 psi, the controller is effective down to a sweep rate of 15 Hz/sec. At that point, the controller briefly tracks incorrectly approximately once every five sweep cycles. For the 334 Hz resonance at 900 psi, the control tracks correctly only 50% of the time at 0.67 Hz/sec, and only 25% of the time at 2 Hz/sec. Note the large difference in tracking rates due to the lower sensitivity of the 900 psi resonances to clamping pressure. Once the controller starts tracking, it works well for a few sweep cycles. However, it soon starts mis-tracking. This is probably due to the difficulty in obtaining a full release without manual intervention. Typically, the controller overshoots the correct control pressure by several hundred psi and then requires several more sweep cycles to reach the lower correct control pressure again. After the controller overshoots the correct control pressure, it takes a long time to recover since the plate basically behaves as if it were fully clamped above 200 psi. This means that the response is fairly constant above 200 psi. Since the controller relies on the variability in the response to determine the control pressure, it requires more time to control when the response is not changing much. Limiting the maximum control pressure may be one way to improve the tracking ability for clamped frequencies.

4.2 Acoustic Radiation Control

To control acoustic radiation, the error signal used by the controller is obtained from a microphone instead of an accelerometer. Control is achieved by minimizing the sound pressure level at the microphone location. The microphone is suspended approximately 30 cm (approx. 1 foot) above the plate in an arbitrary off-center location.

One difficulty in implementing acoustic radiation control is the background noise level produced by the hydraulic control system. The hydraulic pump produces noise at harmonics of 366 Hz as shown in Figure 4.7. Control of acoustic radiation is attempted at various single resonance frequencies. The frequencies that are chosen to control are resonance frequencies that fall between the spectral peaks produced by the pump. Figure 4.8 shows the results of controlling the free plate acoustic radiation resonance at 724 Hz. In this case, a reduction of 17 dB is obtained. Similarly, in Figure 4.9, a reduction of 14 dB is obtained at the 484 Hz clamped plate acoustic radiation resonance. In addition, the second harmonic at 968 Hz is reduced 12 dB and the third harmonic at 1,452 Hz is reduced about 20 dB. In both of these plots, the pump harmonics are clearly seen and tend to be more prominent at higher pressures. This is because the pump noise level increases when the pump produces a higher pressure.

The results obtained show that acoustic radiation is controlled as successfully as vibration for single frequency 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 results in less radiation globally since the plate is being forced into a condition where it vibrates less efficiently, experimental confirmation is required.

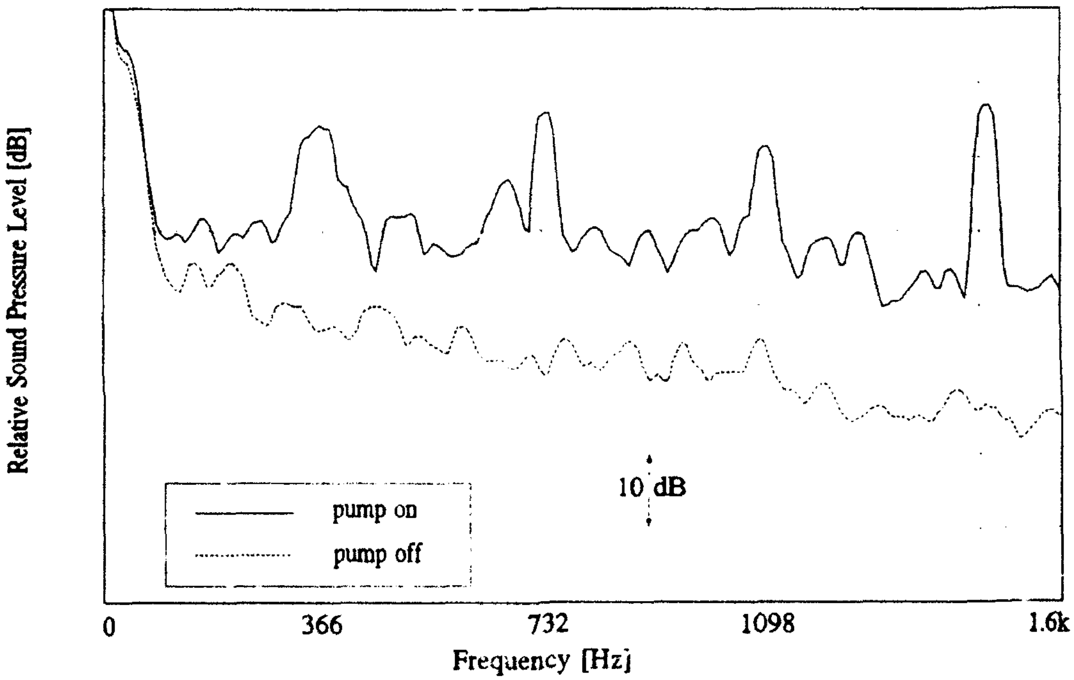

Figure 4.7: Pump Noise Spectrum

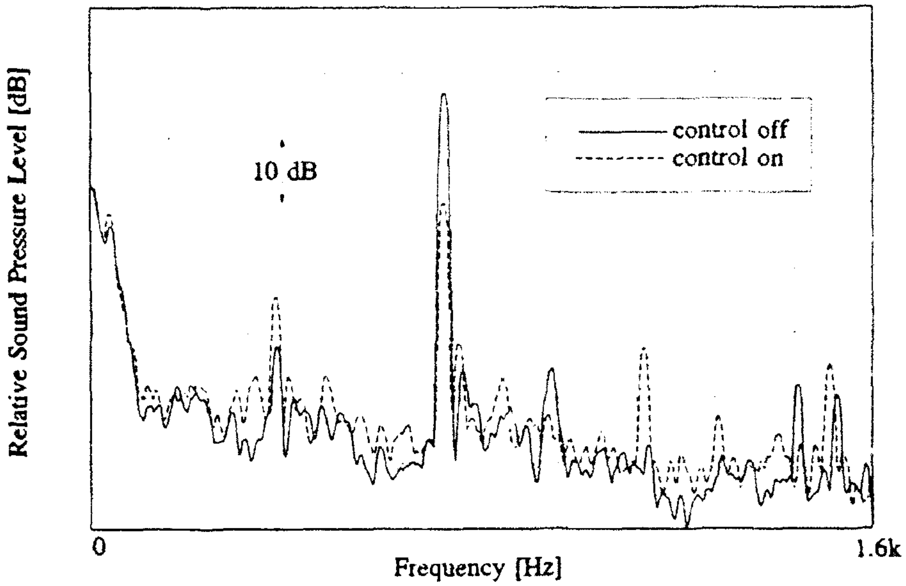

Figure 4.8: Acoustic Radiation Control of 724 Hz Free Plate Resonance

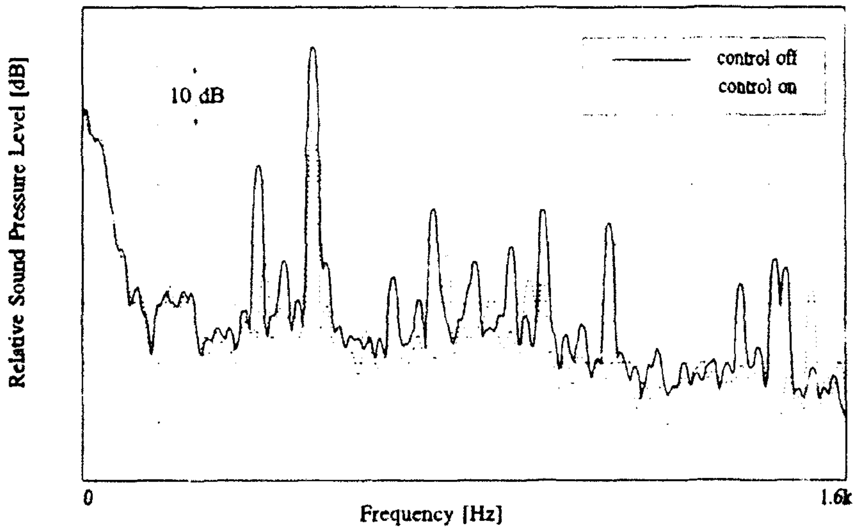

Figure 4.9: Acoustic Radiation Control of 484 Hz Clamped Plate Resonance

Chapter 5

CONCLUSIONS AND RECOMMENDATIONS

This thesis has shown that adaptively controlling plate vibration and acoustic radiation by adaptively varying the boundary conditions is possible. The plate can be continuously detuned from pure-tone excitation for both single frequency and, more importantly, variable frequency excitation. This type of control is effective at low frequencies where traditional passive methods of noise control fail. It is not effective at higher frequencies that exhibit high modal overlap.

One of the main limitations in this project was the hydraulic pump. It produced noise at about the same level as the radiating plate, limiting attempts at controlling acoustic radiation. Variable frequency control of acoustic radiation was not attempted because of the noise produced by the hydraulic pump. This noise could possibly be dealt with by the use of filters, signal processing techniques, or reducing the pump noise at the location of the microphone.

Also, the output hydraulic pressure level was not consistent with the control signal input to the pump. As the pump warmed up its response would vary, so that it would not release fully without manual intervention. Because of this the control algorithm would appear not to be working since the plate would always respond as if it were clamped, and never as if it were free. Clearly, improvements in the actuator at the plate boundaries are required before this technique could be implemented outside the laboratory.

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 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 for an attempt at more global reduction. However, further study should be done with the current control method to determine global effects of controlling with only one error sensor.

Combining traditional and adaptive controls would prove to be an effective means of noise control over wide frequency ranges. The technique presented in this thesis 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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