ACTIVE CONTROL METHODS FOR IMPROVING THE INSERTION LOSS OF ACOUSTICAL ENCLOSURES

by

Aaron J. Layos

Thesis submitted to the Faculty of the Virginia Polytechnic Institute and State University in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

APPROVED:

[Signatures]

Dr. Ricardo A. Burdisso, Chairman

Dr. Cathy R. Guigou

Dr. Alfred L. Wicks

September, 1995

Blacksburg, Virginia
ACTIVE CONTROL METHODS FOR IMPROVING THE INSERTION LOSS OF ACOUSTICAL ENCLOSURES

by

Aaron J. Layos

Committee Chairman: Ricardo A. Burdisso
Mechanical Engineering

(ABSTRACT)

A common method of noise control is the implementation of passive transmission loss enclosures. An acoustical enclosure surrounds a noise source for the purpose of interrupting the noise transmission path. The effectiveness of these enclosures is, however, limited by various drawbacks. These drawbacks include leaks in the enclosure walls, transmitted vibrations to the enclosure due to physical coupling, and the limited reduction of noise at low frequencies. Recent advances in active control have shown the potential for applying some of this technology to passive acoustical enclosures to improve their performance.

Here, further studies were carried out in this research area by developing active control systems for the purpose of reducing the noise radiation from a boxed-shaped enclosure. This enclosure was constructed of Plexiglas and had the capability of either being completely sealed or modified so it contained air gaps. The disturbance source was a small speaker located at the bottom center of the enclosure and driven with a sinusoidal signal. Center frequencies of 1/3 octave bands from 125 Hz to 1600 Hz were used to drive the disturbance speaker. First, some analytical predictions and experimental tests were done to determine the physical and acoustical characteristics of the enclosure. Some active control experiments were then performed on this system. The types of active control systems that were studied ranged from single input, single output (SISO) to
multiple input, multiple output (MIMO). The SISO control systems used an error microphone located in the enclosure interior. It was found that the most effective SISO control set-up used an interior control speaker very close to the disturbance source. Five accelerometers or microphones were used as error sensors and PZT actuators mounted on the five panels were the control sources in the MIMO control systems. The effectiveness of these control systems improved as the error sensors were moved farther from the exterior of the enclosure. A series control set-ups using a single control source and multiple error sensors (MISO) located around the enclosure exterior were also evaluated. The purpose of this system was to compare the effectiveness of exterior error sensors with an interior error sensor used in the SISO experiments. These MISO active control systems used the same control sources that were used in the SISO experiments. It was found that the MISO control systems were more effective when the PZTs were the control actuators and less effective when an interior control speaker was used. Most of the control set-ups improved the insertion loss of the enclosure primarily at frequencies no higher than 400 Hz. The most effective case, a SISO system with a control speaker very close to the disturbance, improved the insertion loss up to 630 Hz. Based on the analytical and experimental work of the passive acoustical characteristics of the enclosure, active control improved the insertion loss in the low frequency region and the beginning of the intermediate frequency region (i.e. low modal density).
Acknowledgments

I would like to thank my major professor, Dr. Ricardo Burdisso, for the opportunity to obtain this degree. This work would not have been fulfilled without his help and guidance. Also, I express my appreciation for Dr. Cathy Guigou and Dr. Alfred Wicks for being my committee members and for their advice and help in completing this thesis. I would also like to acknowledge Dr. Marcus Bronzel for helping me understand the principle of active control and the limitations of the computer codes and hardware. Many thanks to research associates and fellow students for their assistance and friendship: Jerome Smith, Steve Booth, Sam Beyene, Florence Deneufve, Julian Maillard, Hugo Camargo, Hector Rodriguez, and many others who are not named. Last, a very special thanks to my parents Joseph and Susan for their love, encouragement, and patience throughout my degree.
1 INTRODUCTION ................................................................. 1
   1.1 Passive Acoustical Enclosures ............................................. 1
   1.2 General Description of Active Control ................................... 3
   1.3 Previous Work Involving Active Control with Enclosure .............. 4
   1.4 Motivations and Objectives for this Research .......................... 7
   1.5 Research Approach .......................................................... 8
   1.6 Brief Overview of the Thesis .............................................. 9

2 CHARACTERISTICS OF PASSIVE ACOUSTICAL ENCLOSURES AND
   WAYS ACTIVE CONTROL CAN IMPROVE THEM ............................. 11
   2.1 Characteristics of the Passive IL of Sealed and Open Enclosures .......... 11
       2.1.1 Main Frequency Regions ............................................. 12
       2.1.2 Openings and Their Detrimental Effects on Insertion Loss ............. 19
   2.2 Passive Ways of Improving Enclosure Performance ...................... 22
   2.3 Possible Ways Active Control can Improve Enclosures ................. 24

3 EXPERIMENTAL SET-UP .............................................................. 26
   3.1 General Description of Experimental Test Set-up ........................ 26
   3.2 General Description of Test Environment .................................. 34
   3.3 Physical and Acoustical Characteristics of the Enclosure .............. 40
       3.3.1 Analytical Study of the Passive Characteristics of the Enclosure .... 40
3.3.2 Experimental Analysis of the Passive Characteristics of the Enclosure ........... 42

4 ACTIVE CONTROL EXPERIMENTS AND THEIR RESULTS .................. 60

4.1 Active Control Experimental Methodology ....................................... 60
4.2 SISO Control Experiments ....................................................... 62
4.3 Results of the SISO Experiments ............................................... 70
4.4 MIMO Control Experiments ..................................................... 75
4.5 Results of the MIMO Experiments .............................................. 81
4.6 MISO Control Experiments ..................................................... 90
4.7 Results of the MISO Experiments .............................................. 90

5 CONCLUSIONS AND RECOMMENDATIONS .................................... 95

5.1 Conclusions ............................................................................ 95
5.2 Recommendations .................................................................. 97

REFERENCES .................................................................................. 99

APPENDIX A ACTIVE CONTROL ALGORITHM .................................. 102

APPENDIX B CALCULATIONS OF THE PHYSICAL AND ACOUSTICAL
CHARACTERISTICS OF THE ENCLOSURES ................................. 106
List of Figures

2.1 Various Paths of Noise Transmission Through an Enclosure
(source: Ver, [23]) ........................................................................................................ 13

2.2 Passive Insertion Loss (IL) for a Sealed Enclosure
(source: Beranek and Ver, [4]) ................................................................................... 14

3.1 Diagram of the Sealed Enclosure ........................................................................... 27

3.2 Diagram of the Open Enclosure ........................................................................... 29

3.3 Acoustic Driver Used for the Disturbance Source .................................................. 30

3.4 Fold-out View of Enclosure Showing Panel Numbers
and Location of PZT Actuators .................................................................................. 31

3.5 PZT Excitation Used to Produce Pure Bending Moments
(source: Banks et al., [30]) ....................................................................................... 32

3.6 Reverberation Chamber and Experimental Set-up ................................................. 37

3.7 Computer-Generated Plot of the IL for the Sealed and Open Enclosures .......... 41

3.8 Microphone Location for the Acoustic FRF of the Sealed Enclosure ................. 44

3.9 Microphone Location for the Acoustic FRF of the Open Enclosure ................. 45

3.10 FRF Plots of Panel 1 and the Acoustic Volume for the Sealed Enclosure .......... 46

3.11 FRF Plots of Panel 2 and the Acoustic Volume for the Sealed Enclosure .......... 47

3.12 FRF Plots of Panel 3 and the Acoustic Volume for the Sealed Enclosure .......... 48

3.13 FRF Plots of Panel 4 and the Acoustic Volume for the Sealed Enclosure .......... 49

3.14 FRF Plots of Panel 5 and the Acoustic Volume for the Sealed Enclosure .......... 50
3.15 FRF Plots of Panel 1 and the Acoustic Volume for the Open Enclosure............. 51
3.16 FRF Plots of Panel 2 and the Acoustic Volume for the Open Enclosure............. 52
3.17 FRF Plots of Panel 3 and the Acoustic Volume for the Open Enclosure............. 53
3.18 FRF Plots of Panel 4 and the Acoustic Volume for the Open Enclosure............. 54
3.19 FRF Plots of Panel 5 and the Acoustic Volume for the Open Enclosure............. 55
3.20 Passive Insertion Loss for the Sealed and Open Enclosures.......................... 58
4.1 Location of Error Microphone ........................................................................ 63
4.2 Schematic of SISO Control System................................................................. 65
4.3 SISO Control - PZTs Used for the Control Source........................................... 66
4.4 SISO Control - Control Speaker, Configuration 1 ............................................. 67
4.5 SISO Control - Control Speaker, Configuration 2 ............................................. 69
4.6 SISO Control Results -- PZTs Used as Control Source ..................................... 71
4.7 SISO Control Results -- Speaker, Config. 1 Used as Control Source ............... 72
4.8 SISO Control Results -- Speaker, Config. 2 Used as Control Source ............... 73
4.9 Schematic of MIMO Control System.............................................................. 77
4.10 MIMO Control Using Accelerometers as the Error Sensors........................... 79
4.11 MIMO Control Using Microphones as the Error Sensors.............................. 80
4.12 MIMO Control Results -- Accelerometers Used as Error Sensors............... 83
4.13 MIMO Control Results -- Error Mics Placed 12.7 mm Away From Panels...... 84
4.14 MIMO Control Results -- Error Mics Placed 50.8 mm Away From Panels...... 85
4.15 MIMO Control Results -- Error Mics Placed 177.8 mm Away From Panels.... 86
4.16 MIMO Control Results -- Error Mics Placed 25.4 mm Away From Openings .. 87
4.17 MISO Control Results -- PZTs Used as Control Source ....................... 91
4.18 MISO Control Results -- Speaker, Config. 1 Used as Control Source ......... 92
A.1 Block Diagram of Feedforward Adaptive Control ............................. 103
List of Tables

3.1 Acoustical Characteristics of the Chamber ......................................................... 36
3.2 Background Noise Levels for the Chamber at Each 1/3 Octave Band .............. 39
3.3 Acoustic and Structural Resonances with Damping Ratios .......................... 56
4.1 Improvement of IL Using SISO Control ............................................................ 74
4.2 Improvement of IL Using MIMO Control ......................................................... 88
4.3 Improvement of IL Using MISO Control ............................................................ 93
Chapter 1

Introduction

The purpose of this chapter is to give a summary of the essential background material of passive acoustical enclosures which is needed to understand their behavior. The passive insertion loss (IL) of enclosures is defined. Then various drawbacks of acoustical enclosures are presented along with the ways these drawbacks degrade the IL. A general description of active control is given followed by a few examples of industrial applications for this technology. Then a description of previous research related to active control applications for enclosures is presented. The motivation for doing this research, its objectives, and the research approach is stated. Finally, a brief overview of this thesis is presented.

1.1 Passive Acoustical Enclosures

Noise pollution is a common problem in modern society that can lead to physical discomfort and health problems. Methods of noise control may involve reduction of noise at the source, control of noise transmission paths, and protection of the receiver [1]. One way of interrupting noise transmission paths is the implementation of acoustical enclosures. An acoustical enclosure is defined as a rigid, usually airtight, box, closely surrounding a noise source [2]. A properly designed enclosure can significantly reduce the amount of noise radiated by the noise source. Acoustical enclosures provide a means of quieting noise sources for both indoor as well as outdoor applications. Typical uses for enclosures include gas turbines, portable air compressors, electrical transformers, and so forth [3].
The most appropriate way of quantifying the performance of any type of acoustical enclosure is by measuring its passive insertion loss (IL) [4]. This passive IL is defined as the difference between the sound pressure level (SPL) or sound power level (L_w) radiated by the noise source without and with the enclosure [4]. For IL based on SPL, a microphone is used to measure the SPL at a number of locations surrounding the enclosure. The measurement locations may be chosen on a circle that is centered at the source location with a radius at least three times the longest dimension of the enclosure [4]. The average radiated SPL is then computed from these measurements and used to compute the IL. For IL based on L_w, a sound intensity meter is used to measure the L_w with and without the enclosure [4]. The typical IL for acoustical enclosures varies with frequency and has three main frequency regions where different physical aspects of the enclosure affect it [4]. These regions are called the low, intermediate, and high frequency regions. The low frequency region is characterized by a uniform distribution of interior acoustic pressure and the absence of acoustic and structural resonances. The IL in the intermediate frequency region is dominated by acoustic and structural resonances and is characterized by numerous fluctuations [4]. For the high frequency region, interior sound absorption and the transmission loss characteristics of the enclosure walls controls the IL. A more detailed description of these regions will be presented in Chapter 2.

The IL of acoustic enclosures is most effective for systems that are airtight and have rigid walls [2]. Openings have a significant degradation on the noise reduction of passive noise control elements, i.e. walls, enclosures, and so forth. It has been shown that an opening with an area of only 1 percent in a wall with a transmission loss of 50 dB, reduces it to only 20 dB [3]. Another consequence of having an opening in the enclosure is that it can introduce an acoustic resonance that can actually amplify the sound pressure level inside and outside the enclosure [5]. Thus, at this frequency, more sound is radiated with the enclosure than without it. This resonance is analogous to the well known one found in a Helmholtz resonator. Unfortunately, openings limit the applications for passive
enclosures since they cannot be used to reduce noise emitted from devices requiring air gaps for cooling, power cables, and so forth.

Another problem with enclosures is the transmission of vibration to the enclosure structure from the source it is covering [4]. This is especially a problem in a factory environment where machine vibrations can be transmitted to the enclosure through the floor. These transmitted vibrations can cause additional sound to be radiated from the enclosure. This problem can be solved by isolating either the machine or the enclosure from the floor. In some applications, isolation of both systems is desired [4].

1.2 General Description of Active Control

The primary objective of active control is to cancel unwanted responses such as sound or vibrations on a structure at a particular location or over a region [6]. The unwanted response is called the disturbance and the canceling source is called the control. This destructive interference can be accomplished by using one of two types of control approaches: feedback and feedforward techniques [7]. In the feedback control of acoustical systems, the pressure fluctuations detected by a microphone are reduced by amplifying the microphone signal and feeding it back to an acoustic control source, i.e. a loudspeaker. This loudspeaker produces an acoustic signal which is added to the disturbance signal at the detecting microphone [7]. The feedback loop contains an electroacoustic transfer function between the loudspeaker input and resulting microphone output and a transfer function of the electrical network feedback path. These transfer functions shift the signal from the microphone 180 degrees so that the loudspeaker can produce a sound that cancels the disturbance.

The second control approach, feedforward, depends on a reference signal that contains the same frequency components as the disturbance [7]. A control signal is
obtained by feeding this reference signal through a filter, producing a response that cancels the disturbance at a location. The detecting sensor at this location is called the error sensor. The control source produces a response that drives the error signal, represented by the sum of the disturbance and control responses, to zero [7]. These active control systems can be simple, having one control source and error sensor, or they can be complex with multiple control sources and error sensors. Theoretically, the number of control sources must be less than or equal to the number of error sensors to keep the feedforward control system mathematically determinable [7].

Active control has been implemented in many industrial applications. Some examples include the active control of propeller-induced passenger cabin noise and the engine-induced noise in automotive interiors [8]. Rotating or reciprocating machinery such as internal combustion engines that are prevalent in these applications produce periodic sound waves and provide a means of obtaining a reference signal that is unaffected by the control source [7]. This is accomplished by using a sensor that observes the motion of the machine. Active noise control has also been implemented to silence noise in ventilation ducts [9]. This application can be treated as one dimensional, noise cancellation along the axis of the duct [9]. Another application is an active noise control communications headset system [10]. This device, as described by Gauger and Sapiejewski [10], was used on the around-the-world flight of the experimental aircraft Voyager.

1.3 Previous Work Involving Active Control with Enclosures

There has been much research carried out involving active noise control of sound in enclosed cavities [8, 11-19]. While this type of work has practical significance for automobile cabins [15], and aircraft passenger compartments [19], it is marginally related to the research presented here. One of the issues addressed in this research is the
possibility of reducing the level of radiated sound from an enclosure by reducing the noise level in its interior. Analytical performance evaluations have shown that for three-dimensional enclosures, the best applications appear to be when the source is localized and accessible, when the response within the frequency band of interest is dominated by relatively few modes, or when only local control is desired i.e. control over a small or limited region [11]. Research carried out by Elliot and Nelson [8], have demonstrated that the total acoustic potential energy in an enclosure can be reduced using active control methods. Computer simulations were done to calculate the maximum possible reduction of acoustic energy produced by a pure tone source located in one corner of an enclosure using a given set of control sources. The reduction of the interior acoustic energy was also illustrated by minimizing the squares of the acoustic pressure in the four corners of the enclosure. It was shown that both methods of active control proved to be effective at low frequencies where there is a low density of acoustic modes. For high frequencies, global reductions in the acoustic energy can be achieved if the disturbance source is compact and easily accessible [8].

Some experimental research was also carried out with active noise control in a reverberant three-dimensional space [12, 13, 14]. Miyoshi et al. [12] have shown that localized noise reduction can be achieved using speakers and error microphones located inside an enclosed room. By using three control speakers and two error microphones, a reduction of over 6 dB was obtained for an oval-shaped region surrounding the error microphones. The disturbance source used for this experiment was random noise with a bandwidth of 50 to 400 Hz.

Limited research has been carried out involving the active control of the sound radiation from acoustical enclosures [20, 21]. Experiments performed by Chatel et al. [20] have shown that active control is effective at reducing the radiated sound from an engine enclosure. The enclosure used in this work was parallelepiped in shape having
dimensions of 1.60 X 1.20 X 1.50 m. The enclosure walls were constructed of 2 mm thick steel sheet and the interior sides were covered with a 50 mm thick sound absorbing material. A bass-reflex loud-speaker that simulated engine noise was used as the disturbance source and five loud-speakers were used as the control sources. The frequency range 50 to 250 Hz was considered. Two different methods of implementing active control were investigated. The first method used five microphones located inside the enclosure and the SPL was minimized at these places. This control set-up provided a reduction in the radiated sound power level mainly below 150 Hz. The second technique consisted of minimizing the total sound power radiated by the enclosure walls by modifying the vibration characteristics of the enclosure. This method decreased the radiated sound power over the whole frequency range and also provided more reduction than the first technique.

Fuller and White [21] demonstrated that active control methods can also be an effective means of improving the performance of enclosures with air gaps. The enclosure used in this research was a 305 X 406 X 305 mm box constructed of 6.35 mm thick acrylic sheet. Machine screws were used to join together the enclosure panels. Air gaps were created by using metal spacers to separate the top panel and the enclosure was also raised on wooden blocks forming an air space around the bottom. The disturbance source was a 165 mm acoustical speaker mounted in a 184 X 184 X 114 mm reflex speaker box and was excited at a sinusoidal frequency. The speaker was driven at two separate frequencies, 200 and 450 Hz. This source occupied 10% of the inside volume of the enclosure. Five piezoceramic wafer elements (PZTs), mounted at the center of each panel, were used as the control actuators. A three-input, three output control system was implemented in which the control actuators on opposite side panels were driven in phase with each other while the top panel actuator was independent. Three microphones placed 90 degrees apart in a semi-circle at a distance of 1.7 m from the center of the enclosure
were used as error sensors. The acoustic radiation directivity was measured over this semi-circle.

All of the experimental tests were performed in an anechoic chamber which provided a free field acoustic environment. Tests were performed using a 13% and 16% open area of the enclosure surface. At 200 Hz, the enclosure with 13% open area provided no passive IL but active control added 20 dB of IL. The enclosure with 16% open area increased the acoustic radiation by 5 dB at 200 Hz due to internal resonance phenomena. Active control provided reduction of 20 dB in the radiated SPL giving an overall IL of about 15 dB. At 450 Hz, the enclosure with 13% open area had a passive IL of about 15 dB and active control added an additional 20 dB. The enclosure with 16% open area provided 5-7 dB of passive IL and active control added another 5-7 dB giving a total IL of about 15 dB.

1.4 Motivations and Objectives for this Research

Passive acoustical enclosures are a common method of noise control in many industrial applications. Limitations of these enclosures which range from required openings for air-cooled vents or power cable access to noise sources that radiate a large amount of low frequency acoustic energy are difficult to control using only passive control techniques. Active control can improve many of these shortcomings. An enclosure aided by active control has more versatility than a strictly passive enclosure by making it possible for them to be constructed with less material, thus making them light and easy to manipulate.

Previous research related to active control with enclosures were primarily numerical simulations or experimental investigations that studied the reduction of interior noise in a cavity. Chatel et al., [20] have shown that low frequency noise radiation can be
reduced by placing error microphones inside the enclosure and by minimizing the estimated sound power radiation. Other research has shown that active control can reduce the noise radiation from leaky enclosures in free field environments [21].

The research presented here studies the effect of active control on the IL for both sealed and leaky enclosures in a reverberant environment and examining the frequency range where active control proves to be most effective. Ultimately, it is desired to investigate if the active control system can be compact. This means that all of the elements of the control system are located on the surface or in the interior of the enclosure. Another desirable feature of the active control system is for it to be cost effective. A relatively cheap system can be developed using a simple control system (i.e. using few error sensors and control sources). A simple control system also has the attractive feature of having few components that can malfunction, making this control system more robust than a complex system.

1.5 Research Approach

The research performed here is mainly experimental. The approach used for this study is to first consider the effectiveness of passive enclosures and how active control could improve their IL characteristics (i.e. what frequency regions active control could the IL). An analytical study of a passive enclosure, both with and without air gaps, is performed. In addition to this analytical work the passive characteristics of the enclosure are experimentally obtained.

The next step is then to design and perform a series of active control experiments on the enclosure to investigate how well active control improves its IL. The active control methods studied in this work involve reducing the sound pressure level in the interior as well as the exterior of the enclosure. A reverberation chamber is chosen as the test
environment since a large percentage of the applications requiring acoustical enclosures involve at least semi-reverberant environments (i.e. a noisy machine in a factory). The enclosure is designed such that it can be completely sealed and easily modified so it has air gaps for open enclosure experiments. The active control systems used for these experiments are single input/single output (SISO), multiple input/multiple output (MIMO), and multiple input/single output (MISO). All of the SISO experiments use a single microphone located in the interior of the enclosure as the error sensor and either PZTs or a speaker as a control source. The MIMO and MISO experiments use microphones placed in an array surrounding the enclosure as the error sensors and PZT actuators mounted on the enclosure panels as the control sources. One of the MIMO control systems use accelerometers mounted on the enclosure walls as error sensors. After the data is collected, the results of the various control experiments are compared. Finally, the feasibility and applications of the best active control schemes are considered and the conclusions and recommendations are given.

1.6 Brief Overview of the Thesis

Chapter 2 presents a more detailed explanation of the passive acoustics of enclosures. Various paths through which acoustic energy can be transmitted are discussed. Some techniques used to analytically calculate the passive IL in each frequency region for both sealed and leaky enclosures are also presented. Then some passive ways of improving the performance of enclosures are discussed. This is followed by possible uses of active control techniques to make enclosures more effective.

The hardware used in the experimental study and the characteristics of the enclosure are explained in Chapter 3. This chapter begins with a description of the enclosures, disturbance, control sources, error sensors, and the hardware used to produce the control signal. This is followed by a detailed description of the test environment and
its acoustical properties. Then a brief outline of the experimental methodology is presented. Finally, some analytical and experimental results of the physical and acoustical properties of the enclosures are given.

The main purpose of Chapter 4 is to describe each active control experiment and present their test results. This chapter begins with a description of the single channel control experiments and their results are presented. This is followed by a detailed explanation of the different multichannel control experiments along with their results. The results of each experiment are compared to show where they are effective in improving the IL.

Finally, Chapter 5 presents the conclusions based on the test results and gives recommendations for further study. The conclusions explain where active control proved to be effective and any trends in the results are also presented. The active control experiments that worked well are described along with any potential for applications in industry.
Chapter 2

Characteristics of Passive Acoustical Enclosures and Ways Active Control Can Improve Them

This chapter explains the general behavior of passive acoustical enclosures and how they provide the motivation for studying various ways of implementing active control to improve their IL. First, the characteristics affecting the passive IL of an enclosure are presented along with the various paths through which acoustic energy is transmitted through an enclosure. A detailed explanation of the main frequency regions for passive IL is given and analytical equations for predicting the IL for sealed and leaky enclosures are presented. Some passive methods of improving the enclosure performance are then given followed by some possible applications for active control.

2.1 Characteristics of the Passive IL of Sealed and Open Enclosures

In order to design an active control system for an enclosure the passive acoustics of these structures must first be understood. The amount of sound radiated from an enclosure is affected by two components: (1) the direct field from the source and (2) the internal reverberant sound field [22]. The IL of an enclosure is affected by its physical characteristics such as its size, the type of material it is constructed from, the presence of any openings, physical coupling of the disturbance source to the enclosure, and so forth.
Figure 2.1 shows a typical acoustic enclosure and the different ways noise is transmitted through it. As illustrated by this figure, there are three main paths through which sound is transmitted [23]. These paths include the enclosure wall itself; any openings in the enclosure such as ventilation ducts and gaps around rotating shafts; and physical connections from the machine to the enclosure. Since many practical enclosures have openings and are often physically connected to the disturbance source, the effectiveness of an enclosure could be compromised if these factors are not considered in the design process.

2.1.1 Main Frequency Regions

As mentioned in Chapter 1, there are three frequency regions that control the passive IL of an acoustical enclosure. Figure 2.2 shows how a typical IL varies with frequency and illustrates the main characteristics of the three regions. These regions are called respectively the low, intermediate, and high frequency regions.

Low Frequency Region (sealed enclosure)

Region 1, the low frequency region, includes the range where there are no acoustic or structural resonances. Since the frequency range of this region is below frequency of the first enclosure panel resonance, the panel motion is stiffness controlled [23]. In this region the IL is independent of frequency as shown in Figure 2.2. This region is also characterized by uniform acoustic pressure in the interior of the enclosure [4].

If the acoustic wavelength is such that

\[ L_{\text{max}} \leq \frac{1}{10} \lambda \]  

(2.1)
Noise Radiation Path:  
- through the enclosure wall  
- through openings  
- through structure-borne paths

Figure 2.1  Various Paths of Noise Transmission Through an Enclosure  
(*source*: Ver, [23])
Figure 2.2 Passive Insertion Loss (IL) for a Sealed Enclosure  
(source: Beranek and Ver, [4])
where $L_{max}$ is the largest dimension of the enclosure volume, the interior acoustic pressure will be uniformly distributed [4].

In this region, the IL is a function of the ratio of the volume compliance of the enclosed air to the sum of the compliances of the walls [23]. All that is required is knowledge of the physical dimensions of the enclosure and properties of the material from which it is constructed. The IL for a sealed enclosure in the low frequency region is given by [23]

$$\text{IL(dB)} = 20 \log \left( 1 + \frac{C_v}{\sum_{i=1}^{n} C_{pi}} \right),$$  \hspace{1cm} (2.2)

where

$$C_v \left( m^5 / N \right) = \frac{V_o}{\rho_o c^2}$$  \hspace{1cm} (2.3)

is the compliance of the air volume inside the enclosure. The interior air volume is given by $V_o$, $\rho_o$ is the density of air which is 1.21 kg/m$^3$ at 20°C, and $c$ is 343 m/s, the speed of sound in air.

The compliance of the $i$th enclosure panel, $C_{pi}$, is given by [23]

$$C_{pi} \left( m^5 / N \right) = \frac{0.001 A_{pi}^3 F_i(\alpha)}{B_i},$$  \hspace{1cm} (2.4)

where $A_{pi}$ is the surface area of the $i$th panel; $F_i(\alpha)$ is a function of the boundary conditions of a panel and its aspect ratio given by $\alpha = \ell/w$, where $\ell$ and $w$ are respectively the longest
and shortest edge dimensions of the panel; \( B_i \) is the bending stiffness. The function, \( F_i(\alpha) \) is found with an empirical fit for Lyon’s curves [24]. For clamped edges, \( F_i(\alpha) \) is given by [24]

\[
F_i(\alpha) = 3.68\beta^3 - 2.26\beta^2 - 0.336\beta + 0.376, \quad 1 < \alpha < 2.5
\]

\[
\frac{0.743}{\alpha^2}, \quad \alpha > 2.5
\]

where \( \beta = \log_{10}(\alpha) \).

Equation (2.2) shows that to achieve a high IL in the low frequency region requires either a large interior volume compliance \( C_v \), or a small compliance of the enclosure panels, \( C_{ps} \). This means that an enclosure with a large volume \( V_o \), and constructed with a stiff material (large \( B_i \)) will have a high IL in this region. On the other hand, the bandwidth of the low frequency region will decrease as the size of the enclosure increases since the frequencies of the acoustic and structural resonances will decrease.

**Intermediate Frequency Region (sealed enclosure)**

Region 2, the intermediate frequency region, is where the IL is dominated by the resonant behavior of the enclosure structure and the interior acoustic resonances [4]. The main characteristics of this region are the numerous fluctuations in the IL as shown in Figure 2.2. These fluctuations are caused by resonant behavior, both acoustical and structural, of the enclosure system. The minimums in the IL occur at frequencies corresponding to acoustic and/or structural resonances.

The beginning of the intermediate frequency region is characterized by a sharp drop in the IL as shown in Figure 2.2. The lowest order enclosure panel resonance is
associated with a large loss in the IL at the resonance frequency [22]. At the first panel resonance, the walls will exhibit very high volume displacements [23]. The frequency of this first resonance, $f_{11}$, of a panel is given by [23]

$$f_{11} = \frac{c^2}{2A_p f_c} \zeta,$$  

(2.6)

where $f_c$ is the critical frequency of the panel and $\zeta$ is a factor depending on the aspect ratio $\ell/w$ and on the boundary conditions of the panel, namely:

$$\zeta = \frac{1}{2} \left( \frac{\ell}{w} + \frac{w}{\ell} \right) \gamma,$$  

(2.7)

where $\gamma = \begin{cases} 1 & \text{for simply supported edges} \\ < 1 & \text{for clamped edges} \end{cases}$.

When the panels are driven at and above their first resonance frequency the panel compliance cannot be determined by equation (2.4), which is valid only in the stiffness controlled (low frequency) region [23].

Since the IL varies significantly over this region, it is very difficult to make analytical predictions of the IL [4]. A finite-element analysis, model-scale or full-scale experiments, or crude approximations can be used to predict the IL [4]. Although no diffuse sound field exists in this frequency range, it has been found that the assumption of a diffuse sound field usually results in a conservative estimate of the IL [25, 26]. In an ideal diffuse field the average acoustic energy density is the same throughout the volume of the chamber and all directions of propagation are equally probable [27].

**High Frequency Region (sealed enclosure)**
Region 3, the high frequency region, is where the IL is controlled by interior sound absorption and the sound transmission loss of the enclosure panels [4]. The wavelength in this region is much shorter than the enclosure dimensions resulting in an interior sound field that is diffuse [23].

In the high frequency region, the panels and interior acoustic volume exhibit a very large number of resonances (i.e. high modal density) [23]. The dip in the curve of the IL shown in Figure 2.2 corresponds to the coincidence frequency of the panels [4]. The beginning of the coincidence region of determined by the critical frequency of the panels which is given by [27]

\[ f_c = \frac{\sqrt{3}}{\pi} \frac{c^2}{h} \sqrt{\frac{\rho}{E}}, \]  

where \( h \) is the panel thickness, \( \rho \) is the density of the panel, and \( E \) is Young's modulus.

For this region, statistical methods of room acoustics can be used to predict the IL. First, the space-average mean-square pressure of the diffuse sound field within the enclosure is calculated [23]. Once this is computed, the sound power escaping through various paths can be determined [23]. An energy balance is done by equating the sound power of the source with the sound power loss through dissipation, air leaks, and transmission. The IL is then determined by [23]

\[ IL \equiv 10 \log \left( \frac{W_o}{W_{Tw} + W_{TG} + W_{TS} + W_{SB}} \right), \]  

(2.9)
where $W_{TW}$, $W_{TG}$, $W_{TS}$, $W_{SB}$ are respectively the sound power transmitted through the enclosure walls, through air gaps, to the exterior of a silencer, and through structure-borne paths.

### 2.1.2 Openings and Their Detrimental Effects on Insertion Loss

In many industrial applications, it is not practical to use a completely sealed enclosure as a means of noise control. Often there must be openings in the enclosure for disturbance sources requiring cooling, power cable and product line access, and so forth. As mentioned in Chapter 1, openings have a serious detrimental effect on the IL of an enclosure. In addition to having a general decrease in the IL, these openings can also introduce resonances similar to those generated by Helmholtz resonators. As explained in Chapter 1, this resonance can actually amplify the radiated sound. The acoustic volume of the enclosure together with the openings comprise this “Helmholtz resonator”.

If the leaky enclosure is modeled as a simple Helmholtz resonator, the resonance frequency, $\omega_r$, of the resulting cavity-pipe system is given by [23]

$$\omega_r = \sqrt{\frac{A_n}{\rho \cdot h' \left( C_r + \sum_{i=1}^{n} C_{pi} \right)}} \quad ,$$  \hspace{1cm} (2.10)

where $A_n$ and $h'$ are respectively the cross-sectional area and the effective length of the pipe. This length is estimated by [4]

$$h' \equiv h + 1.2a \quad ,$$  \hspace{1cm} (2.11)
where h is the enclosure wall thickness and a is the pipe radius. If r is the resistive impedance of the pipe, the interior acoustic pressure at this resonance frequency reaches its maximum which is higher by the factor $A_o\rho_o h'/r$ than that obtained for the equivalent airtight enclosure [23]. This amplification is due to the small impedance of the system at and around its resonance frequency [23].

The IL of a leaky enclosure in the low frequency range is given by [5]

$$\text{IL(dB)} = 20 \log \left| 1 + \frac{C_v}{\sum_{i=1}^{n} C_{pi} 1 + 1/j\omega \sum_{i=1}^{n} C_{pi} Z_L} \right|,$$

(2.12)

where $C_v$ and $C_{pi}$ are defined in equations (2.3) and (2.4). $Z_L$ is the acoustic impedance of the opening and $\omega$ is the angular frequency (rad/s). For an enclosure with a hole of radius a (in meters) and a is less than $0.002/\sqrt{f}$, the acoustic impedance of the hole is given by [5]

$$Z_L = \frac{8\eta h'}{\pi a^4} + j\omega \frac{4\rho_o h'}{3\pi a^2},$$

(2.13)

where $\eta$ is $1.86 \times 10^{-5}$ kg/m·s, the viscosity coefficient of air at 20°C and 760 mm Hg, and f is the frequency (Hz). If the hole radius a is such that $0.01\sqrt{f} < a < 10/f$, then the acoustic impedance is [5]

$$Z_L = \frac{\rho_o}{\pi r^2} \left[ \sqrt{2\omega \mu (2 + h/a)} + j\omega h' \right],$$

(2.14)
where $\mu$ is $1.56 \times 10^{-5}$ m$^2$/s the kinematic viscosity coefficient of air at 20° C and 760 mm Hg.

For a narrow slit having a width of $s$ and height of $d$ that is cut in one wall the acoustic impedance is [5]

$$Z_L = \frac{6\rho_o}{ds} \left[ \frac{2\mu \left( h + 1.2 \sqrt{ds/\pi} \right)}{d^2} + j\omega \frac{h + 1.2 \sqrt{ds/\pi}}{5} \right]. \quad (2.15)$$

For enclosures with very narrow slits, the IL gradually decreases with decreasing frequency instead of having the resonance dip found with round holes [5].

Well below the resonance frequency given by equation (2.10), nearly all the volume change caused by the enclosed source is pushed through the pipe and the IL is near zero [23]. In the vicinity of this resonance the interior acoustic pressure and particle velocity both reach their maximum values and the volume displacement due to the compliance of the panels and the volume displacement at the duct entrance both contribute to the net volume velocity [23]. As a result, the IL approaches a minimum. At and below this resonance, the total sound radiation is determined by the net volume displacement of the stiffness controlled panels and the mass controlled pipe [23].

The IL for an open enclosure given by equation (2.12) has a pronounced maximum at the frequency [23]

$$\omega_v = \frac{A_o}{\sqrt{\rho_v h' \sum_{i=1}^{n} C_{pi}}} \quad (2.16)$$
At this frequency the volume displacements of the enclosure panels and the pipe opening are equal in magnitude but are 180 degrees out of phase, so the net volume displacement is zero and the enclosure radiates as an inefficient dipole [23]. At frequencies well above \( \omega_c \), the IL of the leaky enclosure approaches the IL of a sealed enclosure [23].

2.2 Passive Ways of Improving Enclosure Performance

There are several ways of improving the passive IL of an enclosure. These include choosing a material that has a high transmission loss to construct the enclosure, lining the interior of the enclosure with sound-absorbing treatment, isolating either the enclosure or the disturbance or both from any vibration transmission paths, and so forth [4]. Lining the interior of the enclosure with sound-absorbing treatment improves the IL in the high frequency range and adds damping to interior acoustic resonances. Isolating the enclosure from external vibrations can improve the IL of the enclosure where these disturbances add vibration-induced noise.

Different materials vary in the amount of sound they absorb, reflect, and transmit [1]. Transmission loss is a measure of sound insulation provided by a panel [1]. This quantity is defined as the difference between the sound power (in dB) incident on the panel minus the sound power transmitted through the panel. Transmission loss is affected by the material that constructs the panel, its thickness, and the frequency of incident sound [1]. Since different materials have different transmission losses, the IL of an enclosure is also affected by the type of material constructing it.

As a rule of thumb, the enclosure panels should be selected such that their coincidence frequency is above the frequency region of interest [4]. This is true for most commonly used panels (1-2 mm steel or aluminum). Stiff, lightweight panels (such as honeycombs) generally have low critical frequencies [4]. Consequently, such panels
should not be used to construct an enclosure unless the design requires a high IL only at low frequencies. The panels should also be large enough that their first structural resonance is below the same region of interest but heavy enough to yield a field-incidence mass law transmission loss matching the IL requirements [4].

Treatment can be added to the enclosure walls to improve its performance. Fluctuations in its passive IL due to structural and acoustic resonances can be decreased respectively by applying damping treatment to the walls and internal sound-absorbing lining [4]. Properly choosing the right sound-absorbing treatment is more crucial than the right material and thickness of the enclosure wall since it has a more significant effect on the IL [4]. Sound-absorbing treatment helps increase the IL by (1) reducing reverberant buildup in the enclosure at middle and high frequencies, (2) increasing the transmission loss of the enclosure walls at high frequencies, and (3) covering up some of the unintentional leaks in joints between enclosure panels [4]. The absorption coefficient is the measure of efficiency of sound-absorbing material [28]. This parameter gives the percentage of sound energy absorbed for a given amount of incident sound energy. For example, if 55% of the incident sound energy is absorbed by a material, its absorption coefficient is 0.55. The thickness, as well as the type of material, also affects this coefficient [28]. Whenever feasible, interior sound-absorbing treatment for enclosures should have a thickness that yields an absorption coefficient of 0.8 in the frequency region of interest [4].

Another factor affecting the IL of an acoustical enclosure is the amount of vibration that is transmitted from the noise source to the enclosure [4]. This is degrading to the IL by causing additional sound to be radiated from vibrating surfaces such as enclosure walls [4]. One way of combating this problem is by isolating the enclosure, the noise source, or both from any transmitted vibrations [4]. Vibration isolation is defined as a means of decreasing transmission of vibratory motions or forces from one structure to
another [2]. This isolation is accomplished by installing a flexible element called a vibration isolator between the two structures. There are two types of vibrating-isolating applications: (1) those which prevent transmission of vibration from a machine to its foundation, and (2) those which reduce the transmission of motion of a foundation (or a substructure) to a machine [2]. Both of these applications are used for acoustical enclosures: (1) isolating the machine from transmitting vibrations to its foundation, and (2) isolating the enclosure from any vibrations transmitted to it through the floor [4].

2.3 Possible Ways Active Control can Improve Enclosures

All of the preceding discussions presented passive ways of optimizing the IL of acoustical enclosures. However, passive methods alone have their limitations. For example, the lower limit in frequency that can be effectively absorbed by sound-absorbing material is controlled by the type and thickness of the material. Other factors, including the environment where a noise source is located, necessity for air gaps at the enclosure panels, accessibility to the noise source to perform regular maintenance, and so forth can limit the amount the type of passive treatment. This is where active control methods can help improve the IL of an enclosure.

As previously mentioned, a large enclosure with stiff walls will yield a high IL in the low frequency region. However, a large enclosure may not always be feasible in a practical application due to cost or limitations of space [22]. Another consequence of a large enclosure is that the bandwidth of the low frequency region is small. Using a small enclosure increases the bandwidth of the low frequency region but decreases the IL in this region. Active control can be used for small enclosures to increase their IL in the low frequency region. This application is especially attractive since active control is generally more effective at low frequencies.
Another degradation to the IL is the presence of leaks or openings in the enclosure. As previously mentioned, openings are necessary for many disturbance sources to provide ventilation. Active control can help improve the performance of these types of enclosures by compensating for excess noise that leaks through these openings.

The vibrations transmitted to the enclosure are also detrimental to the IL. An alternative to using vibration isolators to control these transmitted disturbances is active control. The purpose of active control is to cancel unwanted disturbances whether they are acoustic or structural in nature. Therefore, active control can provide a means of reducing vibration by canceling unwanted vibrations on the enclosure panels. Active vibration control is not studied with much detail in this research since many other researchers have investigated this topic.
Chapter 3

Experimental Set-up

In this chapter, the hardware used for the experimental tests and the physical and acoustical characteristics of the test environment and enclosures are explained. First, a description of the enclosure, disturbance source, control actuators, and error sensors is presented. This is followed by a brief explanation of the control algorithm and computer hardware used to calculate a control signal. The test environment and its acoustical characteristics are then described. Some analytical predictions of the physical and acoustical behavior of the sealed and open enclosures are presented. Then an experimental approach to measuring these characteristics is described followed by their results. These characteristics include the resonance frequencies and damping ratios for the panels and interior acoustic volume and the passive IL for the sealed and open enclosures. The experimental results are then explained and compared with the analytical predictions.

3.1 General Description of Experimental Test Set-up

Enclosure

The enclosure used for the active control experiments is made of 6.4 mm thick Plexiglas and is boxed-shaped with inside dimensions of 406.4 X 304.8 X 304.8 mm. It is constructed with five panels joined together with machine screws. Figure 3.1 shows a diagram of this enclosure without any air gaps. For experiments involving a leaky enclosure, metal spacers are used to raise up the top panel, creating a 6.4 mm gap as
Figure 3.1 Diagram of the Sealed Enclosure
shown in Figure 3.2. The enclosure is sealed at the bottom by gluing it to a plywood base using clear silicone sealant.

**Disturbance Source**

It is desired to use a disturbance source that can be modeled as a monopole located at the bottom center of the enclosure. The reason for selecting this type of source is because of its simplicity. A Model ID60 heavy-duty, high-output compression driver made by University Sound is used as this source. This acoustic driver has a frequency response of 250-4,000 Hz ±5 dB. As shown in Figure 3.3, the throat of the driver is threaded and has a diameter of 31.8 mm. Taking advantage of this, a 31.8 mm hole is drilled at the center of the bottom of the plywood base through which the driver is screwed into and facing upward.

**Control Actuators**

For most of the experiments, piezoceramic wafer elements (PZTs) are used as the control actuators. A PZT is a material that changes length when an electric potential is applied [29]. This change in length is proportional to the magnitude of the applied voltage. Flat, rectangular-shaped PZTs are mounted at the center of each enclosure panel to achieve maximum deflection of the panels which efficiently produces the most sound especially at low frequencies. Figure 3.4 shows a fold-out view of the enclosure along with the location of the PZT actuators. Numbers, which are used to label each panel, are also shown in this figure. Later in this chapter the panels will be referred to by their numbers. Two piezoceramic wafer elements are mounted on each side of a panel and excited out of phase with sinusoidal control signals as shown in Figure 3.5. This excitation causes a PZT to be stretched while the corresponding PZT on the opposite side of the panel is compressed which produces pure bending moments acting on the panel.
Figure 3.2 Diagram of the Open Enclosure
Figure 3.3 Acoustic Driver Used for the Disturbance Source
Figure 3.4 Fold-out View of Enclosure Showing Panel Numbers and Location of PZT Actuators
Figure 3.5 PZT Excitation Used to Produce Pure Bending Moments
(source: Banks et al., [30])
A speaker similar to the disturbance speaker is also used as a control source in some of the experiments. This acoustic driver, made by Atlas Sound, also has a threaded throat with a diameter of about 25.4 mm. The small size of this source allows it to be modeled as a monopole.

**Error Sensors**

Two types of error sensors, microphones and accelerometers, are used for the active control experiments. For the single channel control cases, one 6.4 mm diameter Realistic microphone is mounted inside the enclosure. This microphone has a frequency response of 50 - 15,000 Hz. The multiple channel control cases use five 12.7 mm diameter Bruel and Kjaer, (B&K) microphones as error sensors which are placed in various locations around the enclosure. Typical frequency responses for these microphones are flat within ±2 dB below 20 kHz or 10 kHz. Any of these microphones are adequate since the frequency range studied in these experiments does not exceed 1,600 Hz. One of the multiple input/multiple output experiments involve mounting five accelerometers directly to the outside of the enclosure panels.

**Control Algorithm and Computing Hardware**

The control algorithm used to produce the control signal is the feedforward filtered-x LMS algorithm [7]. The filtered-x LMS algorithm minimizes the sum of the square of the error signals. A detailed explanation of this algorithm for a single input and single output is found in Appendix A. This control algorithm is implemented on a Texas Instruments TMS320C30 digital signal processing (DSP) board equipped with expansion boards to allow multiple channel control. The boards are housed in an IBM 80386 PC host computer. The codes used to operate the DSP are written in “C” language with
embedded "ASSEMBLY" code instructions. A user interface code allows the user to operate the controller.

3.2 General Description of Test Environment

A reverberation chamber was selected as the test environment for the active control experiments. As mentioned in Chapter 1, a large number of applications involving acoustical enclosures are located in at least semi-reverberant environments. Therefore, it is necessary to give a description of the type of acoustic field that is found in a reverberation chamber.

In general, a noise source in a reverberation chamber produces an acoustic field that has two components [1]. These two components are called the direct and reverberant sound fields. The direct field is the region where progressive sound waves coming directly from the source dominate the acoustic field. In the reverberant field, reflected sound from walls and other acoustically hard surfaces dominate. The total sound field produced by a source has a mean squared pressure amplitude, \( (p_{\text{rms}})^2_{\text{tot}} \), given by [1]

\[
[p_{\text{rms}}(r, \theta, \phi)]_{\text{tot}}^2 = (p_{\text{rms}})_{d}^2 + (p_{\text{rms}})_{r}^2 = \rho_o c W \left( \frac{Q(\theta, \phi)}{4 \pi r^2} + \frac{4}{R} \right), \tag{3.1}
\]

where \( (p_{\text{rms}})_{d}^2 \) and \( (p_{\text{rms}})_{r}^2 \) are respectively the mean square pressures of the direct field and reverberant field; \( W \) is the source acoustic power in watts; \( \rho_o \) is the density of air; \( c \) is the speed of sound in air; \( Q(\theta, \phi) \) is the directivity factor of the source; \( r \) is the radial distance from the source, and \( R \) is the room constant of the chamber.

The direct and reverberant sound fields are considered "separated" by a boundary whose size is determined from the radius of reverberation [27]. This radius is determined
from the ratio of the intensity of the reverberant field to the intensity of the direct field when this ratio is equal to 1. Within this radius, the direct field is considered the dominant component while beyond this distance the reverberant field dominates. This radius of reverberation is estimated for a point source by [27]

\[ R_o = \left( \frac{R}{16\pi} \right)^{1/2} \approx 0.1 \left( \frac{V}{T_{60}\pi} \right)^{1/2}, \quad (3.2) \]

where \( V \) is the volume of the room and \( T_{60} \) is the reverberation time of the chamber. At \( 2R_o \) the direct field contribution to the total sound field is only 1 dB higher than that of the reverberant field alone, while at \( 0.5R_o \) the reverberant field contribution is only 1 dB higher than that of the direct field alone [31].

The reverberant field can be divided into two cases: diffuse and non-diffuse depending on the frequency of the sound and the acoustic characteristics of the reverberation chamber [31]. These two cases are separated by the Schroeder cutoff frequency. Above this frequency the sound field is theoretically diffuse. The Schroeder cutoff frequency, \( f_{Sch} \), is given by [32]

\[ f_{Sch} = \left( \frac{c^3}{4 \ln 10} \right)^{1/2} \left( \frac{T_{60}}{V} \right)^{1/2} = 2093 \left( \frac{T_{60}}{V} \right)^{1/2}. \quad (3.3) \]

As stated in Chapter 2, an ideal diffuse field is characterized by uniform average acoustic energy density throughout the volume of the chamber and all directions of propagation are equally probable [27]. At frequencies below the Schroeder cutoff, a standing wave pattern (normal acoustic modes) exists in rooms having any rectangular arrangement of large flat walls [33].
A schematic of the reverberation chamber and its physical dimensions are shown in Figure 3.6. The dimensions of this chamber yield a volume of 47.76 m³ [31]. The reverberation time was measured at the four corners and at the center of the room [31]. Figure 3.6 also shows the location of the enclosure and the traverse microphone used to measure the SPL. Some research involving the use of reverberation chambers require a two-chamber set-up. Only one of the chambers is used for these experiments. Table 3.1 shows the reverberation times, radius of reverberation, and Schroeder cutoff frequency for this chamber at the one-third octave bands from 80 Hz to 2,000 Hz. The times shown in the table are an average of the reverberation times measured at five locations in the chamber as shown in Figure 3.6.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Reverb. Time, $T_{60}$ (s)</th>
<th>Radius of Reverb., $R_0$ (m)</th>
<th>Schroeder cutoff freq., $f_{sch}$ (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>6.24</td>
<td>0.16</td>
<td>757</td>
</tr>
<tr>
<td>100</td>
<td>3.96</td>
<td>0.20</td>
<td>603</td>
</tr>
<tr>
<td>125</td>
<td>6.45</td>
<td>0.15</td>
<td>769</td>
</tr>
<tr>
<td>160</td>
<td>7.59</td>
<td>0.14</td>
<td>834</td>
</tr>
<tr>
<td>200</td>
<td>7.80</td>
<td>0.14</td>
<td>846</td>
</tr>
<tr>
<td>250</td>
<td>6.85</td>
<td>0.15</td>
<td>792</td>
</tr>
<tr>
<td>315</td>
<td>6.71</td>
<td>0.15</td>
<td>785</td>
</tr>
<tr>
<td>400</td>
<td>6.45</td>
<td>0.15</td>
<td>769</td>
</tr>
<tr>
<td>500</td>
<td>6.12</td>
<td>0.16</td>
<td>749</td>
</tr>
<tr>
<td>630</td>
<td>5.37</td>
<td>0.17</td>
<td>702</td>
</tr>
<tr>
<td>800</td>
<td>4.56</td>
<td>0.18</td>
<td>647</td>
</tr>
<tr>
<td>1000</td>
<td>4.27</td>
<td>0.19</td>
<td>626</td>
</tr>
<tr>
<td>1250</td>
<td>4.48</td>
<td>0.18</td>
<td>641</td>
</tr>
<tr>
<td>1600</td>
<td>4.19</td>
<td>0.19</td>
<td>620</td>
</tr>
<tr>
<td>2000</td>
<td>3.76</td>
<td>0.20</td>
<td>587</td>
</tr>
</tbody>
</table>

Equation (3.3) shows that the Schroeder cutoff frequency is a function of the reverberation time. Since the reverberation time varies with frequency the Schroeder
Height of the chambers: 4.32 m

1, 2, 3, 4, 5 - reverberation time measurement locations

Figure 3.6 Reverberation Chamber and Experimental Set-up
cutoff also varies with frequency as illustrated by Table 3.1. An estimate of the frequency ranges of the diffuse and non-diffuse fields are obtained as follows: the Schroeder cutoff frequencies are calculated using the reverberation time corresponding to the center frequency of each one-third octave band. Each Schroeder cutoff is compared to the center frequency of each corresponding one-third octave band. If the Schroeder cutoff frequency is greater than this frequency, it is assumed that the sound field is non-diffuse. Likewise, a diffuse field is assumed if the Schroeder cutoff frequency is less than the one-third octave band center frequency. Table 3.1 shows that the one-third octave bands 80 to 630 are non-diffuse and 800 to 2,000 Hz are diffuse.

For a diffuse field, the sound power level $L_w$, in the reverberation chamber can be determined as follows: [1]

$$L_w(\text{dB}) = L_{w0}(\text{dB}) - 10 \log(T_{w0}) + 10 \log(V) + 10 \log\left(1 + \frac{S\lambda}{8V}\right) + 10 \log\left(\frac{B}{1000}\right) - 14 \text{ dB}.$$  \hspace{1cm} (3.4)

where $L_{w0}$ is the measured space-energy average sound pressure level in the chamber. The variable $S$ is the total surface area of the reverberation chamber including any diffusers, $\lambda$ is the wavelength in meters at the center of the frequency band, and $B$ is the barometric pressure (mbar). Since the frequency of the disturbance for a large part of each experiment is driven below this Shroeder cutoff, the sound field in the chamber for these frequencies cannot be assumed diffuse. This non-diffuse nature of the reverberant field and the fact that equation (3.4) assumes a diffuse field are the reasons for presenting final results of the sound pressure levels instead of the sound power levels.

One of the concerns of performing acoustical experiments is background noise. The background noise is defined as the amount of noise produced by the cumulative effect
of all the other sources that are not the source(s) of interest [1]. Table 3.2 shows the background noise level for the frequencies used in the active control experiments.

Table 3.2: Background Noise Levels For The Chamber At Each 1/3 Octave Band

<table>
<thead>
<tr>
<th>1/3 Octave Band Center Frequency</th>
<th>Background Noise Level</th>
</tr>
</thead>
<tbody>
<tr>
<td>125 Hz</td>
<td>37.0 dB</td>
</tr>
<tr>
<td>160 Hz</td>
<td>13.6 dB</td>
</tr>
<tr>
<td>200 Hz</td>
<td>15.8 dB</td>
</tr>
<tr>
<td>250 Hz</td>
<td>14.1 dB</td>
</tr>
<tr>
<td>315 Hz</td>
<td>8.7 dB</td>
</tr>
<tr>
<td>400 Hz</td>
<td>7.0 dB</td>
</tr>
<tr>
<td>500 Hz</td>
<td>4.8 dB</td>
</tr>
<tr>
<td>630 Hz</td>
<td>7.2 dB</td>
</tr>
<tr>
<td>800 Hz</td>
<td>-0.1 dB</td>
</tr>
<tr>
<td>1000 Hz</td>
<td>-4.6 dB</td>
</tr>
<tr>
<td>1250 Hz</td>
<td>-4.9 dB</td>
</tr>
<tr>
<td>1600 Hz</td>
<td>-3.8 dB</td>
</tr>
</tbody>
</table>

To measure the background noise of the chamber, the traverse microphone is used to measure the SPL at six different equally spaced locations along the traverse path with the disturbance source turned off and the chamber door tightly shut. The average SPL is computed from the six measurements as follows [1],

\[
SPL_{avg} = 10 \log \left( \frac{1}{N} \sum_{i=1}^{N} 10^{\frac{SPL(i)}{10}} \right) \quad (3.5)
\]

where \( N=6 \) is the number of measurements. The average SPL of the background noise is calculated for each frequency that is used in the experiments.
3.3 Physical and Acoustical Characteristics of the Enclosure

3.3.1 Analytical Study of the Passive Characteristics of the Enclosure

Some of the analytical equations presented in Chapter 2 are used to predict the acoustical behavior of the sealed and open enclosures. The purpose of this study is to estimate the boundaries of the three frequency regions of the passive IL and obtain an understanding of the physics of the enclosures. It is assumed that the panels of the sealed and open enclosures have clamped boundary conditions. The estimated characteristics of the enclosures include: the extent of the frequency region where uniform interior acoustic pressure occurs, first panel resonance frequencies, critical frequency, and Helmholtz frequency of the open enclosure. A computer-generated plot of the IL is also included. A step by step procedure of the calculations for these parameters is given in Appendix B.

The extent of the low frequency region for the sealed enclosure is approximated using the first resonance of the panels and the condition of uniform interior acoustic pressure given by equation (2.1). The wavelength of the highest frequency that gives a uniform distribution of interior sound pressure is about 6m which gives a frequency of 57 Hz. The first panel resonances of the two small panels and three large panels of the enclosure are estimated using equation (2.6). This equation, however, does not include the coupling effects of the five joined panels. For the small panels, the first resonance is about 90 Hz and about 70 Hz for the large panels. Based on these analytical calculations, the range of the low frequency region for the sealed enclosure is 0 to 70 Hz.

Equations (2.2) and (2.12) are used to compute the IL for the sealed and open enclosures respectively. Figure 3.7 shows a plot of the IL for these enclosures over the frequency range 0 to 500 Hz. This frequency range is chosen to observe and compare the behavior of the IL for the sealed and open enclosures. According to this plot, the
Figure 3.7 Computer-Generated Plot of the IL for the Sealed and Open Enclosures
Helmholtz resonance frequency, indicated by the sharp drop in the IL, is about 115 Hz and the frequency where the maximum IL occurs (given by equation (2.18)) is about 300 Hz. For higher frequencies the IL for the open enclosure approaches the IL of the sealed enclosure as shown in Figure 3.7. The IL for the sealed enclosure is constant over the entire frequency range which is shown in equation (2.2). Since these analytical predictions are valid only for uniform interior acoustic pressure, they do not include the effects of acoustic and structural resonances and the transmission loss characteristics of the panels at high frequencies.

Equation (2.8) is used to calculate the critical frequency of the panels which determines where the coincidence effect occurs. This frequency is used as an estimate of the boundary of the high frequency region. The coincidence effect occurs when the wavelength of the bending wave in a panel coincides with the length of an incident sound wave at the angle at which it strikes the panel [1]. This effect reduces the IL of the enclosure. The critical frequency of the panels is about 6,480 Hz. Therefore, the boundary of the high frequency region is at least as high as 6.0 kHz.

3.3.2 Experimental Analysis of the Passive Characteristics of the Enclosure

The physical and acoustical characteristics of the sealed and open enclosures are also obtained experimentally. These characteristics include the acoustic and structural resonances of the enclosures and their passive IL. As mentioned in Chapter 1, the middle frequency region is where alternating maximums and minimums in the IL are caused by acoustic and/or structural resonances [4]. To obtain these resonances, frequency response functions (FRFs) of each panel and of the acoustic interior of the enclosures are measured. The disturbance speaker is driven with white noise having a bandwidth of 0 to 1,600 Hz. Structural resonances are obtained from the output signals of accelerometers placed in the
center of each panel. A microphone placed in a top corner of the enclosure is used for the acoustic resonances of the sealed enclosure as shown in Figure 3.8. For the acoustic resonances of the open enclosure, a microphone is placed right at one of the openings as shown in Figure 3.9. The ratio of the autospectrums between the input signal sent to the speaker and the output signals from these sensors form the FRFs. The acoustic and structural resonances for the enclosure system can be observed from these FRF plots.

Figures 3.10 to 3.19 show the FRFs of each panel for both the sealed and open enclosure. These figures indicate the frequencies where the resonance peaks are located which is useful for examining the physical nature of the enclosure system. The acoustic FRF is also plotted on these graphs so that the structural and acoustic resonances can be easily compared to observe how they interact with each other.

The first two structural resonances for the sealed enclosure occur at 62 Hz and 114 Hz. These frequencies are also about equal to the resonances calculated analytically (70 Hz and 90 Hz). Generally, the acoustic resonance frequencies for the open enclosure are slightly higher than the resonance frequencies for the sealed enclosure. This is caused by the change of the total acoustic impedance due to the presence of the openings.

Since panels 1 and 5 of the sealed enclosure have the same size and boundary conditions, their FRFs have nearly the same shape as illustrated by Figures 3.10 and 3.14. Therefore, they have the same resonance frequencies. This phenomenon is also true for panels 2 and 4 as shown in Figures 3.11 and 3.13. Comparing Figures 3.10 to 3.14 with Figures 3.15 to 3.19 shows that the magnitude of the first two structural resonances of each panel of the open enclosure are much higher than the magnitude of the first two resonances for the sealed enclosure. This along with the Helmholtz resonance are contributing factors to the degradation of the IL for the open enclosure at low frequencies.
Figure 3.8  Microphone Location for the Acoustic FRF of the Sealed Enclosure
Figure 3.9  Microphone Location for the Acoustic FRF of the Open Enclosure
Figure 3.10 FRF Plots of Panel 1 and the Acoustic Volume for the Sealed Enclosure
Figure 3.11 FRF Plots of Panel 2 and the Acoustic Volume for the Sealed Enclosure
Figure 3.12 FRF Plots of Panel 3 and the Acoustic Volume for the Sealed Enclosure
Figure 3.13  FRF Plots of Panel 4 and the Acoustic Volume for the Sealed Enclosure
Figure 3.14 FRF Plots of Panel 5 and the Acoustic Volume for the Sealed Enclosure
Figure 3.15 FRF Plots of Panel 1 and the Acoustic Volume for the Open Enclosure
Figure 3.16 FRF Plots of Panel 2 and the Acoustic Volume for the Open Enclosure
Figure 3.17 FRF Plots of Panel 3 and the Acoustic Volume for the Open Enclosure
Figure 3.18 FRF Plots of Panel 4 and the Acoustic Volume for the Open Enclosure
Figure 3.19 FRF Plots of Panel 5 and the Acoustic Volume for the Open Enclosure
Table 3.3 presents the acoustic and panel resonance frequencies for the enclosures along with their damping ratios. These values are obtained from the data used to produce the plots in Figures 3.10 to 3.19. The frequencies where structural and acoustic resonances coincide are indicated with an asterisk.

Table 3.3: Acoustic and Structural Resonances with Damping Ratios

<table>
<thead>
<tr>
<th>Sealed Enclosure</th>
<th></th>
<th>Open Enclosure</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Acoustic</td>
<td>Structural</td>
<td>Acoustic</td>
</tr>
<tr>
<td></td>
<td>Freq. (Hz)</td>
<td>Damp. Ratio</td>
<td>Freq. (Hz)</td>
</tr>
<tr>
<td>570</td>
<td>0.0123</td>
<td>62</td>
<td>0.0839</td>
</tr>
<tr>
<td>* 858</td>
<td>0.0035</td>
<td>114</td>
<td>0.0588</td>
</tr>
<tr>
<td>* 1032</td>
<td>0.0078</td>
<td>170</td>
<td>0.0418</td>
</tr>
<tr>
<td>* 1144</td>
<td>0.0017</td>
<td>210</td>
<td>0.0498</td>
</tr>
<tr>
<td>* 1278</td>
<td>0.0050</td>
<td>478</td>
<td>0.0296</td>
</tr>
<tr>
<td>1540</td>
<td>0.0065</td>
<td>* 870</td>
<td>0.0103</td>
</tr>
<tr>
<td></td>
<td>* 1032</td>
<td>0.0159</td>
<td>* 1564</td>
</tr>
<tr>
<td></td>
<td>* 1156</td>
<td>0.0043</td>
<td></td>
</tr>
<tr>
<td></td>
<td>* 1278</td>
<td>0.0166</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1368</td>
<td>0.0132</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1438</td>
<td>0.0062</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1572</td>
<td>0.0088</td>
<td></td>
</tr>
</tbody>
</table>

* - resonances where structural and acoustic resonances coincide

The damping ratios presented in Table 3.3 are estimated using the half-power method [34]

\[
\frac{1}{2\xi} = \frac{\omega_n}{\omega_2 - \omega_1},
\]

(3.6)
where $\omega_n$ is the natural frequency and $\omega_l$ and $\omega_h$ are respectively the lower and higher frequencies that are half the power or 3 dB lower than the magnitude at the resonance frequency. The variable $\xi$ is the damping ratio.

The passive IL for the sealed and open enclosures over the frequency range 0 to 1,600 Hz is obtained experimentally by simply measuring the amount of noise radiated by the disturbance source with and without the enclosure. White noise having a bandwidth of 0 to 1,600 Hz is used to drive the disturbance. Spectrums of SPLs are then measured at six different locations along the traverse path in the reverberation chamber. These six spectrums are then averaged to obtain an overall noise level. The IL is obtained by subtracting the spectrum of the overall noise level with the enclosure from the spectrum without the enclosure.

Figure 3.20 shows plots of the passive IL for both the sealed and open enclosures. These plots show that the IL has a high density of fluctuations over the entire frequency range. Many of these fluctuations, which are more pronounced at low frequencies, are due to the variations of the SPL in the chamber and measuring at only six locations in this acoustic field. Figures 3.10 to 3.14 show that the magnitude of structural resonances below 170 Hz are 10 to 20 dB lower than the magnitude of the other resonances. This means that these low frequency resonances do not provide a significant amount of IL degradation. By comparing these figures with the dip in the IL for the sealed enclosure near 170 Hz, it is estimated that the range of the low frequency region for this enclosure is 0 to about 170 Hz. The first two minimums in the IL for the open enclosure occur at 134 and 176 Hz. The former minimum is close to the Helmholtz frequency calculated analytically while the latter is about equal to a structural resonance given in Table 3.3. By comparing the frequencies of the sharp drops in the IL above 480 Hz with the frequencies of the acoustic resonances shown in Figures 3.10 to 3.19, it can be seen that they
Figure 3.20 Passive Insertion Loss for the Sealed and Open Enclosures
correspond. This shows that acoustic resonances significantly degrade the passive IL for both the sealed and open enclosures.
Chapter 4

Active Control Experiments and Their Results

The purpose of this chapter is to present the various active control experiments and explain their results. First, an outline of the experimental methodology is given. This description includes a general explanation of the experimental procedure and type of data used to evaluate the control experiments. Next, a detailed description of the different SISO active control experimental configurations is given followed by the results. Then various MIMO control experiments are described along with a presentation of their results. The final set of experiments are then presented. These experiments use MISO control systems. Summaries of the results explaining which of the various control systems are most effective are also given.

4.1 Active Control Experimental Methodology

A series of active control experiments are performed in a reverberation chamber. Three different types of control systems are studied: single input/single output (SISO); multiple input/multiple output (MIMO); and multiple input/single output (MISO). The general procedure for these experiments is to obtain the average SPL in the chamber when the disturbance source is driven without the enclosure, with the enclosure, and when active control is used in conjunction with the enclosure. These average SPLs are used to compute the IL for the passive and hybrid passive/active enclosures. The IL for each enclosure is then plotted at each disturbance source frequency to show the effectiveness of
the passive enclosure and where active control improves its performance. This procedure is used for both a sealed and open enclosure.

Sound pressure levels in the chamber created by the disturbance source are measured with a 25.4 mm diameter B&K microphone placed in a traverse mechanism. This traverse mechanism moves the microphone in a straight line path from the bottom corner of the chamber to the opposite top corner as shown in Figure 3.6. SPL measurements are taken at eleven equally spaced locations along this traverse path. Since the reverberation time varies with frequency, the radius of reverberation also varies with frequency as shown in Table 3.1. Using equation (3.2), the longest radius of reverberation is 0.20 m and occurs at 100 and 2,000 Hz. The traverse microphone passes no closer than a distance of about 0.38 m to the enclosure panels. Therefore, the SPLs are measured in reverberant field.

The disturbance speaker used for experiments is driven with a sinusoidal signal. Since many acoustical applications use one-third octave band center frequencies, a range of these frequencies from 125 Hz to 1,600 Hz is used to drive the speaker. Since its frequency response is 250 - 4,000 Hz ±5 dB, the speaker is not driven below 125 Hz. A signal magnitude with a 2.0 Vrms voltage is used for all disturbance frequencies.

After an experiment is set up and ready to be tested, a system identification (ID) is performed. A system ID is a method of experimentally measuring the transfer functions from each control source to each error sensor. These transfer functions are used by the filtered-x LMS algorithm to compute a control signal as shown in the Appendix A. Once the system ID is completed the disturbance speaker is driven. Then the active control system is turned on and a signal(s) is(are) fed to the control source(s). This signal(s) is(are) adjusted by the controller until the error sensor(s) is(are) minimized. The controller is then “locked” at this setting so it can be quickly toggled between on and off
without having to compute a control signal(s). The SPL(s) is(are) recorded for the error sensor(s) when the active control is both on and off. After the control system is set, the traverse microphone is moved to each measurement location along the traverse path and the SPL is recorded both with and without active control. The average SPL in the chamber is then computed using equation (3.5) for N=11 measurements.

4.2 SISO Control Experiments

The first set of experiments that are performed use single-input, single output (SISO) active control systems. The main purpose of the SISO control experiments is to investigate if any significant improvement in the enclosure IL can be achieved using only one error sensor and one control source. As stated in Chapter 1, it is desirable to develop a simple, robust, and cost effective control system.

There are three types of SISO control configurations that are studied. Each case involves using one error sensor in a fixed location and three different control sources. One of the control sources is constructed using the PZTs mounted on each enclosure panel and driving them in phase with the same control signal. The other two SISO experiments use a speaker placed at two different locations inside the enclosure as control sources. As mentioned in Chapter 1, previous research has shown that noise radiation from an enclosure can be decreased by reducing the interior cavity noise [20]. Therefore, the SISO control systems have an error microphone which is located inside the enclosure at distances of 152.4 mm from the bottom and 101.6 mm from the inside of two panels as indicated in Figure 4.1. This error microphone reduces the SPL inside the enclosure for the purpose of reducing the amount of sound radiated from the enclosure. Each SISO experiment is performed on both the sealed and open enclosures.
Figure 4.1 Location of the Error Microphone
Figure 4.2 shows a schematic of the SISO active control system. A B&K spectrum analyzer is used to provide a signal for the disturbance source and also to obtain the sound pressure levels of the traverse and error microphones. The disturbance signal is fed through a power amplifier (PA) and then onto the disturbance speaker. This signal is also sent as a reference signal to the computer where it is used in the filtered-x LMS control algorithm. An Ithaco analog low-pass filter (LPF) is used to smooth out the control signal obtained from the computer since the signal is sent through a zero-order-hold. This control signal is then fed through the power amplifier before being sent to the control source. The error microphone signal is sent through two Ithaco filters before being fed into the computer. Both a LPF and high-pass filter (HPF) are used to amplify the error signal and eliminate any unwanted noise. The cutoff frequencies of these filters are adjusted to accommodate the change in the frequency of the disturbance source.

**PZTs Used as the Control Source**

The first control source that is studied involves using the PZTs mounted in the center of the enclosure panels. All five panels are excited with a single control signal. Therefore, all five panels vibrate in phase producing a control source that drives the interior acoustic pressure to a minimum. Figure 4.3 shows a photograph of this control set-up.

**Speaker Used as the Control Source, Configuration 1**

The control speaker used in this experiment is placed at two different locations inside the enclosure. Figure 4.4 shows the first location (Configuration 1) that is tested. Due to the small size of the throat, the control speaker can be modeled as a monopole source. This control source is placed inside the enclosure with its throat pointing upward and located at distances of 196.9 mm from the bottom, 88.9 mm from one side, and 82.5
Figure 4.2  Schematic of SISO Control System
Figure 4.3  SISO Control - PZTs Used for the Control Source
Figure 4.4 SISO Control - Control Speaker, Configuration 1
mm from an adjacent side. As illustrated by Figure 4.4, this location places it almost directly across from the error microphone. The distance from the control source to the error microphone is less than the distance from the disturbance to the error microphone which makes up a causal system. This is important because of time delays in calculating a control signal. The control system is not capable of instantaneously producing a control sound. This is also true for a practical application requiring control of broadband random noise.

**Speaker Used as the Control Source, Configuration 2**

The second location of the control speaker (Configuration 2) is shown in Figure 4.5. This configuration puts the control source 82.5 mm directly above the disturbance speaker with the throat pointed in the direction of the error microphone. This location places the control source at a closer distance to the disturbance source than the location used in configuration 1. This configuration is chosen to investigate if a wider frequency range of IL improvement can be achieved. It has been shown that to drive the acoustic pressure of a single monopole disturbance to zero in the far field then a monopole control source must be placed a distance, d, from the disturbance such that [7]

\[
d < \frac{\lambda}{12},
\]

(4.1)

where \(\lambda\) is the wavelength of the disturbance. It can be seen from this relationship that a smaller separation, d, between the sources implies a smaller wavelength and thus a higher frequency range that can be globally reduced. The system is causal since the control source is closer to the error microphone than the disturbance.
Figure 4.5 SISO Control - Control Speaker, Configuration 2
4.3 Results of the SISO Experiments

The IL is computed for the passive enclosure and the hybrid passive/active enclosure from the average SPL in the chamber and relative to the disturbance source. This shows where active control improved the IL. Figures 4.6 to 4.8 show these results for the SISO active control experiments. Each figure shows both the sealed and open enclosures for easy comparison.

The amount of IL improvement over a frequency range is defined as the reduction of the average SPL in the chamber between the passive enclosure and the passive/active enclosure. This improvement is calculated by using equation (3.5) to compute the average SPL of the one-third octave band center frequencies in the range of interest.

Results of Using the PZTs as Control Actuators

Using all five PZTs together as one control source proves to be ineffective for the sealed enclosure as Figure 4.6 illustrates. This means that using PZTs to reduce the interior acoustic pressure does not reduce the noise radiation from the enclosure. In fact, the PZTs increase the amount of noise radiation which decreases the IL. When the active control system is utilized the average SPL in the chamber increases at all frequencies. An improvement in the IL of 3.4 dB over the frequency range 125 to 250 Hz is achieved when this system is applied on the open enclosure. Figure 4.6 shows that this control setup does not yield a significant improvement in the IL for the open enclosure.

Results of Using the Speaker as a Control Actuator

The control speaker in configuration 1 yields an IL improvement of 9 dB from 125 to 200 Hz and 1.8 dB at 1,600 Hz for the sealed enclosure as shown in Figure 4.7. For
Figure 4.6 SISO Control Results - PZTs Used as Control Source
Figure 4.7  SISO Control Results -- Speaker, Config. 1 Used as Control Source
Figure 4.8 SISO Control Results -- Speaker, Config. 2 Used as Control Source
the open enclosure, this configuration improves the IL by 9.7 dB over the frequency range 125 to 250 Hz and 2.4 dB at 1,600 Hz.

The second configuration for the control speaker is more effective than configuration 1 as shown in Figure 4.8. This control speaker location yields an IL improvement of 16.3 dB from 125 to 250 Hz and 4.2 dB from 400 to 630 Hz for the sealed enclosure. The second location of the control speaker improves the IL for the open enclosure by 20.9 dB in the frequency range 125 to 630 Hz.

Summary of All SISO Results

Table 4.1 presents a summary of the experimental results for the SISO active control systems. This table shows the frequency ranges where these control systems are effective and gives the corresponding IL improvement for each range. The IL improvement computed for the entire frequency range (125 to 1,600 Hz) is also presented.

Table 4.1 Improvement of IL Using SISO Control

<table>
<thead>
<tr>
<th>Type of Control Source</th>
<th>sealed or open</th>
<th>Frequency Range (Hz)</th>
<th>Amount of IL Improvement (dB/dBA)</th>
<th>IL Improve. Over Entire Range (dB/dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PZTs</td>
<td>sealed</td>
<td>none</td>
<td>none</td>
<td>-23.1/-23.3</td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 250</td>
<td>4.2/3.8</td>
<td>-14.8/-15.1</td>
</tr>
<tr>
<td>Spkr., Config. 1</td>
<td>sealed</td>
<td>125 to 200</td>
<td>9.0/7.5</td>
<td>0.3/0.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1600</td>
<td>1.8/1.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 250</td>
<td>11.7/10.1</td>
<td>1.5/1.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1600</td>
<td>2.4/2.4</td>
<td></td>
</tr>
<tr>
<td>Spkr., Config. 2</td>
<td>sealed</td>
<td>125 to 250</td>
<td>16.3/17.7</td>
<td>-3.6/-3.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>400 to 630</td>
<td>4.1/4.2</td>
<td></td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 630</td>
<td>17.8/16.4</td>
<td>-0.4/-0.4</td>
</tr>
</tbody>
</table>

74
Table 4.1 shows that the least effective method of control uses the PZTs mounted on the enclosure panels as the control source. This shows that for a SISO control system using an interior error microphone, a control source should also be located inside the enclosure to effectively improve the IL. The most effective case is when the control speaker is located at configuration 2 as shown in Table 4.1. This active control system has the widest frequency range of IL improvement. This wide range is expected since shorter distances between the control and disturbance sources give wider frequency ranges of global reduction as described by the relationship given in equation (4.1). The drawback of this configuration is that the disturbance source must be locally accessible which is often not the case in a practical application. The control speaker at configuration 1 is a more practical location than configuration 2. The frequency range where this configuration improves the IL is, however, more narrow since the distance between the disturbance and control sources is greater. Therefore, if a speaker is used as the control source in a SISO active control system, it should be placed as close as possible to the dominate disturbance source in the enclosure.

In general, active control works better for low frequencies which is illustrated in Table 4.1. Most of the SISO control systems give an IL improvement in the first three or four one-third octave bands (up to 200 or 250 Hz). The results also show that all of the SISO active control systems are more effective for open enclosures. This means that for a given case, there are greater IL improvements for the an open enclosure than the corresponding sealed enclosure. This can be expected intuitively since the passive IL of open enclosures is less than that of sealed enclosures.

### 4.4 MIMO Control Experiments

The next set of active control experiments that are carried out have multiple inputs and multiple outputs (MIMO). These MIMO control systems use five error sensors and
five control sources. The main purpose of performing these MIMO experiments is to compare them with the SISO experiments by investigating if using multiple error sensors and control sources gives a better IL improvement than one error sensor and control source. The disadvantage of this complex control system is that it is economically more costly and not as robust as a simple SISO control system.

Once again, each MIMO control experiment is tested for both the sealed and leaky enclosures. These control experiments are performed using a sinusoidal disturbance source at one-third octave bands from 125 Hz to 1,600 Hz as in the SISO experiments. For each case, the traverse microphone is used to measure the SPL at the same locations in the chamber as is done with the SISO experiments.

The MIMO active control systems use five microphones placed in various locations around the enclosure or five accelerometers placed on each enclosure panel as the error sensors. These error sensor locations are chosen to reduce the amount of noise in the radiation path around the exterior of the enclosure. The five control sources consist of PZTs placed on the enclosure panels which are each driven by separate control signals. The schematic of this control system, shown in Figure 4.9, is basically the same as the schematic for the SISO control with the only difference being that the computer receives five error sensor inputs and has five control outputs. Each error signal is separately filtered and amplified in the same manner as with the SISO control system before being fed into the computer. A six-channel PA is used to amplify the disturbance and the five control signals. Each control signal is filtered with a LPF before being fed through the PA in the same way as with the SISO experiments.

**Configuration Using Accelerometers as Error Sensors**
Figure 4.9  Schematic of MIMO Control System
Experiments using accelerometers as error sensors involve placing one sensor on each enclosure panel. Since the deflections of the panels are greatest at their centers for the most radiating modes, the sensors are mounted near the center of each panel using blue wax. Each accelerometer is placed right next to, but not touching, a PZT. By using accelerometers, the vibration of each panel are minimized at those error sensor locations. Figure 4.10 shows a photograph of this MIMO set-up.

**Configurations Using Microphones as Error Sensors**

Another way of using multiple error sensors is by placing microphones at various locations around the enclosure for the purpose of reducing the SPL at these positions. These microphones form an array that surrounds the enclosure. This array is constructed by mounting the microphones on stands and placing them in front of the center of each panel. Three different distances between each microphone and the center of each panel are employed: 12.7 mm, 50.8 mm, and 177.8 mm. By placing the error microphones at various distances a trend can be observed as to how the IL is affected as the error sensors are moved farther from the panels. The farthest distance (177.8 mm) should give the most IL improvement since a larger area of the vibrating panel contributes to the acoustic field at this distance [35]. Figure 4.11 shows a photograph of this set-up. This photo shows the error microphones at the distances of 12.7 mm away from each panel.

The final MIMO experiment is performed using only the leaky enclosure. The purpose of this experiment is to investigate if the IL can be improved by canceling the noise leakage through the openings. Each microphone is placed 25.4 mm in front of the center of each opening with the top microphone 25.4 mm away from the center of the top panel.
Figure 4.10 MIMO Control Using Accelerometers as the Error Sensors
Figure 4.11 MIMO Control Using Microphones as the Error Sensors
4.5 Results of the MIMO Experiments

The IL for the passive and hybrid enclosures are once again calculated and plotted at each frequency in the same manner as the SISO experiments. Figures 4.12 to 4.16 show these results for the MIMO active control experiments. Each figure shows both the sealed and open enclosures for easy comparison. The amount of IL improvement over a frequency range is also calculated in the same manner as the SISO experiments.

Results of Using Accelerometers as Error Sensors

Figure 4.12 shows the results of the experiments using accelerometers as the error sensors. This active control system improves the IL for the sealed enclosure by 5.3 dB over the frequency range 125 to 200 Hz and 3.8 dB at 800 Hz. For the open enclosure, an IL improvement of 4 dB is achieved from 160 to 250 Hz. This control set-up did not give a large amount of IL improvement. This is expected since the vibration of each panel is reduced at only one point. For complex mode shapes at higher frequencies, strips of PVDF film sensors or many accelerometers need to be used to sense these mode shapes. This further increases the complexity of the MIMO active control system.

Results of Using Microphones as Error Sensors

The first experiment using microphones as the error sensors involves placing them 12.7 mm away from the center of each panel. Figure 4.13 shows the results of this active control system. An IL improvement of 9.6 dB is achieved in the frequency range 125 to 250 Hz for the sealed enclosure. For the open enclosure, an IL improvement of 9.9 dB from 125 to 315 Hz is achieved. An IL improvement of 3 dB is also obtained at 630 Hz for the leaky enclosure.
The error microphones are then moved back to a distance of 50.8 mm away from the panels for the second MIMO experiment. The results of this test, shown in Figure 4.14, give an IL improvement of 8.1 dB over the frequency range 125 to 250 Hz and 3.8 dB at 630 Hz for the sealed enclosure. When this same set-up is tested on the open enclosure, an IL improvement of 14.4 dB from 125 to 250 Hz and 3.1 dB at 630 Hz is achieved.

The third experiment using error microphones involves placing the microphones at distances of 177.8 mm from each panel. This distance places them farther away from the near field effects of the vibrating panels. Figure 4.15 shows the results of this test. An IL improvement of 10.2 dB over the frequency range 125 to 400 Hz and 2.4 dB at 630 Hz is achieved with the sealed enclosure. For the open enclosure, this control system gives an IL improvement of 18 dB from 125 to 315 Hz and 4.8 dB at 630 Hz.

Only the open enclosure is involved with the final MIMO experiment. An error microphone is placed 25.4 mm in front of the center of each opening around the top of the enclosure and 25.4 mm over the center of the top panel. As previously stated, this experiment is performed to investigate if any IL improvement can be obtained by canceling the noise leakage close to the openings. This experiment is the least effective of all the MIMO experiments for improving the IL of the leaky enclosure. Figure 4.16 shows that an IL improvement of 3.4 dB is achieved for only the first two one-third octave bands (125 and 160 Hz) and 1.7 dB is obtained at 630 Hz. At 250 Hz, an IL improvement of 14.2 dB is achieved. This is comparable to the IL improvements achieved with the previous MIMO experiments.

Summary of All MIMO Results
Figure 4.12 MIMO Control Results -- Accelerometers Used as Error Sensors
Figure 4.13 MIMO Control Results -- Error Mics Placed 12.7 mm Away From Panels
Figure 4.14 MIMO Control Results -- Error Mics Placed 50.8 mm Away From Panels
Figure 4.15 MIMO Control Results -- Error Mics Placed 177.8 mm Away From Panels
Figure 4.16 MIMO Control Results -- Err. Mics Placed 25.4 mm Away From Openings
Table 4.2 summarizes the results of the MIMO control experiments. As done in Table 4.1, the frequency ranges where these control systems are effective and the corresponding IL improvement for each range is presented. Once again, the IL improvement computed for the entire frequency range is also presented.

<table>
<thead>
<tr>
<th>Types of Error Sensors</th>
<th>sealed or open</th>
<th>Frequency Range (Hz)</th>
<th>Amount of IL Improvement dB/dBA</th>
<th>IL Improve. Over Entire Range dB/dBA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accelerometers</td>
<td>sealed</td>
<td>125 to 200</td>
<td>5.3/4.8</td>
<td>-0.8/-0.8</td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>800</td>
<td>3.8/3.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>160 to 250</td>
<td>4.0/3.9</td>
<td>-0.1/-0.1</td>
</tr>
<tr>
<td>Error Mics 12.7 mm Away From Panels</td>
<td>sealed</td>
<td>125 to 250</td>
<td>9.6/12.9</td>
<td>-1.6/-1.6</td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 315</td>
<td>9.9/11.4</td>
<td>-4.9/-4.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>3.0/3.0</td>
<td></td>
</tr>
<tr>
<td>Error Mics 50.8 mm Away From Panels</td>
<td>sealed</td>
<td>125 to 250</td>
<td>9.7/12.1</td>
<td>-6.3/-6.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>3.8/3.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 250</td>
<td>14.1/17.6</td>
<td>-1.6/-1.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>3.1/3.1</td>
<td></td>
</tr>
<tr>
<td>Err. Mics 177.8 mm Away From Panels</td>
<td>sealed</td>
<td>125 to 400</td>
<td>12.6/10.3</td>
<td>-1.3/-1.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>2.4/2.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 315</td>
<td>14.6/12.3</td>
<td>-2.7/-2.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>4.8/4.8</td>
<td></td>
</tr>
<tr>
<td>Error Mics 25.4 mm Away From Gaps</td>
<td>open</td>
<td>125 to 160</td>
<td>3.4/3.7</td>
<td>0.7/0.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>250</td>
<td>14.2/14.2</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>1.7/1.7</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4.15 and Table 4.2 shows that the most effective MIMO experiment is when the microphones are placed 177.8 mm away from each panel. Of all the MIMO experiments, this active control system has the widest frequency range of IL improvement.
This shows that the error microphones are more effective at farther distances from the enclosure panels which is expected since a larger area of each vibrating panel contributes to the acoustic pressure sensed by these microphones. The least effective method of MIMO control is when the accelerometers are used as the error sensors. As previously mentioned, only the vibration at a specific point on each panel is reduced. Therefore, a very localized area of the acoustic field produced by the vibrating panels is affected by active control which is about equivalent to a very near-field microphone. The location of the accelerometers can be treated as the closest distance between each panel and each error sensor. As the error sensors (microphones) are moved to farther distances, a greater IL improvement is achieved over a wider frequency range as illustrated in Figures 4.12 to 4.15 and Table 4.2. Placing accelerometers directly on the enclosure panels is, however, more practical than having error sensors at farther distances. Thus, transfer functions between far field microphones and corresponding accelerometers on the panels need to be obtained to implement this type of MIMO control system. The signals from the accelerometers can be fed through these transfer functions to “trick” the control system into receiving signals from far field error microphones.

The MIMO control systems, as with the SISO experiments, are once again effective primarily at low frequencies. Most of the MIMO experiments improve the IL up to 250 Hz. Once again, as with the SISO experiments the MIMO tests give a greater IL improvement for the open enclosure. Comparing Tables 4.1 and 4.2 shows that the most effective SISO experiment (control speaker at configuration 2) gives both a greater IL improvement and wider frequency range than the most effective MIMO experiment (error microphones placed 177.8 mm away from each panel).
4.6 MISO Control Experiments

A series of experiments are performed using various multiple input, single output (MISO) control systems. These MISO control experiments are compared with the SISO experiments. This comparison is done by observing if using multiple error microphones located outside of the enclosure gives a better IL improvement with one control source than using a single interior error microphone. Five microphones are placed 127 mm in front of the center of each enclosure panel similar to the configurations used in the MIMO experiments. There are two types of control sources used in the MISO control experiments: the PZTs driven in phase and a speaker in the same location as configuration 1 in the SISO experiments.

4.7 Results of the MISO Experiments

The IL for the passive and hybrid enclosures are calculated and plotted at each frequency in the same manner as the previous experiments. Figures 4.17 to 4.18 show the results of all the MISO experiments. Once again, both the sealed and open enclosures are shown for easy comparison. The amount of IL improvement over the frequency ranges where active control reduces the average SPL is once again calculated in the same manner as the previous experiments.

Results of Using the PZTs as Control Actuators

Figure 4.17 shows the results of the cases using PZTs as control actuators. The active control improves the IL for the sealed enclosure by 6.5 dB over the frequency range 125 to 160 Hz and 2.9 dB from 250 to 400 Hz. For the open enclosure, an improvement of 10.9 dB is achieved from 125 to 400 Hz.
Figure 4.17 MISO Control Results – PZTs Used as Control Source
Figure 4.18  MISO Control Results -- Speaker, Config. 1 Used as Control Source
Results of Using a Speaker as the Control Source

Figure 4.18 shows the results of the cases using a control speaker in the same location as configuration 1 in the SISO experiment. The active control improves the IL for the sealed enclosure by 4.3 dB over the frequency range 160 to 250 Hz. For the open enclosure, an improvement of 13.0 dB is achieved from 125 to 250 Hz.

Summary of All MISO Results

Active control is effective, as with the previous experiments, at low frequencies. Improvements in the IL up to 400 Hz are achieved with the PZTs as control actuators. The speaker proved to be effective only up to 250 Hz. Once again, as with the SISO and MIMO experiments, the MISO tests are more effective for the open enclosure.

Table 4.3 summarizes the results of the MISO control experiments. As done previously in Tables 4.1 and 4.2, the frequency ranges where these control systems are effective and the corresponding IL improvement for each range is presented. The IL improvement computed for the entire frequency range is again presented.

Table 4.3 Improvement of IL Using MISO Control

<table>
<thead>
<tr>
<th>Type of Control Source</th>
<th>sealed or open</th>
<th>Frequency Range (Hz)</th>
<th>Amount of IL Improvement dB/dBA</th>
<th>IL Improvement Over Full Range dB/dBA</th>
</tr>
</thead>
<tbody>
<tr>
<td>PZTs</td>
<td>sealed</td>
<td>125 to 160</td>
<td>6.5/7.8</td>
<td>-0.6/-0.6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>250 to 400</td>
<td>2.9/2.5</td>
<td></td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 400</td>
<td>10.9/10.4</td>
<td>1.3/1.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>630</td>
<td>3.7/3.7</td>
<td></td>
</tr>
<tr>
<td>Spkr., Config. 1</td>
<td>sealed</td>
<td>160 to 250</td>
<td>4.3/4.4</td>
<td>-0.8/-0.8</td>
</tr>
<tr>
<td></td>
<td>open</td>
<td>125 to 250</td>
<td>13.0/13.0</td>
<td>-2.9/-2.5</td>
</tr>
</tbody>
</table>
If Table 4.3 is compared to Table 4.1, it is shown that the MISO control system produced much better IL improvements when the PZTs are driven in phase and used as a control source. Therefore, when PZTs are used as the control source the error microphones should be located outside the enclosure to reduce the noise radiation from the enclosure. When the speaker located at configuration 1 is used as the control source, the SISO control system gives better IL improvements for the sealed enclosure. This shows it is better to use an interior error microphone to reduce the interior acoustic pressure rather than the noise radiation itself. For the open enclosure, the SISO and MISO control systems have similar results.
Chapter 5

Conclusions and Recommendations

Methods of improving the insertion loss of acoustical enclosures using active noise control has been studied experimentally. Based on the analysis in Chapter 3 of passive enclosures and the experimental observations presented in Chapter 4, the following conclusions and recommendations are made.

5.1 Conclusions

Various active control systems were used to improve the IL of a Plexiglas acoustical enclosure. First, analytical and experimental studies of the passive acoustical-structural system were carried out. These studies helped identify the three different frequency regions that affect the passive IL. Then a series of active control experiments were performed. All of the active control systems were tested primarily in the intermediate frequency region. Single input/single output (SISO), multiple input/multiple output (MIMO), and multiple input/single output (MISO) active control systems using various error sensors and control actuators were investigated. The SISO control systems involved using an error microphone inside the enclosure to reduce the interior sound pressure level. A speaker placed at two separate locations inside the enclosure and piezoceramic wafer elements (PZTs) mounted on the enclosure panels were used as the control sources. The MIMO and MISO control systems used error microphones placed at various distances around the enclosure exterior. One of the MIMO experiments used accelerometers as error sensors which were placed directly on the enclosure walls. Both the MIMO and MISO experiments used the PZTs as the control sources. The speaker
placed in the first location used in the SISO experiments was also used as a control source for one of the MISO experiments.

The key conclusions of this research are:

1. It was shown that active control can improve the IL of both sealed and leaky enclosures in the low frequency region and the beginning of the intermediate frequency region (i.e. low modal density).

2. For the SISO control systems, the greatest IL improvement over the widest frequency range is achieved when the control source is as close to the disturbance source as possible. This approach, however, is often not practical in a real-world application where the disturbance source is complex.

3. The effectiveness of the MIMO control system improves when the error microphones are placed farther from the panels.

4. When the PZTs are used as the control source, microphones located outside the enclosure need to be used as the error sensors to improve the IL. This approach concentrates on placing error sensors in the noise radiation path and reducing the sound pressure level in this region.

5. A compact and cost effective hybrid passive/active acoustical enclosure can be implemented by using a single control speaker and error microphone placed in the enclosure interior.
5.2 Recommendations

In order to expand and improve this area of research, the following tasks are suggested:

1. Investigate applying different active control methods on a more realistic enclosure, i.e. an enclosure made from commonly used materials such as steel.

2. Research possibilities of apply active control on enclosures with more complex disturbance sources. These sources can have complex radiation directivities and emit broadband noise.

3. Develop a hybrid noise control system that combines an optimal passive enclosure with active control to maximize the IL over the low, intermediate, and high frequency ranges.

4. Investigate other types of error sensors such as PVDF strips mounted on the panels to sense modal vibrations or PVDF film lining the interior of an enclosure for the purpose of sensing acoustic pressure on the insides of the panels. The PVDF strips could be used to reduce noise radiation from a modal vibrations point of view. Interior PVDF film lining could be used to reduce the acoustic pressure impinging on the inside of the enclosure walls.

5. Develop a compact active control system that can be used for disturbance sources with complex noise radiation patterns. One possibility is by further investigating using multiple error sensors and control sources that are interior to the enclosure. Another compact system that could be researched is using accelerometers mounted on the enclosure walls as error sensors and obtaining transfer functions between them and far
field microphones. These transfer functions could be used to obtain signals from far field microphones using the accelerometers. This would have the effect of “tricking” the controller.
References


Appendix A

Active Control Algorithm

The control signal is obtained using a filtered-x LMS algorithm. Figure A.1 shows a general block diagram of a SISO active control system. The general principle of this control approach is based on using a reference signal that is coherent with the disturbance signal, $x_k$, and "feeding forward" this reference through a digital filter, $W(z)$. The subscript $k$ indicates a signal sample at time $t_k$. This filter has coefficients that are adjusted by the LMS algorithm in such a way that the error signal, $e_k$, is minimized. From Figure A.1 the error signal is given as [6]

$$e_k = d_k + y_k . \quad (A.1)$$

where $d_k$ and $y_k$ are the output of the system due to the disturbance and control inputs. The transfer function (TF) between the disturbance and the error sensor is $T_{de}(z)$ while $T_{ce}(z)$ is the TF between the control output and the error sensor. Equation (A.1) can be written in terms of the control input by making a substitution,

$$e_k = d_k + T_{ce}(z)u_k . \quad (A.2)$$

The plant consists of that having disturbance and control as inputs and the summation of the two at the error sensor as the output. Since the digital control signal, $u_k$, is obtained by filtering the reference signal through an adaptive filter, a relationship can be written as

$$u_k = W(z)x_k . \quad (A.3)$$
Figure A.1  Block Diagram of Feedforward Adaptive Control
The adaptive FIR digital filter, $W(z)$, is given by

$$
W(z) = W_0 + W_1 z^{-1} + W_2 z^{-2} + \cdots + W_N z^{-N} = \sum_{i=0}^{N} W_i z^{-i}, \quad (A.4)
$$

where the coefficients, $W_i$s, are adapted by the LMS algorithm. Equation (A.2) can then be written as

$$
e_k = d_k + T_{\epsilon \epsilon}(z)W(z)x_k. \quad (A.5)
$$

In order to minimize this error signal, a cost function is defined as [6]

$$
C(W_i) = E[e_k^2], \quad (A.6)
$$

where $E[\bullet]$ denotes the expected value operator. Substituting equations (A.4) and (A.5) into equation (A.7) yields

$$
C(W_i) = E\left[\left( d_k + T_{\epsilon \epsilon}(z)\left( \sum_{i=0}^{N} W_i z^{-i}\right)x_k \right)^2 \right]. \quad (A.7)
$$

This cost function, and thus the error signal, is minimized by a technique known as the steepest decent method [7]. The method involves computing the gradient of the quadratic performance surface and searching along the negative direction toward the minima [6]. The algorithm used to update the filter coefficients is then written [6]

$$
W_i(k+1) = W_i(k) - \mu \frac{\partial C}{\partial W_i}, \quad (A.8)
$$
where the parameter, $\mu$, controls the step size and thus the rate of convergence of the minimization process. This parameter, however, must be carefully selected because the system can go unstable if the convergence parameter is too large. Differentiating the cost function in equation (A.8) with respect to the filter coefficient, $W_i$, gives

$$\frac{\partial C}{\partial W_i} = 2E\left[ e_k \frac{\partial e_k}{\partial W_i} \right] = 2E[e_k T_{ee}(z)x_{k,i}] = 2E[e_k \hat{x}_{k,i}] .$$  \hspace{1cm} (A.9)

The sequence, $\hat{x}_k$, is referred to as the filtered-$x$ signal and is predicted from a model of the control input to error transfer function, $\hat{T}_{ee}(z)$. A system identification is carried out to obtain $\hat{T}_{ee}(z)$ before active control can be applied. This system ID is performed by driving the control source with a signal that is coherent with the disturbance and obtaining a transfer function between this source and the error sensor. Careful location of the error sensor, an accurate system model, and careful selection of a convergence parameter can result in an active control system that produces reduction of noise or vibration in a desired region.

The maximum power reduction of the error signal, $R_{\text{max}}$, which can be calculated only if the reference signal is coherent with the disturbance signal, can be estimated by [36]

$$R_{\text{max}}(\text{dB}) = 10 \log \left[ \frac{1}{1 - \gamma_{xd}^2} \right]$$  \hspace{1cm} (A.10)

where $\gamma_{xd}$ is the coherence between the reference signal, $x_k$, and the disturbance signal, $d_k$. This quantity represents the highest possible reduction of the error sensor and thus is also a measure of the effectiveness the control system.
Appendix B

Calculations of the Physical and Acoustical Characteristics of the Enclosures

The physical properties of the passive acoustical system are presented. A step by step procedure of the calculations of the physical and acoustical characteristics of the sealed and open enclosures is then presented

**Physical Properties of Air and The Enclosure**

for air

density at 20° C: \( \rho_o = 1.21 \text{ kg/m}^3 \)

kinematic viscosity at 20° C, 760 mm Hg: \( \mu = 1.56 \times 10^{-5} \text{ m}^2/\text{s} \)

for Plexiglas enclosure panels

density: \( \rho = 1173.1 \text{ kg/m}^3 \)

Young’s modulus: \( E = 2.9161 \times 10^9 \text{ N/m}^2 \)

Poisson’s ratio: \( \nu = 0.4 \)

panel dimensions: thickness, \( h = 6.35 \text{ mm} \)

length, \( \ell = 304.8 \text{ mm}, \text{ two small panels} \)

\( \ell = 406.4 \text{ mm}, \text{ three large panels} \)

width, \( w = 304.8 \text{ mm}, \text{ all panels} \)
height of openings, \( d = 6.35 \text{ mm} \)

**Highest Frequency That Yields a Uniform Interior Acoustic Pressure**

The condition that determines the highest frequency yielding a uniform interior acoustic pressure is equation (2.1).

\[
L_{\text{max}} \leq \frac{1}{10} \lambda
\]  

(B.1)

First, the largest enclosure dimension of enclosure volume, \( L_{\text{max}} \), is calculated.

\[
L_{\text{max}} = \sqrt{2(.3048m)^2 + (.4064m)^2} \equiv 0.6m
\]

dependence, \( \lambda = 10 \cdot L_{\text{max}} = 6m \)

The condition that produces a uniform interior acoustic pressure is then:

\[
f \leq \frac{c}{\lambda} = \frac{343 \text{ m/s}}{6m} = 57 \text{ Hz}
\]

(B.2)

**Critical Frequency of the Panels, \( f_c \)**

The critical frequency, \( f_c \), of the panels is calculated using equation (2.8).

\[
f_c = \frac{\sqrt{3} c^2}{\pi t} \left( \sqrt{\frac{\rho}{E}} \right)
\]

(B.3)

\[
f_c = \frac{\sqrt{3}(343 \text{ m/s})^2}{\pi (.00635m)} \left( \sqrt{\frac{1173.1 \text{ kg/m}^3}{2.1961 \times 10^9 \text{ N/m}^2}} \right) \equiv 6480 \text{ Hz}
\]
First Panel Resonance, $f_{11}$

Using equation (2.6), the first panel resonance frequencies for the small and large panels are computed.

$$f_{11} = \frac{c^2}{2A_{pi} f_c} \beta$$  

(B.4)

$f_{11}$ for the small panels

area: $A_{pi} = a \cdot b = (0.3048 \text{ m}) \cdot (0.3048 \text{ m}) = 9.2903 \times 10^{-2} \text{ m}^2$

$\beta$: $\beta = \frac{1}{2} \left( \frac{a + b}{b + a} \right) \gamma$, let $\gamma = 0.9$

Since $a=b$ for the small panels, $\beta = \gamma \approx 0.9$.

Therefore, $f_{11}$ is:

$$f_{11} = \frac{(343 \text{ m/s})^2}{2(9.2903 \times 10^{-2} \text{ m}^2)(6480 \text{ Hz})} 0.9 = 90 \text{ Hz}$$

$f_{11}$ for the large panels

area: $A_{pi} = a \cdot b = (0.3048 \text{ m}) \cdot (0.4064 \text{ m}) = 0.12387 \text{ m}^2$

$\beta$: let $\gamma = 0.9$

Then $\beta = \frac{1}{2} \left( \frac{0.3048 \text{ m}}{0.4064 \text{ m}} + \frac{0.4064 \text{ m}}{0.3048 \text{ m}} \right) 0.9 \approx 0.9375$

Therefore, $f_{11}$ is:

$$f_{11} = \frac{(343 \text{ m/s})^2}{2(0.12387 \text{ m}^2)(6480 \text{ Hz})} 0.9375 = 70 \text{ Hz}$$

IL for the Sealed Enclosure

First the compliances of the interior air volume of the sealed enclosure and panels are calculated.
interior air volume: \[ C_v = \frac{V_o}{\rho c^2} \] (B.5)

\[ C_v = \frac{V_o}{\rho c^2} = \frac{3.7756 \times 10^2 \text{ m}^3}{1.21 \text{ kg/m}^3 \cdot (343 \text{ m/s})^2} = 2.6522 \times 10^{-7} \text{ m}^6/\text{N} \]

To compute the total compliance of the panels, the bending stiffness \( B \) and the function \( F_i(\alpha) \), must first be calculated.

bending stiffness: \[ B = \frac{Eh^3}{12(1 - v^2)} \] (B.6)

\[ B = \frac{(2.9161 \times 10^8 \text{ N/m}^2)(0.00635 \text{ m})^3}{12(1 - 0.4^2)} = 74.07 \text{ N.m} \]

Assuming the enclosure has clamped edges the next step is to calculate the function \( F_i(\alpha) \), for the small and large panels.

\[ F_i(\alpha) = 3.68\beta^3 - 2.26\beta^2 - 0.336\beta + 0.376, \quad 1 < \alpha < 2.5 \] (B.7)

\[ \frac{0.743}{\alpha^2}, \quad \alpha > 2.5 \]

\( F_i(\alpha) \) for the small panels

aspect ratio: \( \alpha = a/b = 1 \)

so, \( \beta = \log(\alpha) = 0 \)

therefore: \( F_i(\alpha) = 0.376 \)
$F_i(\alpha)$ for the large panels

aspect ratio: $\alpha = a/b = 1.3333$

so, $\beta = \log(\alpha) = 0.12494$

therefore: $F_i(\alpha) = 3.68(1.12494)^3 - 2.26(1.12494)^2 - 0.336(1.12494) + 0.376$

$F_i(\alpha) = 0.306$

Now the total compliance of each panel can be computed using equation (2.4) and then summed to give the total compliance.

$$\sum_{i=1}^{n} C_{pi} = 3 \left[ \frac{.001(A_{pi})^3F_i(\alpha)}{B} \right] + 2 \left[ \frac{.001(A_{pi})^3F_i(\alpha)}{B} \right]$$

$$\sum_{i=1}^{n} C_{pi} = 3 \left[ \frac{.001(1.14452)^3(.306)}{74.07} \right] + 2 \left[ \frac{.001(.092903)^3(.376)}{74.07} \right]$$

$$\sum_{i=1}^{n} C_{pi} = 4.5550 \times 10^{-8} \text{ m}^2/\text{N}$$

Now the IL in the low frequency region for the sealed enclosure can be calculated.

$$\text{IL(dB)} = 20 \log \left[ 1 + \frac{C_v}{\sum_{i=1}^{n} C_{pi}} \right]$$

$$\text{IL} = 20 \log \left[ 1 + \frac{2.6522 \times 10^{-7}}{4.5550 \times 10^{-8}} \right] = 16.8 \text{ dB}$$

This value is constant throughout the low frequency range.
**IL for the Open Enclosure**

Once again, the compliances of the interior air volume of the open enclosure and panels are calculated.

\[
C_v = \frac{3.8543 \times 10^{-2} \text{ m}^3}{1.21 \text{ kg/m}^3 (343 \text{ m/s})^2} = 2.7075 \times 10^{-7} \text{ m}^5/\text{N}
\]

It is assumed that the total compliance of the panels is the same as the compliance for the sealed enclosure. Once again, the panels are assumed to have clamped edges.

Next, the total acoustic impedance for the openings is computed as follows:

\[
\frac{1}{Z_{tot}} = \frac{2}{Z_{ls}} + \frac{2}{Z_{lt}}
\]  

(B.10)

solving for \(Z_{tot}\):

\[
Z_{tot} = \frac{Z_{ls}Z_{lt}}{2Z_{ls} + 2Z_{lt}}
\]  

(B.11)

where \(Z_{tot}\) = total acoustic impedance for all openings

\(Z_{ls}\) = acoustic impedance for the slits across each of the short sides

\(Z_{lt}\) = acoustic impedance for the slits across each of the long sides

Now the acoustic impedance for each opening is calculated using equation (2.15).

*acoustic impedance for the two short openings*

\[
Z_{ls} = \frac{6\rho}{t w_s} \left[ \frac{2\mu \left( h + 1.2\sqrt{tw_s} / \pi \right)}{t^2} + j\omega \frac{h + 1.2\sqrt{tw_s} / \pi}{5} \right]
\]  

(B.12)
The acoustic impedance for the two long openings is given by:

\[
Z_{\Delta t} = \frac{6\rho}{twt} \left[ \frac{2\mu \left( h + 1.2\sqrt{twt/\pi} \right)}{t^2} + j\omega \frac{h + 1.2\sqrt{twt/\pi}}{5} \right] 
\]

Equation (B.11) is then used to compute the total acoustic impedance of all four openings.

The IL for the open enclosure can be calculated using equation (2.12):

\[
IL(dB) = 20 \log \left| \frac{1}{\sum_{i=1}^{n} C_i \frac{1}{1 + j\omega \sum_{i=1}^{n} C_i Z_{\Delta t, tot}}} \right| 
\]

Equations (B.9) and (B.11) to (B.14) are used in a computer simulation that calculates and plots the IL for the sealed and open enclosures over the frequency range 0 to 500 Hz.
Vita

Aaron Joseph Layos was born April 18, 1970 in Tacoma Park, Maryland. He completed his undergraduate studies in Mechanical Engineering at Virginia Polytechnic Institute and State University in May of 1993. During that period he worked during the summers as an engineering assistant at Marsh-McBirney, Inc. in Frederick, Maryland. In addition, Aaron has performed undergraduate research in fluids mixing processes. Following undergraduate school, he pursued a Master of Science and completed his work in September of 1995. Upon completion of his M.S. degree, he will search for a job.