

University of Southern Queensland  
Faculty of Engineering & Surveying

## **Active Vibration Control**

A dissertation submitted by

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in fulfilment of the requirements of

**ENG4112 Research Project**

towards the degree of

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# Abstract

This project's overall aim involved minimising structural vibration by actively controlling the vibration of the structural member itself. The control procedure implemented one specialised form of active control, namely adaptive feed-forward control. The motivation behind the project was to investigate low cost vibration attenuation systems for machinery panels and enclosures.

Eddy currents have long been used as a form of damping in dynamic systems. When a conductor moves through a magnetic field, eddy currents are induced. These currents oppose the change in flux that produced them and also generate heat due to the impedance of the conductor. This transferral of kinetic energy into eddy currents can be used effectively as a means of damping a vibrating system. A vibrating structure can be set up such that a section contains a conductor moving relatively perpendicular to a magnetic field. This will naturally dampen the vibration. If the magnetic field is then moved exactly 180 degrees out of phase with the conductor, the relative velocity is increased. This increases the magnitude of the eddy currents thereby amplifying the damping effect.

The most defining requirement of the project was to build and test a prototype control system for use in an industrial environment. The final test system utilised an adaptive feed-forward controller in conjunction with an eddy current damper. Testing of the system revealed a significant increase in damping with the control system in place.

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I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

GREGORY LISTER

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Date

# Acknowledgments

I would like thank my supervisors, Chris Snook and John Leis, for their guidance and advice throughout the project. I would also like to extend my gratitude to Terry, Brett and Dean for their help in providing all of the equipment I needed for testing.

GREGORY LISTER

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# Contents

<b>Abstract</b>	<b>i</b>
<b>Acknowledgments</b>	<b>iv</b>
<b>List of Figures</b>	<b>ix</b>
<b>Chapter 1 Introduction</b>	<b>1</b>
1.1 Scope and Organisation of the Dissertation . . . . .	2
1.2 The Foundation of the Literature Review . . . . .	2
1.3 What is Active Vibration Control? . . . . .	3
1.4 Rationale for Noise and Vibration Control . . . . .	4
1.5 History of Active Noise Control . . . . .	5
<b>Chapter 2 Fundamentals of Active Vibration Control</b>	<b>7</b>
2.1 Introduction . . . . .	8
2.2 Physical Control Strategies . . . . .	8
2.3 Eddy Currents as a Means of Damping . . . . .	9

<b>CONTENTS</b>	<b>vi</b>
<hr/>	
2.4 Vibration in Structures . . . . .	10
2.5 Adaptive Feed-Forward Control . . . . .	11
2.6 Digital Filtering . . . . .	14
2.6.1 Wiener Filter Theory . . . . .	14
2.6.2 The LMS Algorithm . . . . .	15
2.6.3 LMS Variants . . . . .	16
2.7 Conclusion . . . . .	17
<b>Chapter 3 The Contemporary Science of Vibration Control</b>	<b>18</b>
3.1 Introduction . . . . .	19
3.2 Passive Eddy Current Damping . . . . .	19
3.2.1 Results and Conclusions . . . . .	21
3.3 Self-Adaptive Fluid Damping . . . . .	22
3.3.1 Results and Conclusions . . . . .	23
3.4 Active Piezoelectric Control . . . . .	24
3.4.1 Results and Conclusions . . . . .	25
3.5 Semi-Active Friction Damping . . . . .	27
3.5.1 Results and Conclusions . . . . .	28
3.6 General Conclusion of the Research . . . . .	30
<b>Chapter 4 Preliminary System Design</b>	<b>31</b>

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4.1	Introduction . . . . .	32
4.2	The Physical System . . . . .	32
4.2.1	The Vibrating Structure . . . . .	32
4.2.2	The Physical Control Strategy . . . . .	32
4.2.3	The Sensors . . . . .	34
4.3	The Digital System . . . . .	34
4.4	Modal Analysis . . . . .	37
4.4.1	Analysis Procedure . . . . .	37
4.4.2	Modal Analysis Results . . . . .	38
4.4.3	Conclusion . . . . .	44
4.5	MATLAB Simulations . . . . .	45
4.5.1	Simulation Procedure . . . . .	45
4.5.2	Simulation Results . . . . .	45
4.5.3	Conclusion . . . . .	53
<b>Chapter 5 Final Design and Real-Time Testing</b>		<b>55</b>
5.1	Introduction . . . . .	56
5.2	The Physical System . . . . .	56
5.3	Program Development . . . . .	58
5.4	Testing Procedure . . . . .	59
5.5	Results . . . . .	61

---

5.5.1	Average Settling Times . . . . .	61
5.5.2	Error Plots . . . . .	62
5.6	Conclusion . . . . .	68
<b>Chapter 6</b>	<b>Conclusions and Further Work</b>	<b>70</b>
6.1	Achievement of the Project Objectives . . . . .	71
6.2	Further Work and Recommendations . . . . .	72
	<b>References</b>	<b>73</b>
	<b>Appendix A Project Specification</b>	<b>75</b>
	<b>Appendix B MATLAB Source Code</b>	<b>77</b>
B.1	The <code>lms.m</code> MATLAB Function . . . . .	78
	<b>Appendix C Assembly and C Source Code</b>	<b>80</b>
C.1	The <code>test.c</code> Test Function . . . . .	81
C.2	The <code>ioportlib.asm</code> Library . . . . .	84
	<b>Appendix D AXIOM AX5412 Specifications</b>	<b>87</b>

# List of Figures

1.1	The Final Test System . . . . .	3
1.2	The Concept of ANC . . . . .	4
1.3	Lueg's 1933 Patent Application . . . . .	5
2.1	Eddy Current Generation . . . . .	10
2.2	Mode Shapes for Typical Beam . . . . .	11
2.3	Feed-Back Control System . . . . .	12
2.4	Feed-Forward Control System . . . . .	13
2.5	The FIR Filter . . . . .	14
3.1	Eddy Current Damping System . . . . .	20
3.2	Optimal Magnetic Field . . . . .	20
3.3	Conducting Plate Motion . . . . .	21
3.4	Eddy Current Damping Results . . . . .	21
3.5	Experimental IVFU . . . . .	22
3.6	Simulation of Second Mode Shape . . . . .	22

---

3.7	Fluid Force and Displacement . . . . .	23
3.8	Beam Configuration for Numerical Analysis . . . . .	24
3.9	Experimental Configuration . . . . .	25
3.10	Displacement Response to Impulse . . . . .	25
3.11	Displacement Response to Random Excitation . . . . .	26
3.12	Experimental Response to Initial Tip Displacement . . . . .	26
3.13	Friction Model used for Simulations . . . . .	27
3.14	Double-Shear Friction Device . . . . .	28
3.15	Total System Energy for Initial Displacement . . . . .	28
3.16	Mass 3 Position for Initial Displacement . . . . .	29
3.17	Improved Energy Dissipation with Control . . . . .	29
4.1	End View of the Proposed Eddy Current Damper . . . . .	33
4.2	Isometric View of the Proposed Eddy Current Damper . . . . .	34
4.3	ANALOG DEVICES SHARC EZ KIT . . . . .	35
4.4	Microchip dsPIC30F4013 . . . . .	35
4.5	ELEXOL USB I/O 24R . . . . .	36
4.6	AXIOM AX5412 Data Acquisition Card . . . . .	36
4.7	Mode 1 . . . . .	39
4.8	Mode 2 . . . . .	39
4.9	Mode 3 . . . . .	40

---

4.10 Mode 4 . . . . .	40
4.11 Mode 5 . . . . .	41
4.12 Mode 6 . . . . .	41
4.13 Mode 7 . . . . .	42
4.14 Mode 8 . . . . .	42
4.15 Mode 9 . . . . .	43
4.16 Mode 10 . . . . .	43
4.17 Non-Lateral Motion of Modes 5 and 9 . . . . .	44
4.18 Ctrl Signal for Harmonic Input, Filter Length = 10, $\mu = 0.001$ . . . . .	46
4.19 Error Signal for Harmonic Input, Filter Length = 10, $\mu = 0.001$ . . . . .	46
4.20 Ctrl Signal for Gaussian Noise Input, Filter Length = 10, $\mu = 0.001$ . . . . .	47
4.21 Error Signal for Gaussian Noise Input, Filter Length = 10, $\mu = 0.001$ . . . . .	47
4.22 Ctrl Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.001$ . . . . .	48
4.23 Error Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.001$ . . . . .	48
4.24 Error Signal for Harmonic + Noise Input, Filter Length = 20, $\mu = 0.001$ . . . . .	50
4.25 Error Signal for Harmonic + Noise Input, Filter Length = 30, $\mu = 0.001$ . . . . .	50
4.26 Error Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.004$ . . . . .	51
4.27 Error Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.008$ . . . . .	51
4.28 Error Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.012$ . . . . .	52
4.29 Error Signal for Harmonic + Noise Input, Filter Length = 10, $\mu = 0.020$ . . . . .	52

---

4.30	Harmonic + Noise Input, Filter Length = 10, $\mu = 0.5$ . . . . .	53
5.1	The Design Change . . . . .	56
5.2	The Whole System . . . . .	57
5.3	The Eddy Current Damper . . . . .	57
5.4	The LMS Filter . . . . .	59
5.5	The Intended Input . . . . .	60
5.6	Test 1 - No Control . . . . .	62
5.7	Test 2 - No Control . . . . .	62
5.8	Test 3 - No Control . . . . .	63
5.9	Test 4 - No Control . . . . .	63
5.10	Test 5 - No Control . . . . .	64
5.11	Test 6 - Semi-Active Control . . . . .	64
5.12	Test 7 - Semi-Active Control . . . . .	65
5.13	Test 8 - Semi-Active Control . . . . .	65
5.14	Test 9 - Semi-Active Control . . . . .	66
5.15	Test 10 - Semi-Active Control . . . . .	66
5.16	Test 11 - Passive Control . . . . .	67
5.17	Test 12 - Passive Control . . . . .	67
5.18	Test 13 - Passive Control . . . . .	68

## Chapter 1

# Introduction

## 1.1 Scope and Organisation of the Dissertation

This dissertation is organised as follows:

**Chapter 2 :**

Involves the fundamental principles of active vibration control and digital filtering.

**Chapter 3 :**

Presents some of the current research in the domain of active vibration control.

**Chapter 4 :**

Looks at the initial design work of the prototype control system.

**Chapter 5 :**

Concerns the final design, implementation and testing of the control system.

**Chapter 6 :**

Concludes the project and presents some future avenues of investigation.

Refer to Appendix A for a detailed list of the project specifications.

## 1.2 The Foundation of the Literature Review

At this stage it is necessary to define the practical aspect of the project so that the reader understands the basis for the upcoming literature review and research. The final and probably most defining requirement of this project was to build and test a prototype vibration suppression system. This system utilised an adaptive feed-forward controller in conjunction with an eddy current damper. The figure below shows a schematic of the final test system.

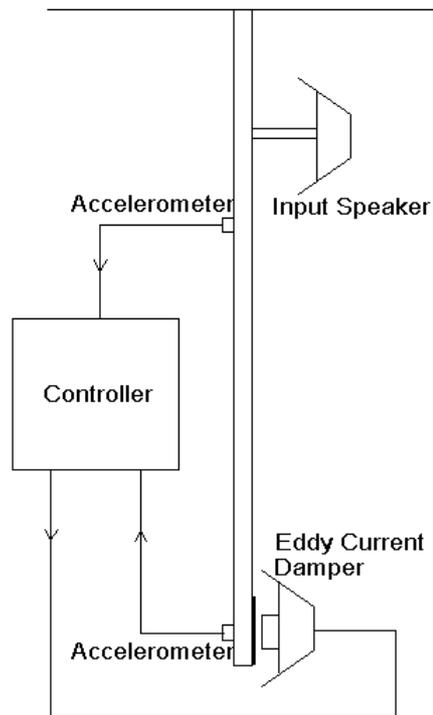


Figure 1.1: The Final Test System

### 1.3 What is Active Vibration Control?

Snyder (2000) defines active *noise* control (ANC) as a technique that “aims to reduce sound levels by canceling the unwanted acoustic waves with a second set of electronically generated acoustic waves.” Active *vibration* control, now referred to as AVC, is often achieved using similar methods to those of ANC but the process involves structural vibration rather than airborne acoustic noise. It should be noted that destructive cancellation is not the only means of actively controlling a vibrating structure. As explained later, methods such as tuned damping and hybrid control can also be used effectively to control structural vibration. The figure shown below illustrates the concept of ANC.

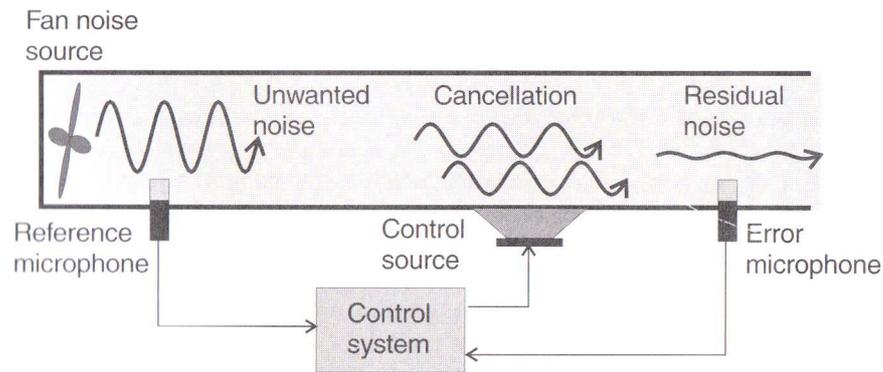


Figure 1.2: The Concept of ANC

(Snyder 2000)

## 1.4 Rationale for Noise and Vibration Control

Modern society is becoming noisier! As industry has developed over the years so has the amount of noise we produce. As “workplace health and safety” standards have increased, the tolerance for noise has decreased. This is not only true in industry as it is also reflected in society as the world’s population has boomed and urban density has increased dramatically.

Traditional methods of sound attenuation have been passive methods which involve absorbing or reflecting the acoustic energy (Snyder 2000). In many cases passive control only protects targeted individuals and in some situations it may not be beneficial to employ this technique. This is not to say that active control can fully take the role of passive techniques as it too has limitations.

Sound attenuation is not the only motive for active vibration control. Applications ranging from high precision medical instruments to military stealth require very low levels of structural vibration. Today, active vibration control is moving from a research topic to a wide range of commercial applications.

## 1.5 History of Active Noise Control

The concept of ANC is over seventy years old. Paul Lueg is widely accepted as one of the first to publish documents concerning ANC. He is known to have submitted a patent application in 1933 which involved a transducer being utilised to generate anti-noise (Hansen & Kestell 1999).

The idea presented by Lueg was feasible but did not exist due to technological constraints and thus the idea died for another twenty years. H.F. Olsen resurrected the concept in the mid 1950's. Olsen was the first person to demonstrate that ANC was possible by constructing an electronic *sound absorber*. It consisted of a microphone immediately adjacent to a loudspeaker (Hansen & Kestell 1999).

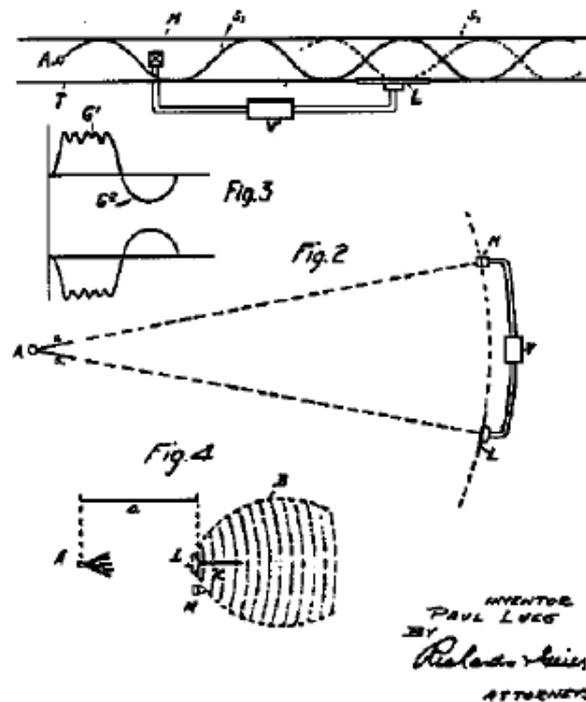


Figure 1.3: Lueg's 1933 Patent Application

(Hansen & Kestell 1999)

Before the 1980's, practical experimentation relied on large expensive laboratory computers and thus the technology was restricted to research. Commercially viable ANC applications emerged during the 1980's. This was due to the production of low cost specialised digital signal processors (Hansen & Kestell 1999).

## **Chapter 2**

# **Fundamentals of Active Vibration Control**

## 2.1 Introduction

This chapter is dedicated to some of the fundamental aspects of active vibration control. This field is rapidly changing and extremely diverse and thus only a specific focus on topics which pertain to this dissertation is provided. Initially, the four broad categories of vibration control are defined. This is followed by a brief investigation of how eddy currents can be used to generate a viscous damping effect. Following that, vibrations in physical structures are qualified and the process of adaptive feed-forward control is described. Finally, the chapter ends with a fairly detailed examination of digital filtering.

## 2.2 Physical Control Strategies

There are various means of controlling structural vibration. Passive and active control forces can be applied and physical parameters such as mass, stiffness and damping can be modified. In a physical sense, there are four broad categories which encompass the control aspect of structural vibration. Zhang (2001) defines these categories as:

**Active Control** – Active control systems use an external source to drive control actuators. These actuators apply forces to the structure in a pre-determined fashion and can be used to add or dissipate energy within the system.

**Passive Control** – In passive control, an external power source is not required as the control devices impart forces as a result of the motion of the structure. Examples of passive control devices are foam padding, friction dampers and counterweights.

**Hybrid Control** – Hybrid control systems utilise a combination of passive and active control devices. For example, a control system may use passive friction dampers as well as active piezoelectric actuators.

**Semi-Active Control** – Semi-active control systems are essentially active systems which require a comparatively small external power source. They generally do not add mechanical energy to the structural system.

All of these control schemes are applicable in different situations and it is nonsensical to simply compare them directly, however, it should be pointed out that semi-active control strategies have the advantage of being simple, reliable and effective. As well as that, they require a relatively small external power source and generally guarantee bounded-input bounded-output stability as mechanical energy is not added to the system (Zhang 2001).

## 2.3 Eddy Currents as a Means of Damping

There are currently many methods of adding damping to a vibrating system. Many of these methods rely on friction and thus require contact with the vibrating structure. The use of eddy currents provides an alternative method that does not involve physical contact.

In such a system, eddy currents are generated in a conductive material when it is subject to a time-changing magnetic field. These eddy currents set up their own magnetic field which oppose the primary fixed field. This results in an applied force that is related to the velocity of the change of magnetic flux. These eddy currents are dissipated as heat as a result of the internal resistance of the conducting material. If a system is set up to allow the generation of velocity-dependent currents and subsequent dissipation as heat, it will behave as a viscous damper (Sodano, Inman & Belvin 2006).

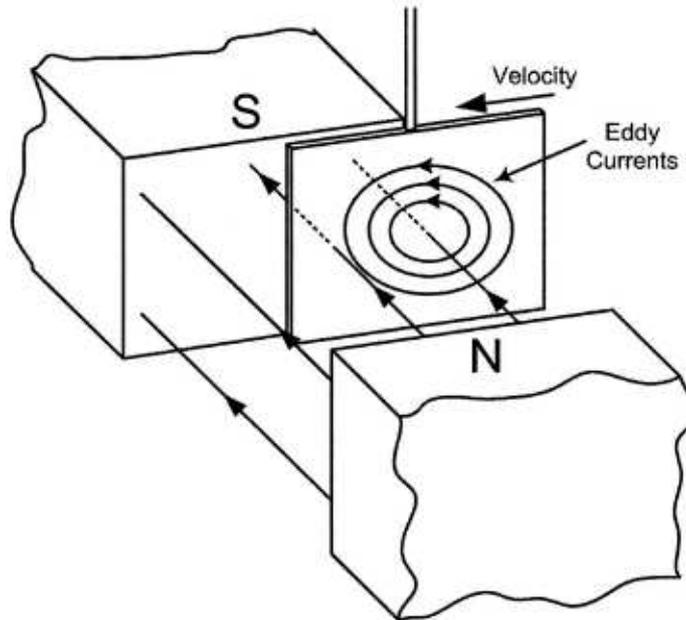


Figure 2.1: Eddy Current Generation

(Sodano et al. 2006)

## 2.4 Vibration in Structures

Vibration in physical structures can be qualified by the propagation of waves. There are three types of waves in physical structures, namely longitudinal, flexural and shear waves. Shear and longitudinal waves are not directly responsible for sound radiation but may be converted to transverse waves at structural discontinuities (Hansen & Snyder 1997). This means that shear and longitudinal waves must also be considered in the active control of a physical structure. Vibration in simple structures can be analysed using analytical methods or numerical methods such as finite element analysis (FEA). Numerical methods are often easier to implement but may be sensitive to parameters such as mesh quality in the case of FEA.

An important consideration of vibrating systems is that of modal dynamics. Modal analysis is the process of determining the resonant frequencies and mode shapes of a

system (Hansen & Snyder 1997). Modal analysis of a vibrating structure is a useful tool when attempting active control of a vibrating structure. If performed correctly, it will portray the natural modes of vibration and the frequencies at which they occur. This is valuable because locations of minimal displacement can be determined for various mode shapes and thus the placement of sensors and actuators in a control system can be avoided at these locations. As well as that, the performance of the control system can be determined at these resonant frequencies. This is important as resonant vibration should be minimised as much as possible due to the large amplitude, complex vibrations incurred as a result of resonance.

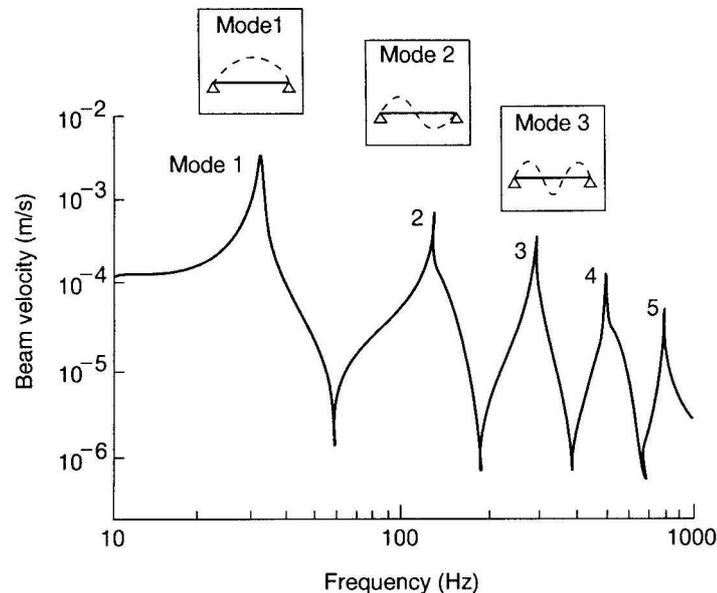


Figure 2.2: Mode Shapes for Typical Beam

(Fuller, Elliott & Nelson 1996)

## 2.5 Adaptive Feed-Forward Control

Before describing feed-forward control, it is necessary to define its counterpart, feed-back control. In feed-back control some output quantity is measured and compared to a desired value and the resulting difference or error is used to adjust the output (Hansen & Snyder 1997).

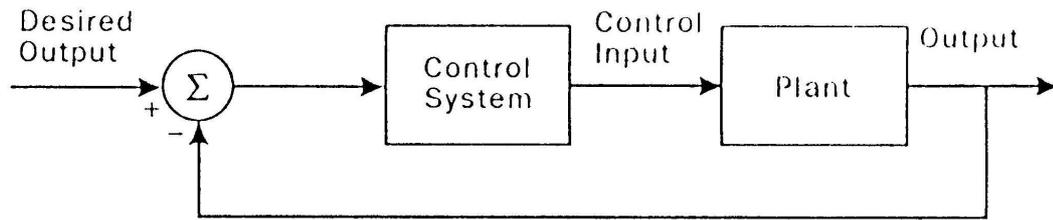


Figure 2.3: Feed-Back Control System

(Hansen &amp; Snyder 1997)

Feed-forward control is fundamentally different from feed-back in that it involves predicting the impending primary disturbance (Hansen & Snyder 1997). In this type of system, the input signal is sampled and fed *forward* to the controller, however, it should be observed that the error signal is also fed *back* to the controller. Feed-forward control is generally applicable in situations where the primary disturbance is harmonic or where the propagation path is down a waveguide and the disturbance at any point is a function of the upstream excitation (Hansen & Snyder 1997). Furthermore, *adaptive* feed-forward systems *modify* their characteristics to achieve specific objectives (Kuo & Morgan 1996). One general limitation of these systems is that they are not robust to rapidly changing signals as they often require a significant amount of time to train.

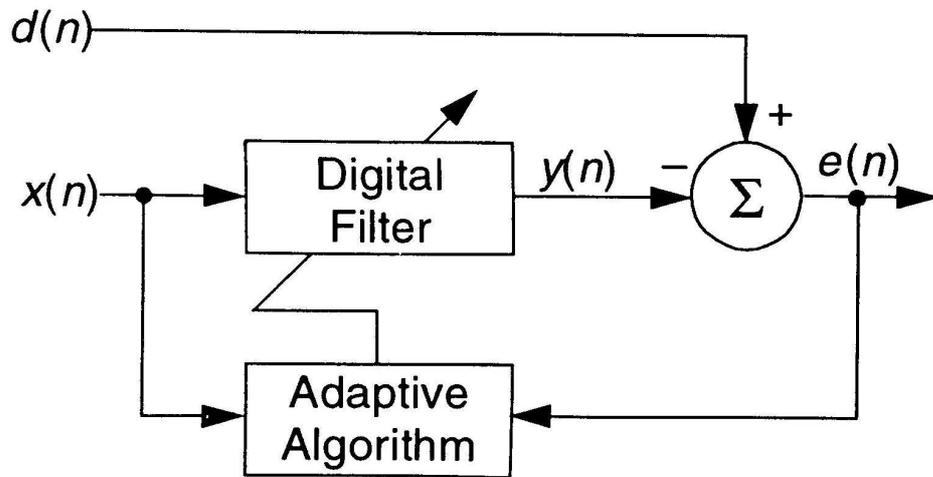


Figure 2.4: Feed-Forward Control System

(Kuo &amp; Morgan 1996)

The figure depicted above shows a block diagram of an adaptive filter.  $d(n)$  is the primary input,  $x(n)$  is the reference input or the sampled primary input,  $y(n)$  is the control output and  $e(n)$  is the error signal. The purpose of the adaptive filter is to continually update the coefficients of the digital filter. The filter coefficients are updated on a sample by sample basis so that the error signal is progressively minimised.

There are two basic filter implementations that can be used in adaptive control namely, Finite Impulse Response (FIR) and Infinite Impulse Response filters (IIR). The following discussion is limited to FIR filters as it is the most commonly used in active control. FIR filters have the distinct advantage of remaining stable all the time as they do not possess system poles (Kuo & Morgan 1996).

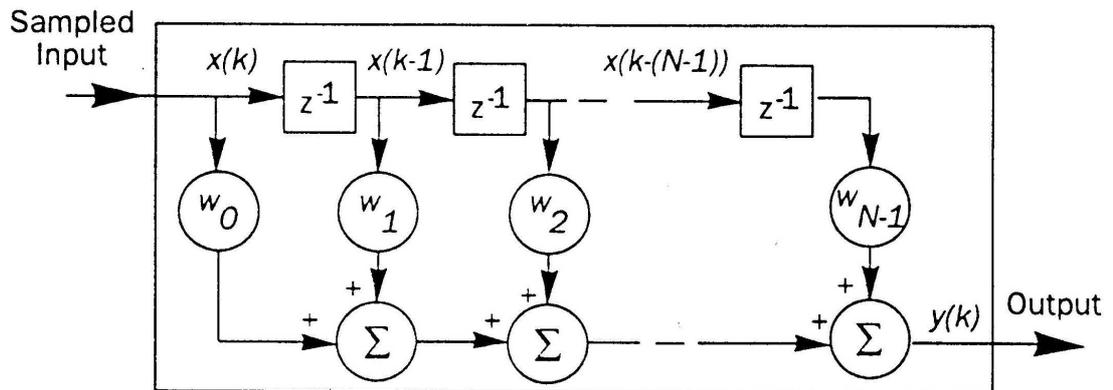


Figure 2.5: The FIR Filter

(Hansen &amp; Snyder 1997)

In this figure,  $x(k)$  is the sampled input from the reference sensor.  $z^{-1}$  represents one sampling delay and there are  $N$  filter outputs called filter taps. The  $w$  values are the filter coefficients and  $y(k)$  is the net output of the filter. The output of the filter is the weighted combination of previous input samples (Hansen & Snyder 1997):

$$y(k) = \sum_{n=0}^{N-1} w_n(k)x(k-n) = \mathbf{w}^T(k)\mathbf{x}(k) \quad (2.1)$$

## 2.6 Digital Filtering

There are two main theories of adaptive control which can be implemented in active control of noise or vibration. These are Wiener and Kalman filter theory. Wiener theory is based on the Least Mean Square (LMS) algorithm and Kalman theory involves the Recursive Least Square algorithm (RLS). This section deals with Wiener theory only.

### 2.6.1 Wiener Filter Theory

The nature of a linear discrete filter is to estimate a desired response, given a set of input samples, such that the mean-square value of the estimation error is minimised. The mean-square error is defined as the difference between the desired response and

the actual response (Haykin 1991). This is illustrated as:

$$y(k) = \sum_{k=0}^{\infty} w_k x(n - k) \quad (2.2)$$

Where:  $x(k)$  = filter input,  $y(k)$  = filter output,  $w(k)$  = filter weights

Haykin (1991) present the filter cost function as:

$$\mathbf{J} = E [|\epsilon(k)|^2] \quad (2.3)$$

Where:  $e(k) = d(k) - y(k)$

This is used to optimise the filter weights by the MMSE criterion. The equations used to do this are the Wiener-Hopf equations (Haykin 1991):

$$\sum_{i=0}^{\infty} w_{o,i} r(i - n) = p(-n) \quad (2.4)$$

or

$$\mathbf{w}_o = \mathbf{R}^{-1} \mathbf{p} \quad (2.5)$$

Where:

$$\begin{aligned} \mathbf{w}_0 &= \text{filter vector} \\ &= [w_{o,0}, w_{o,1}, \dots, w_{o,M-1}]^T \\ \mathbf{p} &= \text{cross-correlation vector} \\ &= E [x(k)d(k)] \\ \mathbf{R} &= \text{correlation matrix} \\ r_{i-n} &= E[x(k - n + 1)x(k - i + 1)] \end{aligned}$$

If the filter coefficients are optimal, the error  $e(k)$  and the input,  $x(k)$ , are orthogonal or uncorrelated. This is the principal of orthogonality and is the basis for the LMS filter.

### 2.6.2 The LMS Algorithm

The simplest implementation of a Wiener filter is the LMS algorithm. This algorithm updates the filter weights on every sample by estimating the gradient of the cost func-

tion,  $J$ . This is done by estimating the cross-correlation vector and auto-correlation matrix (Haykin 1991):

$$\hat{\mathbf{R}}(k) = \mathbf{x}(k)\mathbf{x}^T(k) \quad (2.6)$$

$$\hat{\mathbf{p}}(k) = \mathbf{x}(k)d(k) \quad (2.7)$$

Hansen & Snyder (1997) show a complete derivation of the LMS algorithm. The final expression, used to update the filter weights, is given as:

$$\mathbf{w}(k+1) = \mathbf{w}(k) + 2\mu e(k)\mathbf{x}(k) \quad (2.8)$$

Where,  $\mu$  is the convergence coefficient that determines the speed of adaptation. Hansen & Snyder (1997) also provide a set of steps for implementing an LMS based FIR filter.

1. Advance values in filter and input new reference sample
2. Calculate new filter output
3. Obtain the error signal
4. Calculate new filter weights
5. Repeat

### 2.6.3 LMS Variants

Kuo & Morgan (1996) present several variations of the simple LMS algorithm which can be used to change its characteristics. Whether these variations substantially improve the overall performance is debatable and depends on many factors such as the processing power of the control system.

One variation is the Normalised LMS Algorithm. This is similar to the simple LMS algorithm but is *normalised* in that the step size,  $\mu$ , is scaled and depends on the power of the input signal,  $x$ . This variant is often used to improve the stability without having to calculate the largest eigenvalue of the auto-correlation matrix. The step size is calculated as:

$$\mu = \frac{\mu_n}{2M\hat{P}_x(k)} \quad (2.9)$$

Where,  $\hat{P}_x$  is the estimate of the power of the input signal,  $\mu_n$  is the normalised step size and  $M$  is the number of filter weights.

The stability of the LMS algorithm can also be improved by introducing a leakage factor. This is the Leaky LMS Algorithm and can be expressed as:

$$\mathbf{w}(k+1) = \beta \mathbf{w}(k) + 2\mu e(k)\mathbf{x}(k) \quad (2.10)$$

Where,  $\beta$  is the leakage factor which alters the response to non-stationary signals.

The step size of the algorithm can also be varied according to the difference between the current mean-square error and the minimum mean-square error. This can improve the rate of convergence of the algorithm. In this case a diagonal matrix of step sizes for each filter tap is employed. The algorithm is expressed as:

$$\mathbf{w}(k+1) = \mathbf{w}(k) + \mathbf{U}(k)e(k)\mathbf{x}(k) \quad (2.11)$$

Where,  $\mathbf{U}(k)$  is the diagonal matrix of step sizes.

## 2.7 Conclusion

This chapter strove to introduce some of the main concepts of active vibration control. It should be noted that successful active control of a vibrating structure can be a complicated task. This is because this field draws on the knowledge of many disciplines, such as and not limited to, structural analysis, acoustics, embedded systems design and programming. Many people refer to this domain as an *art*.

## **Chapter 3**

# **The Contemporary Science of Vibration Control**

## 3.1 Introduction

This chapter looks at current research in the field of vibration control. The reasons for this study are firstly, to place this project within the context of current practice and secondly, to provide a basis for the practical experimentation which follows. Examples of passive, active and semi-active control are examined.

## 3.2 Passive Eddy Current Damping

Inman, Sodano, Belvin & Bae (2006) recently performed a study on the use of eddy currents as a means of suppressing vibration of a cantilever beam. The implemented control scheme was purely passive as no external power source was required for control. Two conducting plates were placed on the beam and allowed to vibrate in a magnetic field generated by two opposing permanent magnets. The relative motion of the plate and magnetic field allowed the generation and subsequent dissipation of eddy currents within the plate, causing a viscous damping effect.

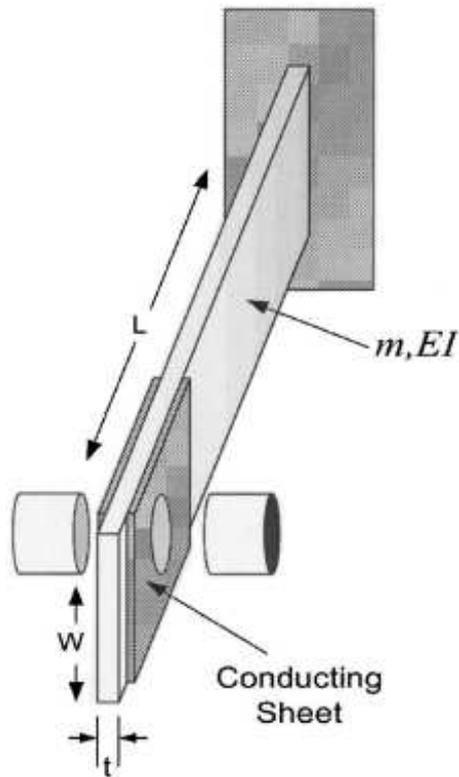


Figure 3.1: Eddy Current Damping System

(Inman et al. 2006)

Inman et al. (2006) realised that two opposing permanent magnets will generate a more dense magnetic field than a single magnet.

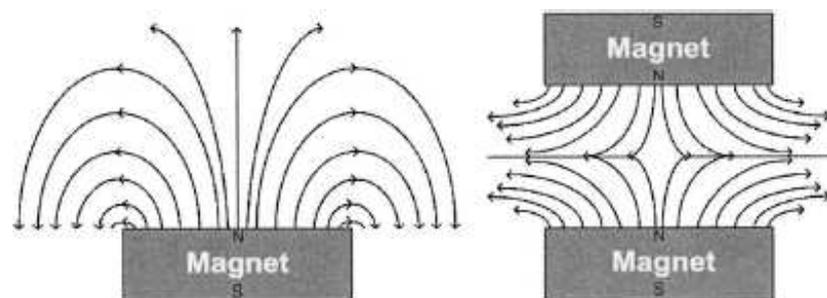


Figure 3.2: Optimal Magnetic Field

(Inman et al. 2006)

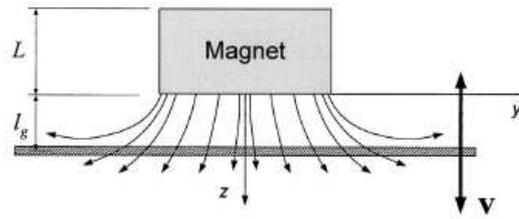


Figure 3.3: Conducting Plate Motion

(Inman et al. 2006)

The relative motion of the conducting plate and the magnetic field allowed the plate to cut the *horizontal* components of magnetic flux as shown above.

Inman et al. (2006) determined a mathematical model of the system and tested the model with a real system. The first four mode shapes of the beam were tested and an initial displacement test was performed.

### 3.2.1 Results and Conclusions

The damping system was very effective in suppressing vibration of the cantilever beam. It was able to critically damp the first mode of vibration and reduce the second through fourth modes by 31, 22.5 and 14 dB respectively.

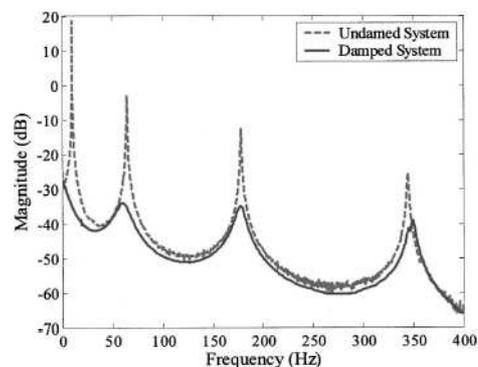


Figure 3.4: Eddy Current Damping Results

(Inman et al. 2006)

### 3.3 Self-Adaptive Fluid Damping

In this study, Zhang, Li, Xu & Li (2006) investigated the use of an Interior Inlay Viscous Fluid Unit (IVFU) as a new method for vibration suppression of flexible structures. The first and second mode shapes of a beam were analysed using the computational fluid package Fluent 6.1. In this experiment, the IVFU consisted of 50% water and 50% air.

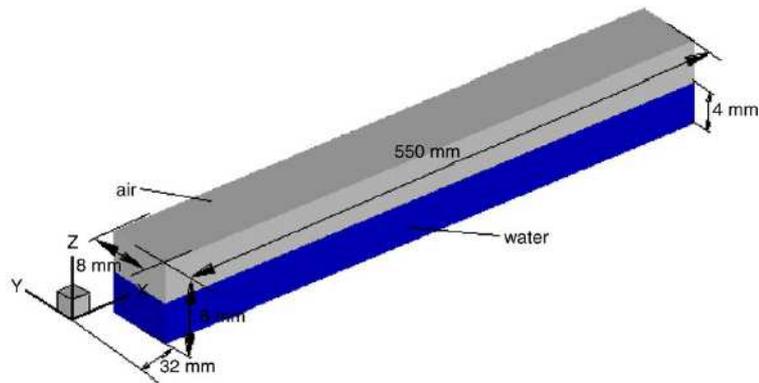


Figure 3.5: Experimental IVFU

(Zhang et al. 2006)

Zhang et al. (2006) developed a numerical model and governing equations for the system and simulated the first two modes of vibration. The fluid forces were determined at areas of maximum beam displacement.

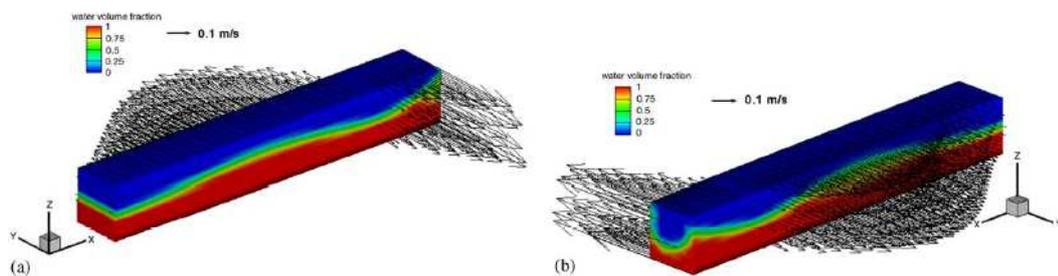


Figure 3.6: Simulation of Second Mode Shape

(Zhang et al. 2006)

## 3.3.1 Results and Conclusions

Fluid inside the IVFU converged to areas of maximum displacement. As well as that, the period of the fluid force was found to be the same as the beam but out of phase. The phase difference between force and displacement was 206 degrees for the first mode and 186 degrees for the second mode. This phase difference resulted in a viscous damping effect which increased if the amplitude of vibration increased.

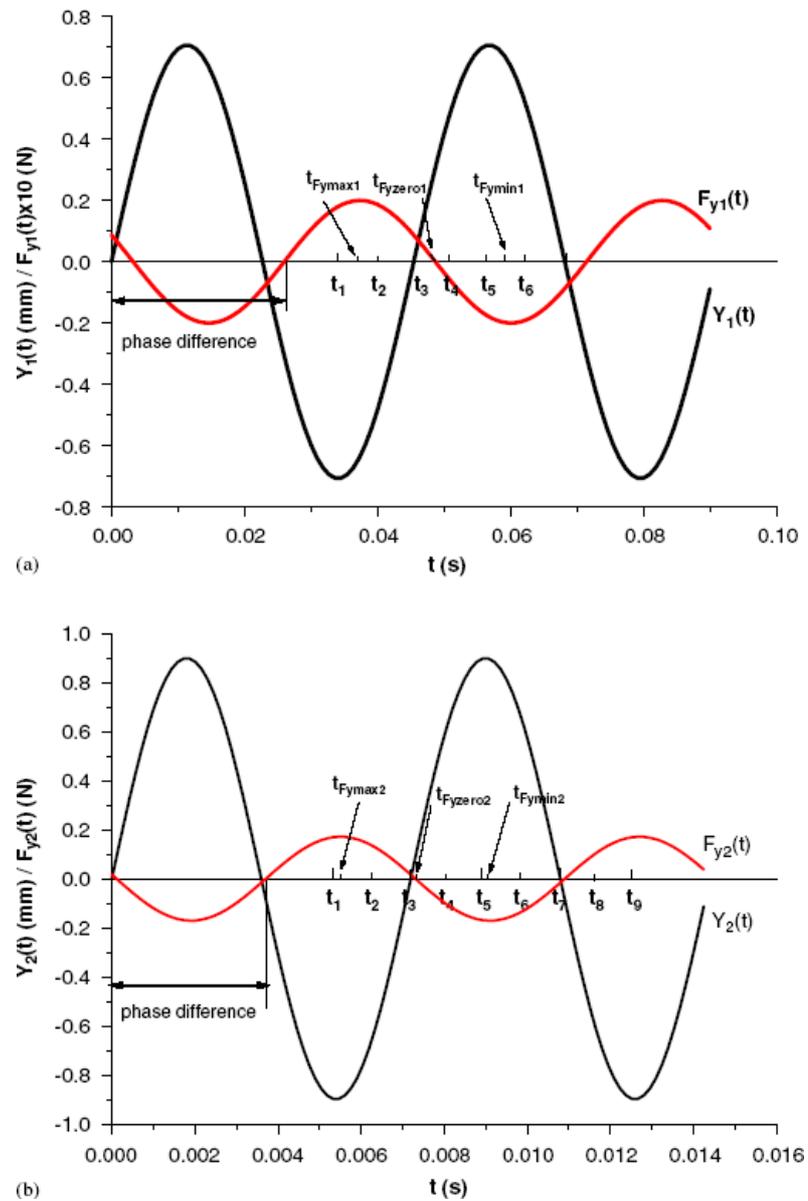


Figure 3.7: Fluid Force and Displacement

(Zhang et al. 2006)

### 3.4 Active Piezoelectric Control

In this analysis, Dong, Meng & Peng (2006) investigated the efficiency of the Observer/Kalman Filter Identification (OKID) technique in the numerical simulation and experimental vibration control of piezoelectric smart structures.

A Linear Quadratic Gaussian (LQG) algorithm was developed and incorporated into an ANSYS model of a cantilever beam. Closed loop simulations were then performed in an effort to gauge the performance of the system. The simulations included an impulse test and a random input test.

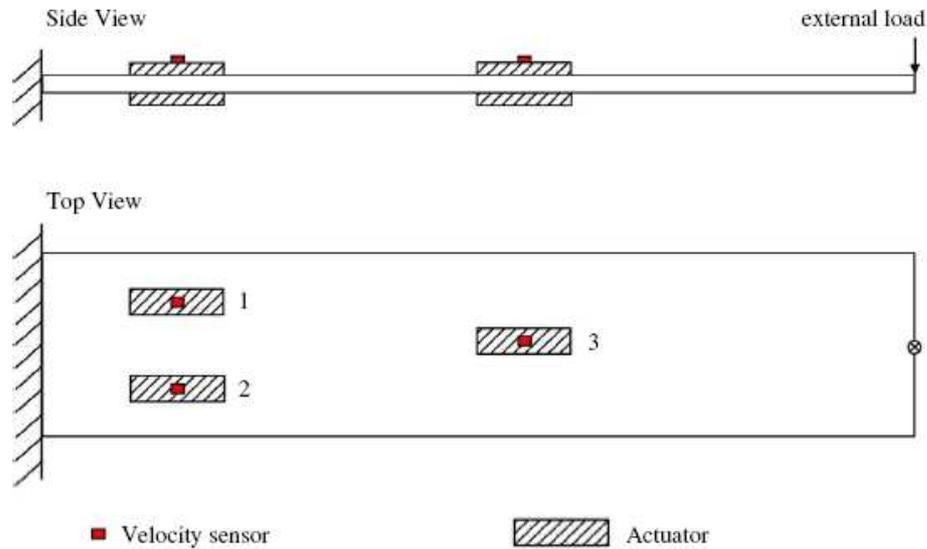


Figure 3.8: Beam Configuration for Numerical Analysis

(Dong et al. 2006)

After that, a state space model of the system was developed from the numerical analysis and the controller was implemented on a Digital Signal Processor (DSP). The experimental test was only performed for an initial displacement of 15mm.

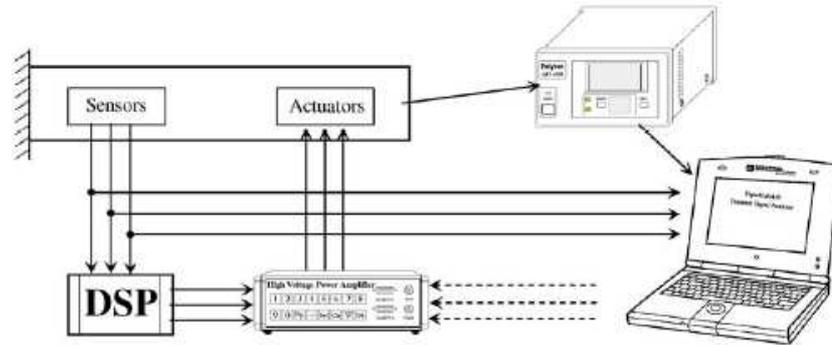


Figure 3.9: Experimental Configuration

(Dong et al. 2006)

### 3.4.1 Results and Conclusions

The numerical analysis resulted in successful control of both the impulse and random noise inputs. Displacement data was plotted for the sensor at the free end of the beam. The figures shown below indicate very efficient vibration suppression.

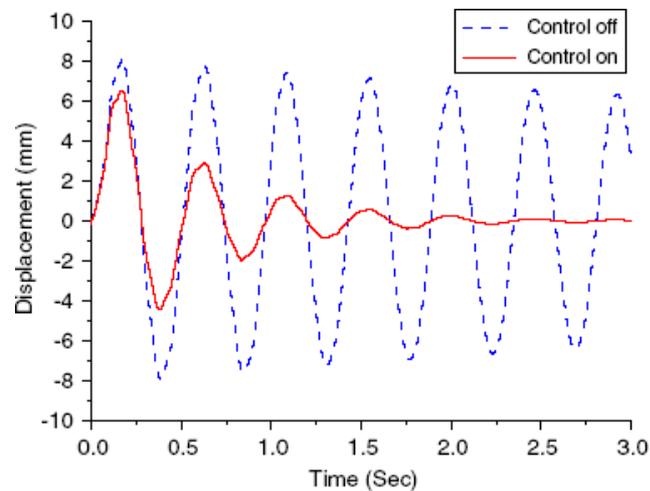


Figure 3.10: Displacement Response to Impulse

(Dong et al. 2006)

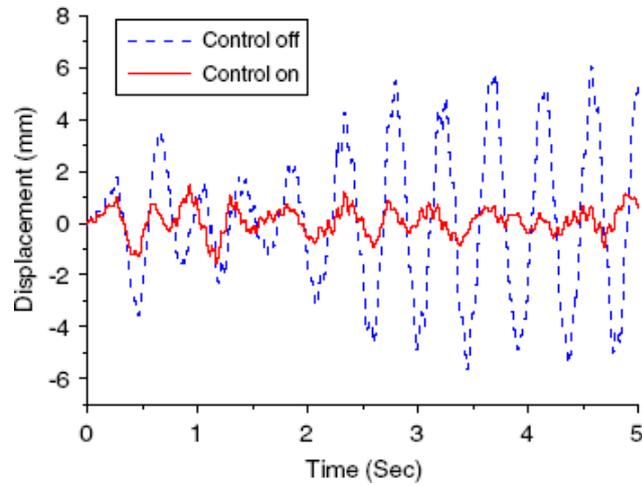


Figure 3.11: Displacement Response to Random Excitation

(Dong et al. 2006)

The experimental results were also impressive but were difficult to compare to the simulated results as different tests were performed for the numerical tests and the real-time implementation. Again, displacement data was plotted for the free end of the beam.

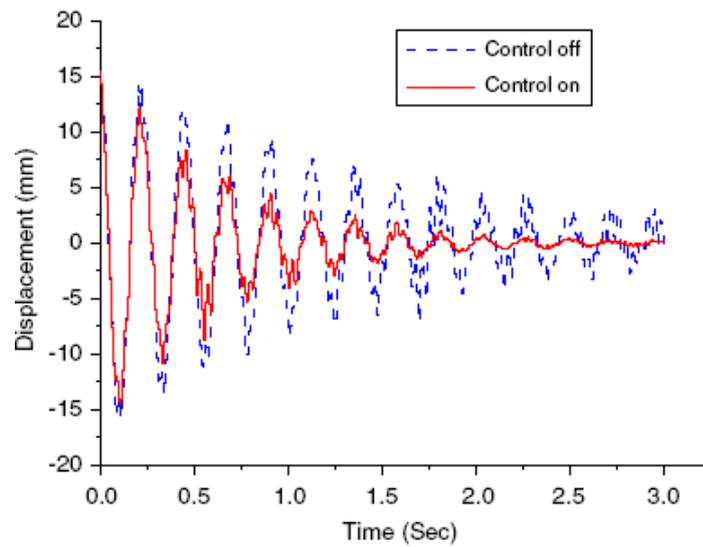


Figure 3.12: Experimental Response to Initial Tip Displacement

(Dong et al. 2006)

### 3.5 Semi-Active Friction Damping

In this example, Dupont, Kasturi & Stokes (1997) performed numerical and experimental tests on semi-active control of friction dampers. They proposed control laws to maximise the instantaneous energy dissipation of the friction dampers. This was done by modulating the normal force at the friction interface. A controller was designed in an effort to prevent stored frictional energy from returning to the system.

Initially, the frictional forces were mathematically modeled to include velocity dynamics. Using this model, Dupont et al. (1997) derived several control laws to maximise energy dissipation at the interface. Three control laws were then simulated and compared using the idealised system shown below.

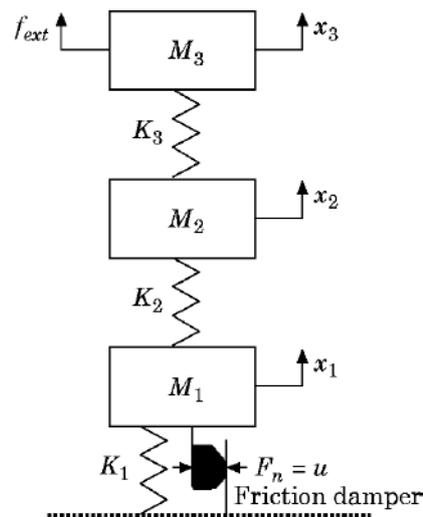


Figure 3.13: Friction Model used for Simulations

(Dupont et al. 1997)

The control laws were then tested experimentally. A double-shear friction device was designed specially for this purpose. The test inputs were provided by a servo-hydraulic Materials Testing Machine (MTM) and the normal force was modulated using piezoelectric stack actuators.

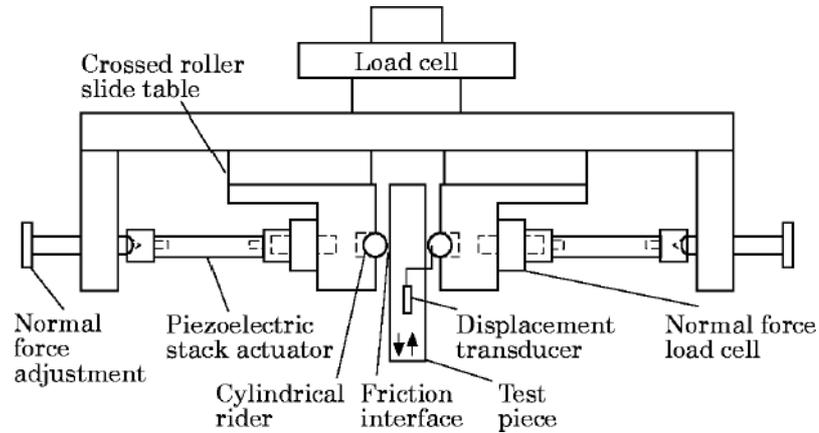


Figure 3.14: Double-Shear Friction Device

(Dupont et al. 1997)

### 3.5.1 Results and Conclusions

The numerical results were obtained using the system shown in figure 3.13. The figure depicted below shows the total system energy dissipation for the three developed control laws. In this case an initial mass displacement was employed as the primary input. The control laws were almost equivalent for large amplitude vibrations, however, the control strategy shown with the solid line was more efficient at suppressing small amplitude vibrations.

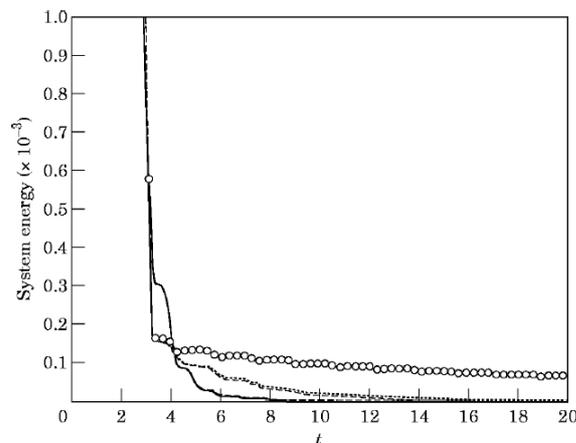


Figure 3.15: Total System Energy for Initial Displacement

(Dupont et al. 1997)

The next plot shows the displacement of mass 3 for the unforced system and with the control schemes in place. The solid line with circles shows the system with purely passive damping and the remaining lines are those with the various control laws in place. The plot clearly shows improved energy dissipation for semi-active control.

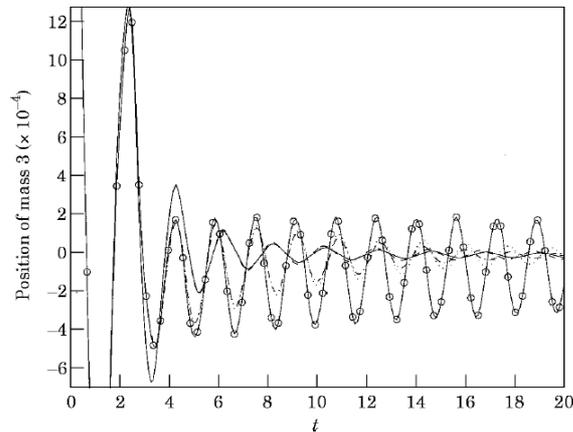


Figure 3.16: Mass 3 Position for Initial Displacement

(Dupont et al. 1997)

Results for the experimental analysis are shown in the following figure. The hysteresis loops are for a displacement amplitude of  $2\mu\text{m}$ . Again, the plot reveals improved energy dissipation when the system was under control.

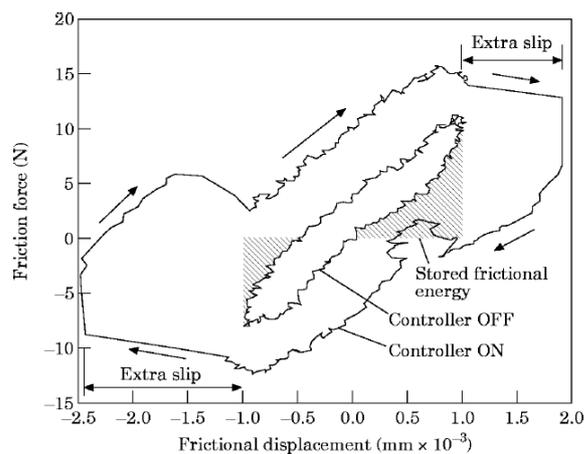


Figure 3.17: Improved Energy Dissipation with Control

(Dupont et al. 1997)

### 3.6 General Conclusion of the Research

The first three chapters have provided a brief review of active vibration control and the current state of practice. It is difficult if not impossible to make any direct comparisons between any of the presented methods of vibration control. The reasons for this are, firstly, because each system was tested on different physical structures and secondly, because various methods of input disturbance were employed. Inputs ranged from tip displacements to resonant frequency testing. The most poignant fact to note is that all of the presented control systems were very effective in suppressing vibration.

The most valuable aspect of the literature review was the general knowledge and insight gained in the area of vibration control. This knowledge provided a foundation for the upcoming practical experimentation.

## Chapter 4

# Preliminary System Design

## 4.1 Introduction

This chapter signifies the end of the research aspect of the project and the beginning of the practical experimentation. It is dedicated to the initial design work of the vibration control test system. The first section details the intended physical test system. This is followed by a brief description of the desired digital system. The remainder of the chapter is committed to the modal analysis and MATLAB simulations.

## 4.2 The Physical System

This section details the physical test system. The methods by which the various components were chosen are presented. As explained, the major limitation in the component choice was resource availability.

### 4.2.1 The Vibrating Structure

The design aspect of this project required the construction and testing of a prototype vibration suppression system if resources permitted. A simple cantilever beam was specified as the physical structure to be controlled. This was arbitrarily chosen to be a 200mm length of annealed aluminium alloy with a rectangular cross-section of 35mm x 2mm. It had a Young's modulus of 69GPa, Poisson's ratio of 0.33 and a density of  $2700\text{kgm}^{-3}$ .

### 4.2.2 The Physical Control Strategy

There are many methods of controlling a vibrating structure, some of which are detailed in the previous chapter. The control strategy for this project was chosen by considering the ease of implementation, availability of equipment and estimated performance. Again, this choice was limited by the project specifications, which required an adaptive feed-forward implementation.

These considerations led to the proposal of a semi-active eddy current damping system. In this system, a magnet is moved in opposition to conductors attached to the vibrating structure such that the conductors cut the perpendicular components of the permanent magnetic field. The magnet's motion is provided by an electromagnetic shaker which is driven by the controller's output signal. This allows the generation and subsequent dissipation of eddy currents in the conductor, resulting in a viscous damping effect. The major advantages of this system are firstly, its simplicity and the requirement of readily obtainable equipment such as permanent magnets and copper plates. Secondly, bounded input, bounded output stability is guaranteed as no mechanical energy is added to the system. Finally, the use of eddy currents in vibration suppression is a new and relatively unexplored concept. The following diagrams show the basic setup of the eddy current damper.

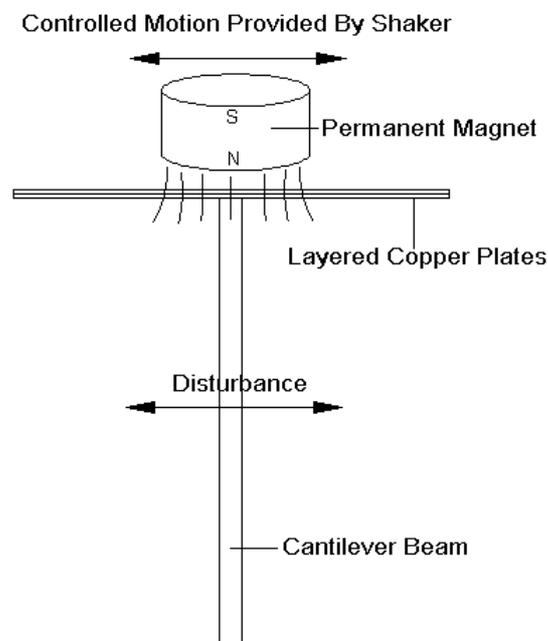


Figure 4.1: End View of the Proposed Eddy Current Damper

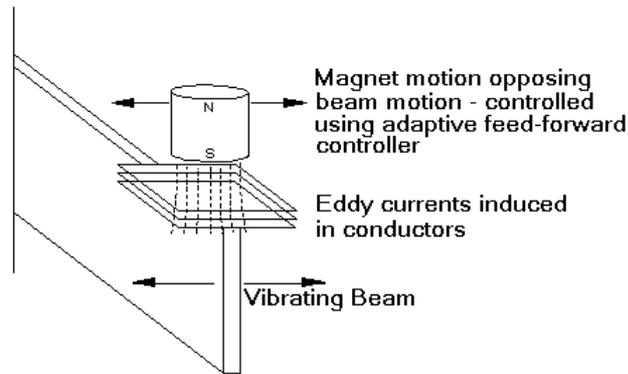


Figure 4.2: Isometric View of the Proposed Eddy Current Damper

### 4.2.3 The Sensors

Adaptive feed-forward systems require sensors to measure the vibration of the structure. The primary input and the error signal need to be measured and fed to the controller. Accelerometers were chosen for this task. The reason for this choice was simply due to availability.

## 4.3 The Digital System

All filter-based control systems require some form of digital processor. This is the heart of the filter. It is used to update the adaptive algorithm and derive the control signal. Several options were considered and analysed according to cost, availability and expected performance.

The first option was an ANALOG DEVICES SHARC EZ KIT digital signal processor. This device is highly specialised for applications such as filtering where fast, repetitive calculations are required. It can be programmed in C or its own assembly language and thus is versatile in this sense. The limitation of this device is that it only possesses one input channel. The proposed control system requires separate input of the primary

disturbance and error signal and thus the SHARC was disregarded.



Figure 4.3: ANALOG DEVICES SHARC EZ KIT

The next device to be considered was the Microchip dsPIC30F4013. This is a single-chip digital signal PIC with several input and output channels, inbuilt ADCs and DACs and its own onboard program memory. This device can also be programmed in C, however it required a costly specialised programmer and thus it too was disregarded.



Figure 4.4: Microchip dsPIC30F4013

The use of a PC and an I/O device was then considered. The first I/O device to be examined was an ELEXOL USB I/O 24R external COM port. This device allows a direct connection to a USB port of the PC and has 24 programmable pins for either input or output. The shortcoming of this external port is that it cannot output analogue signals.

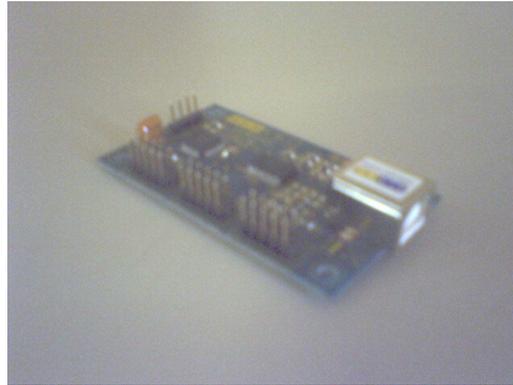


Figure 4.5: ELEXOL USB I/O 24R

This led to the investigation of an AXIOM AX5412 data acquisition card. A PC and the AXIOM card were chosen as the processing and I/O components of the system. This is because the card was available for loan and a device driver for a similar CONTEC card was also available as a programming reference. Refer to Appendix D for a complete list of the device specifications.

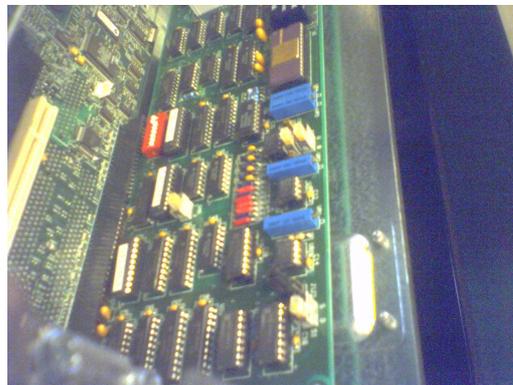


Figure 4.6: AXIOM AX5412 Data Acquisition Card

This choice of a PC and data acquisition card led to another consideration, that of the PC's operating system. Windows98 SE was selected due to the fact that it allows direct access to the computer's physical ports and communication devices.

## 4.4 Modal Analysis

As mentioned previously, a modal analysis of the physical structure allows the control system designer to ascertain the worst case scenarios of the system. It allows the natural modes of vibration and resonant frequencies to be determined. The control system can then be tested over these frequencies so as to gain a complete picture of the performance of the system.

### 4.4.1 Analysis Procedure

In this project, the FEA software, ANSYS, was chosen as the method for the modal analysis. The primary reason for this choice was the availability of the software at the university. Not only that, ANSYS allows a modal analysis to be performed without any detailed knowledge of structural dynamics. An important consideration of finite element analysis is that of mesh sensitivity. Many FEA simulations are sensitive to the nature in which the structure is meshed, however, in this case this was not a big concern due to the simple geometry of the cantilever beam.

The modal analysis was performed in the following manner:

1. Beam modeled and physical properties entered
2. Beam meshed using brick-shaped elements
3. Simulation performed for first ten modes
4. Resonant frequencies and mode shapes recorded

The analysis was performed for the first ten mode shapes above an excitation frequency of 50Hz. This lower frequency limit is a reasonable assumption for common mechanical vibration.

#### 4.4.2 Modal Analysis Results

The table depicted below shows the resonant frequencies at which the various mode shapes were excited. This is followed by figures which portray the animated mode shapes.

Md1	Md2	Md3	Md4	Md5	Md6	Md7	Md8	Md9	Md10
75.9	76.5	112.1	114.5	123.4	152.4	161.5	198.4	200.5	217.2

Table 4.1: Resonant Frequencies for First Ten Mode Shapes

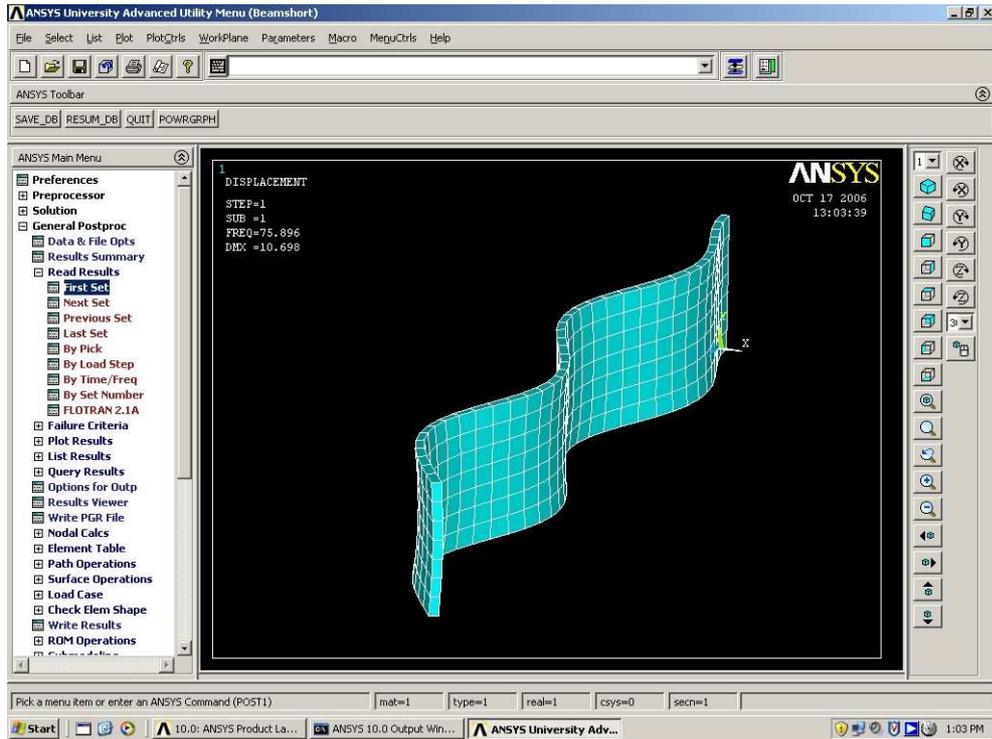


Figure 4.7: Mode 1

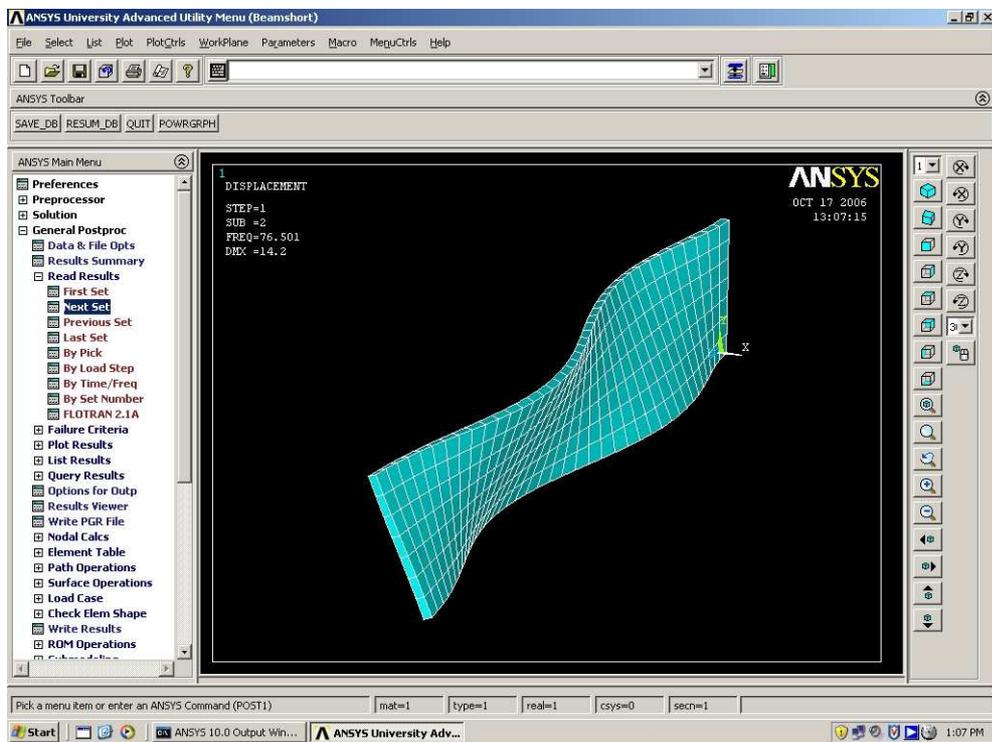


Figure 4.8: Mode 2

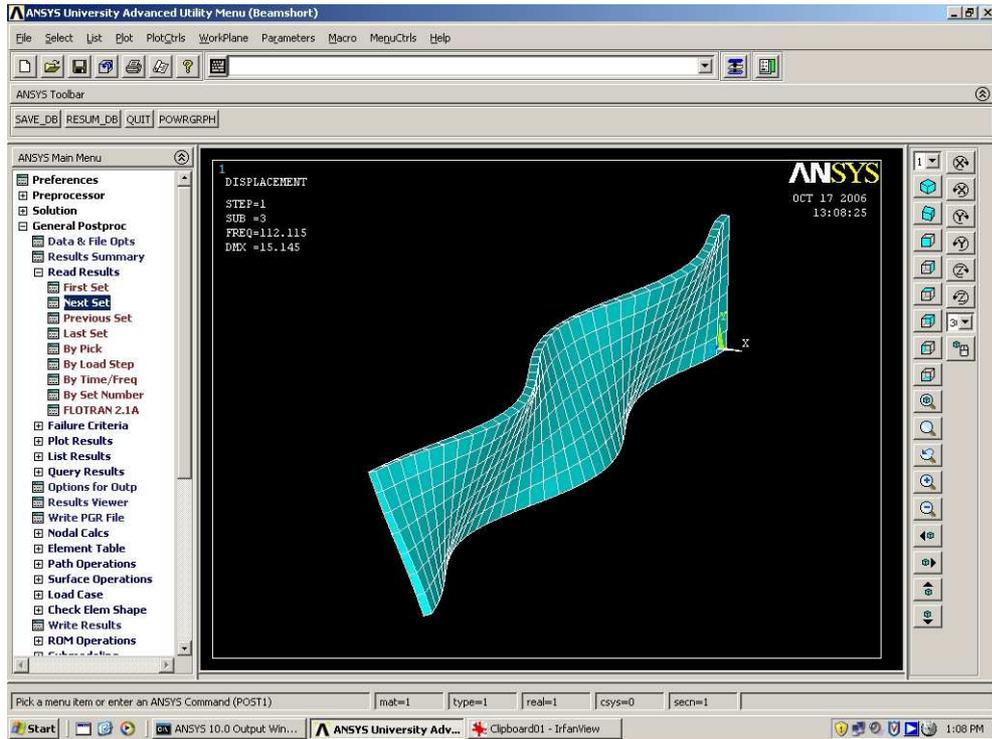


Figure 4.9: Mode 3

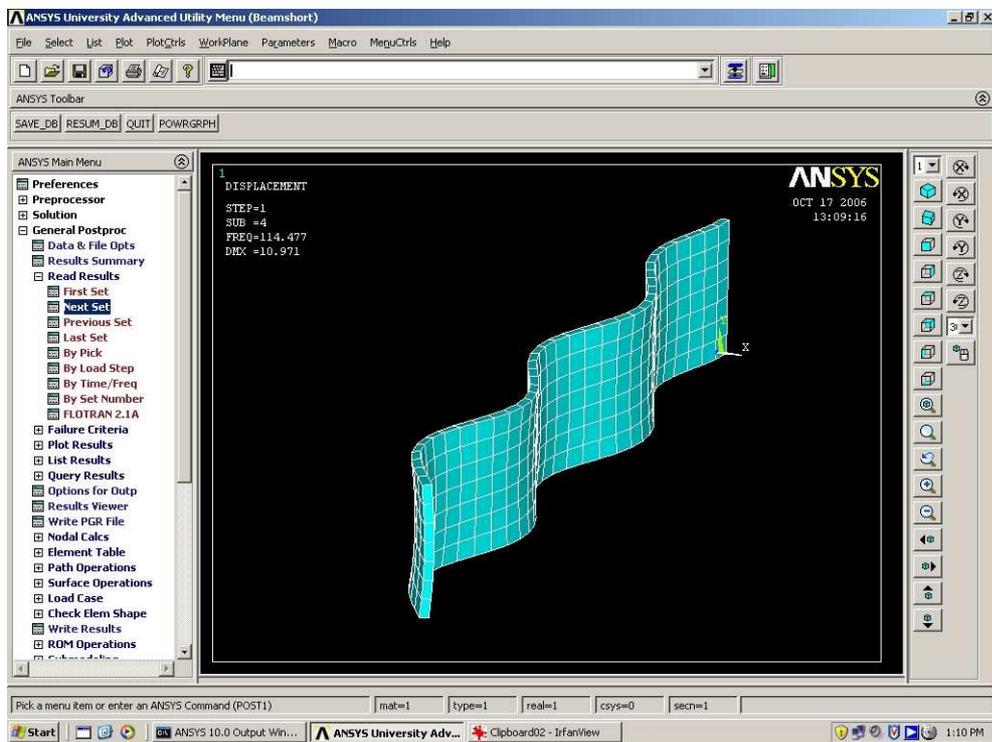


Figure 4.10: Mode 4

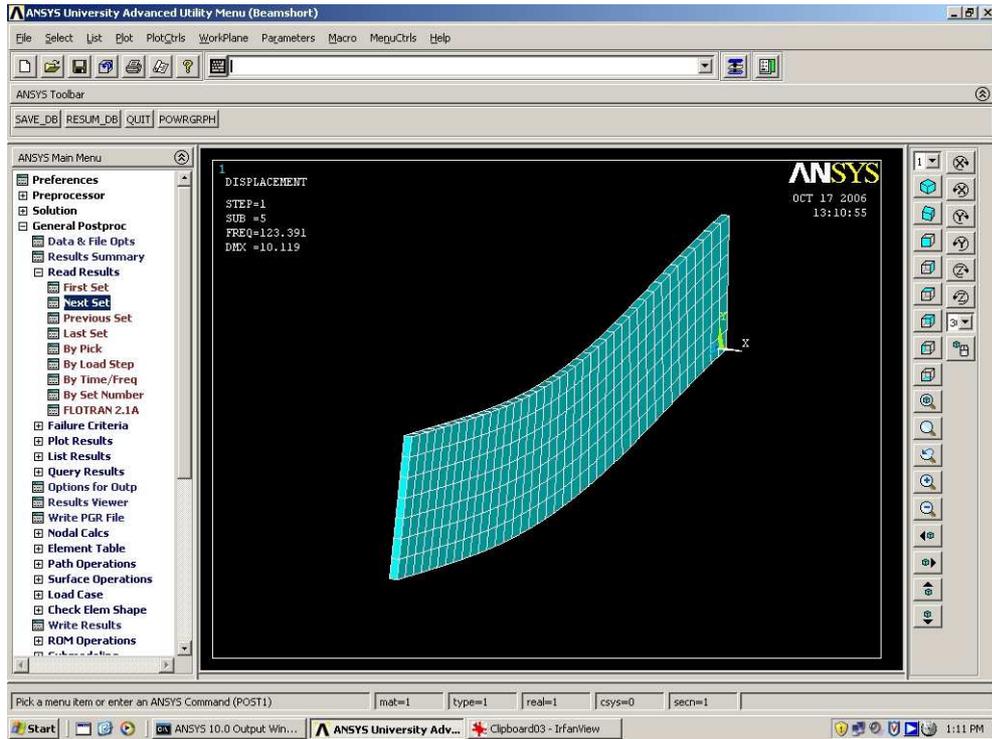


Figure 4.11: Mode 5

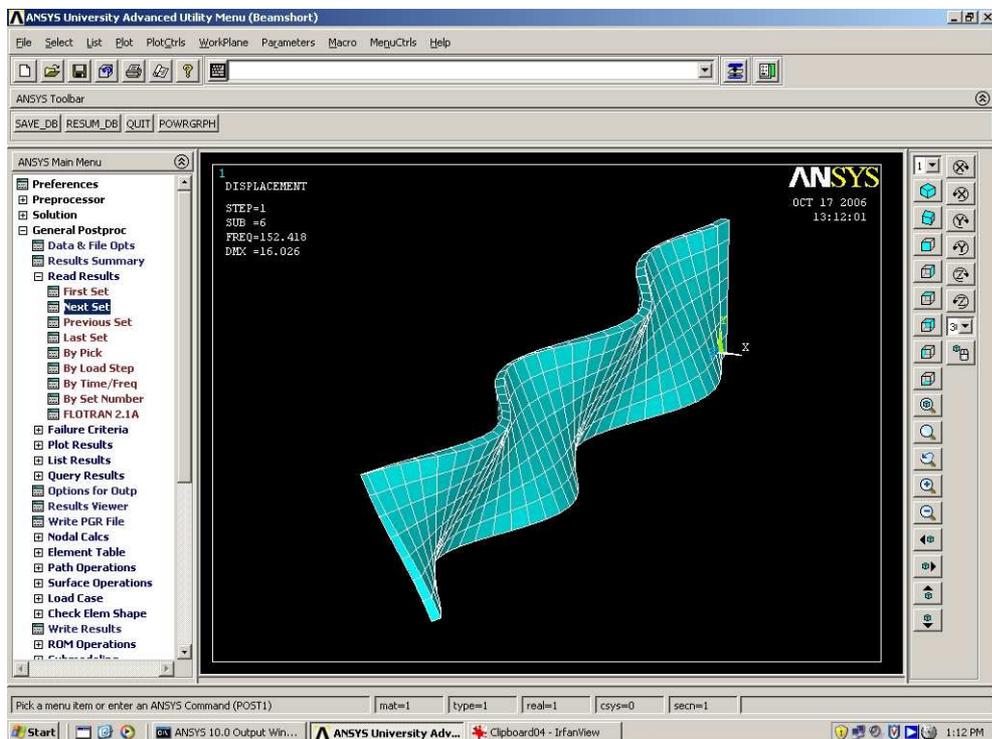


Figure 4.12: Mode 6

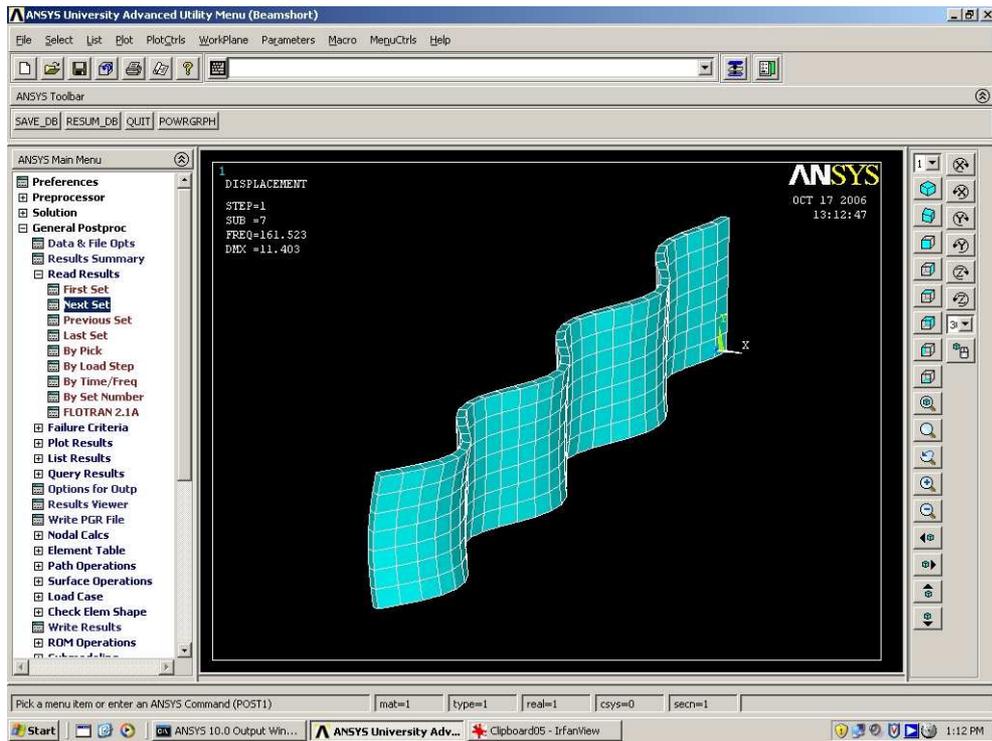


Figure 4.13: Mode 7

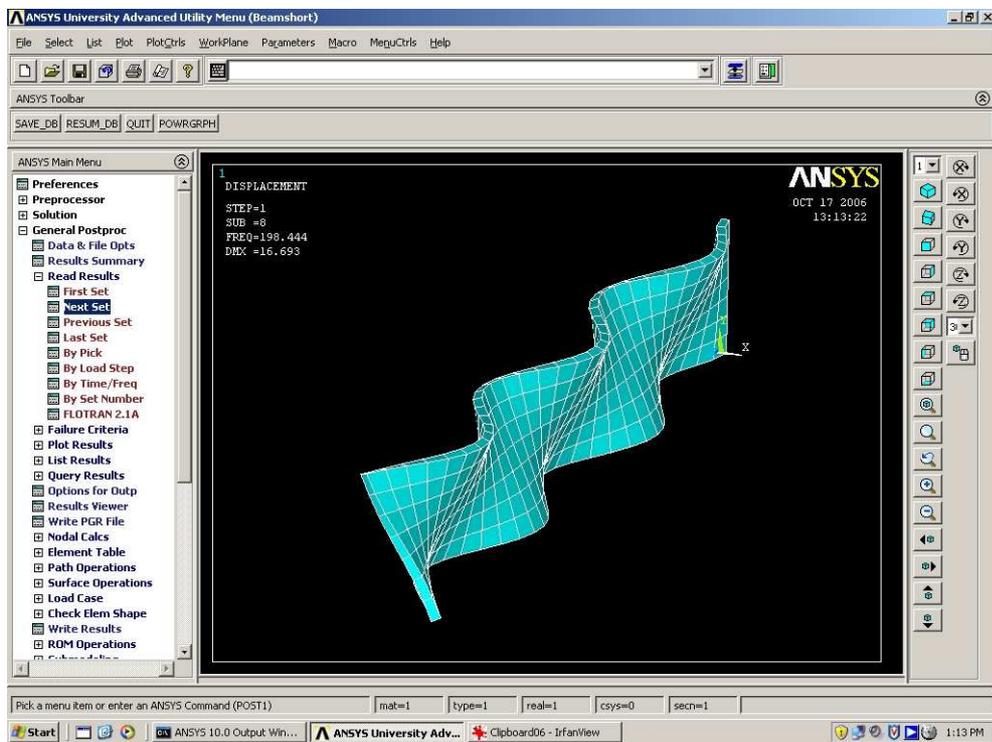


Figure 4.14: Mode 8

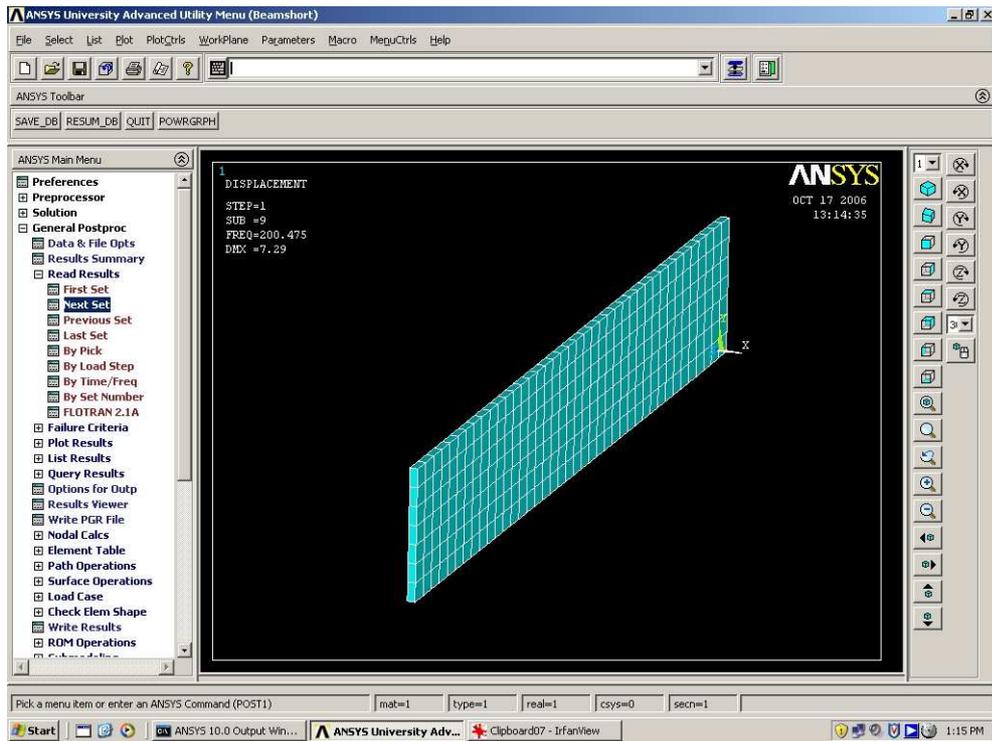


Figure 4.15: Mode 9

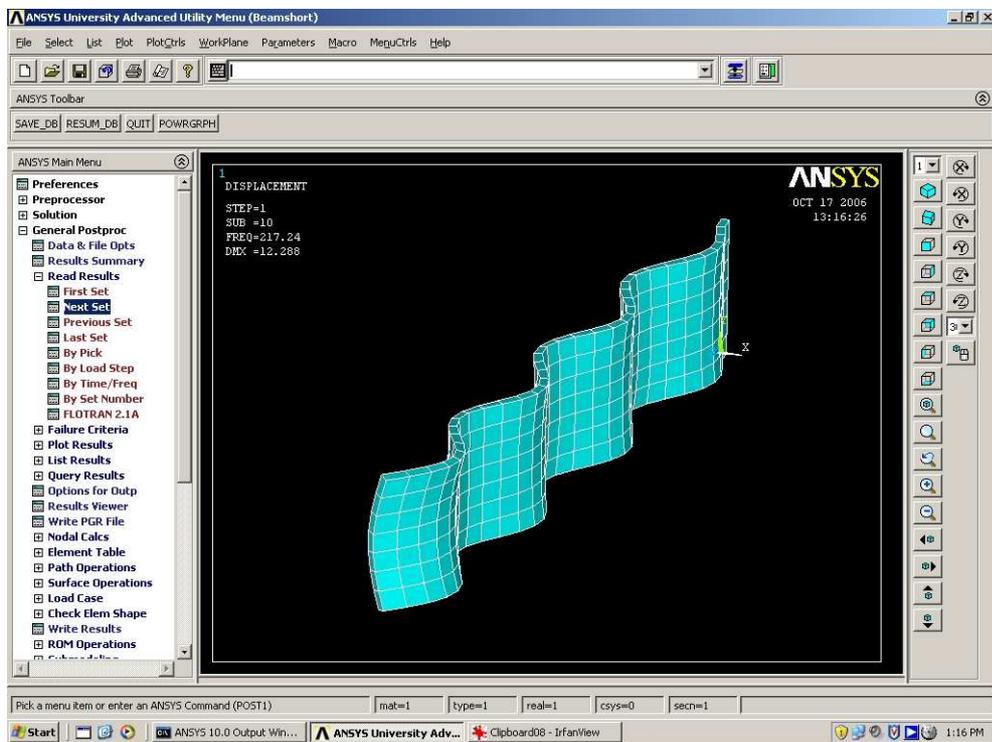


Figure 4.16: Mode 10

### 4.4.3 Conclusion

Although modal analysis provides a set of discrete mode shapes and frequencies, in practice it is unlikely that only one particular mode shape will be excited at any given time. The response of the beam is likely to be a combination of several modes of vibration. In reality, the vibration source often contains many frequency components, each exciting different modes of vibration.

The proposed control system design requires lateral motion at the tip of the beam. Modes 5 and 9 do not involve any lateral motion and thus if either of these modes are dominant, the damping will be predominantly passive. This is because the copper plates will still be subject to a time changing magnetic field, resulting in a passive damping effect, however, the magnet can not be moved in opposition to these motions and thus the active component will be lost.

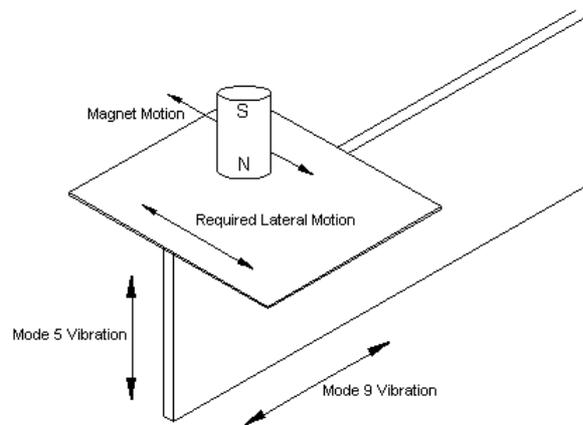


Figure 4.17: Non-Lateral Motion of Modes 5 and 9

The solution to this problem is simple. In practice, the dominant frequency of the input disturbance could be measured and checked to see if it is close to either of these resonant frequencies. If so, the orientation of the copper plates and magnet could be changed so as to suppress these modes of vibration. Passive eddy current dampers could then be included to suppress other small amplitude vibrations.

## 4.5 MATLAB Simulations

The project specification required the development and simulation of one particular adaptive feed-forward algorithm. The LMS algorithm, detailed in Chapter 2, was chosen as the control algorithm for this project. This was chosen due to simplicity and ease of implementation. As well as that, the LMS algorithm is well documented and is a very common noise reduction algorithm. The source code for these simulations can be viewed in Appendix B.

### 4.5.1 Simulation Procedure

A MATLAB function was written to perform the simulations. The function required the user to enter the amount of filter taps to be used and the rate at which the algorithm was to converge. The user then had to choose the type of reference signal, harmonic, harmonic with noise or just noise. The calculations were then performed for a signal length of ten thousand samples and the error and control signals were plotted. This procedure allowed the filter length and convergence coefficient to be varied on each run.

### 4.5.2 Simulation Results

Initially, it was discovered that the algorithm's worst performance was for a harmonic signal with some added noise. This shown in the following figures. The control (blue) and error (red) plots are depicted for the three reference signals. The filter length is 10 and the convergence coefficient,  $\mu$  is 0.001.

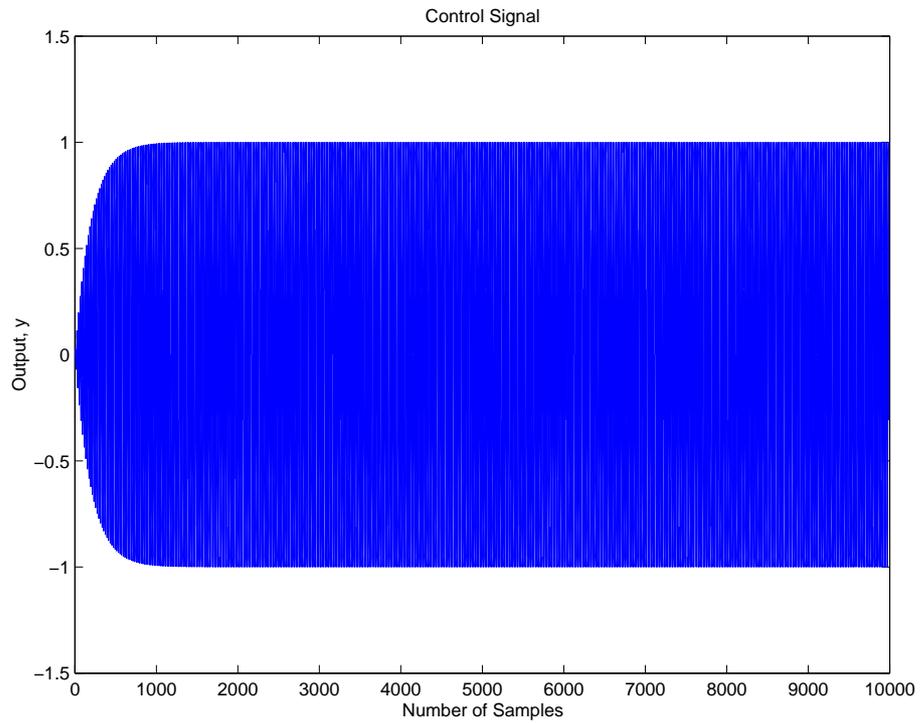


Figure 4.18: Ctrl Signal for Harmonic Input, Filter Length = 10,  $\mu = 0.001$

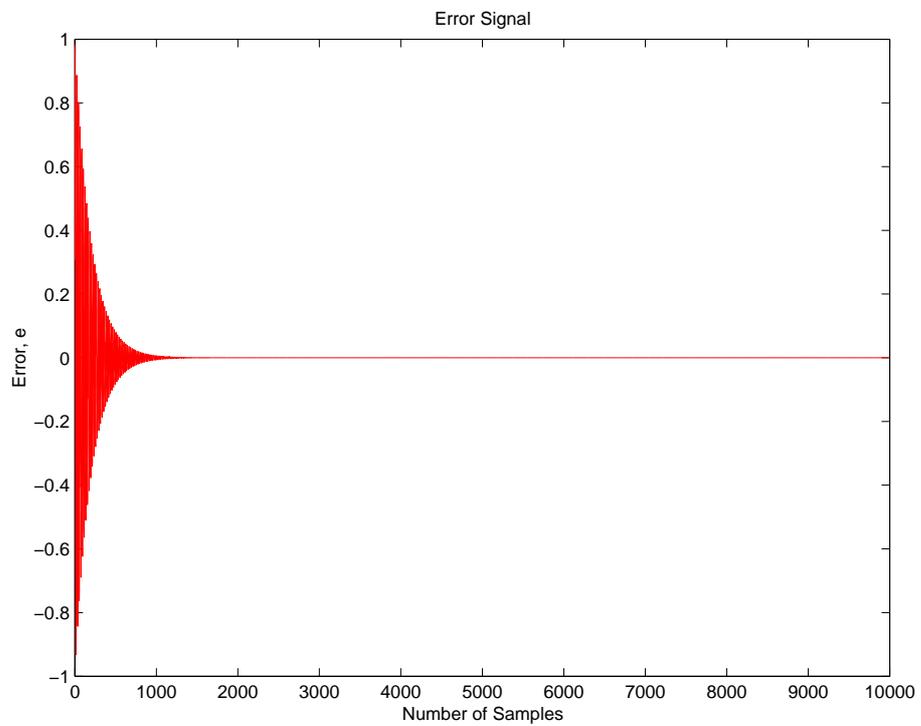


Figure 4.19: Error Signal for Harmonic Input, Filter Length = 10,  $\mu = 0.001$

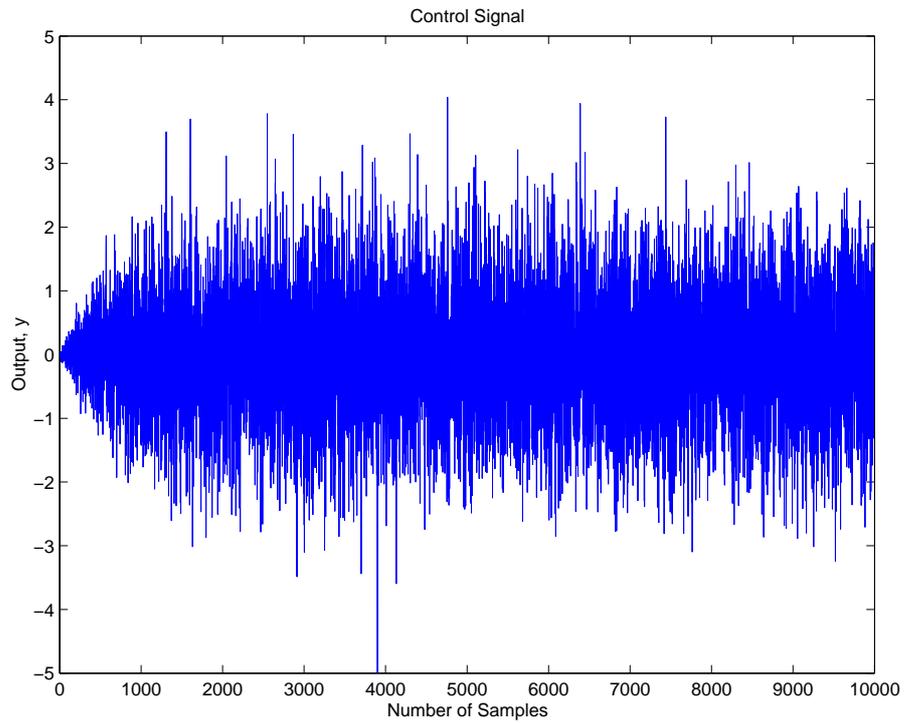


Figure 4.20: Ctrl Signal for Gaussian Noise Input, Filter Length = 10,  $\mu = 0.001$

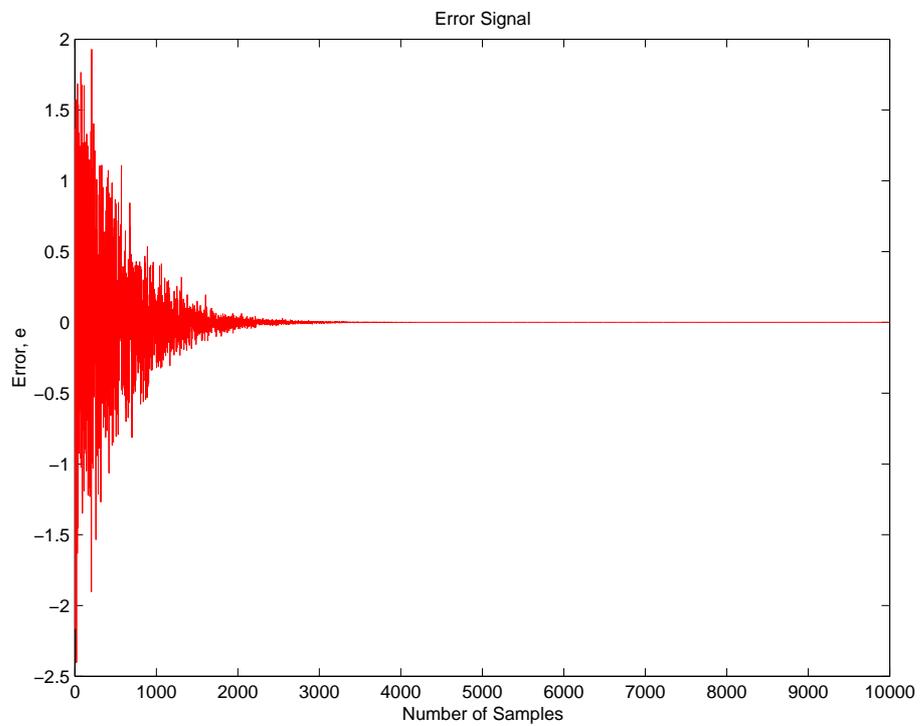


Figure 4.21: Error Signal for Gaussian Noise Input, Filter Length = 10,  $\mu = 0.001$

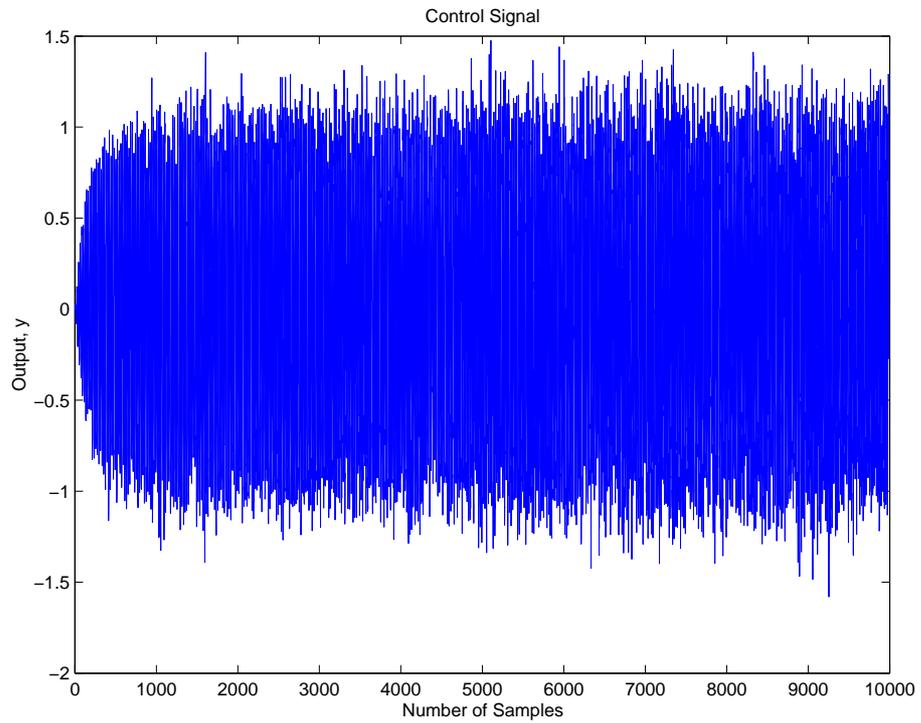


Figure 4.22: Ctrl Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.001$

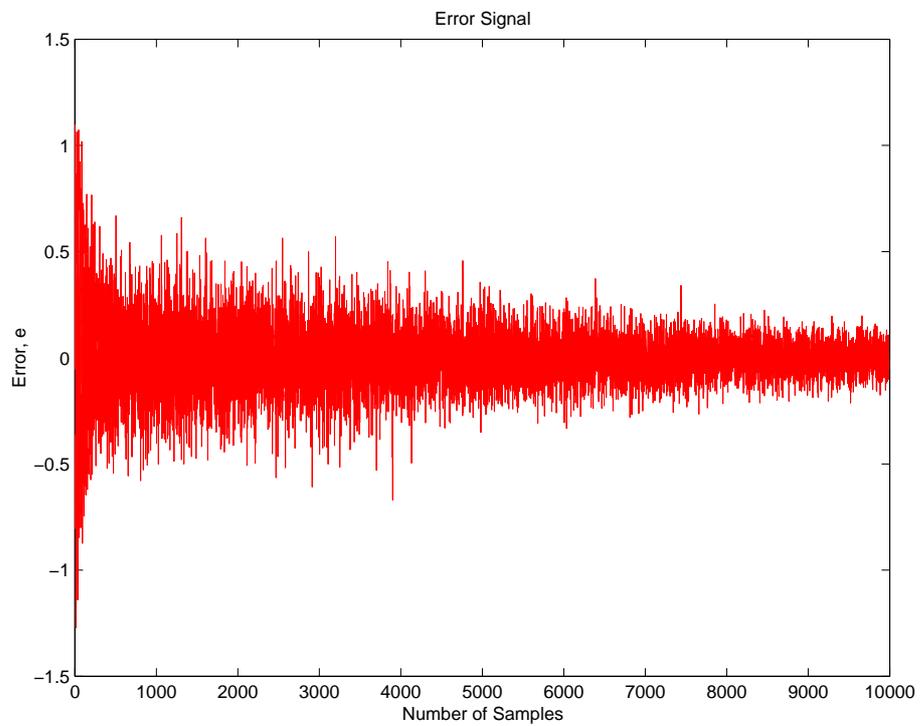


Figure 4.23: Error Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.001$

The error signal for the third case had not converged at ten thousand samples. This led to further testing of the noisy harmonic signal. The filter length and the convergence rate were varied in attempt to increase the algorithm performance. The results of these tests are portrayed below.

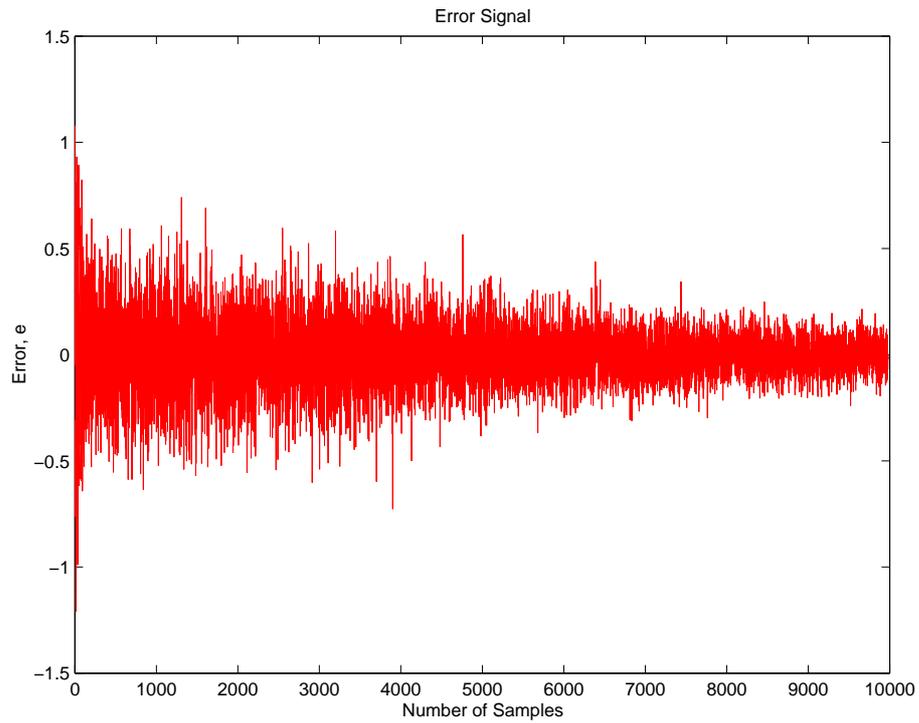


Figure 4.24: Error Signal for Harmonic + Noise Input, Filter Length = 20,  $\mu = 0.001$

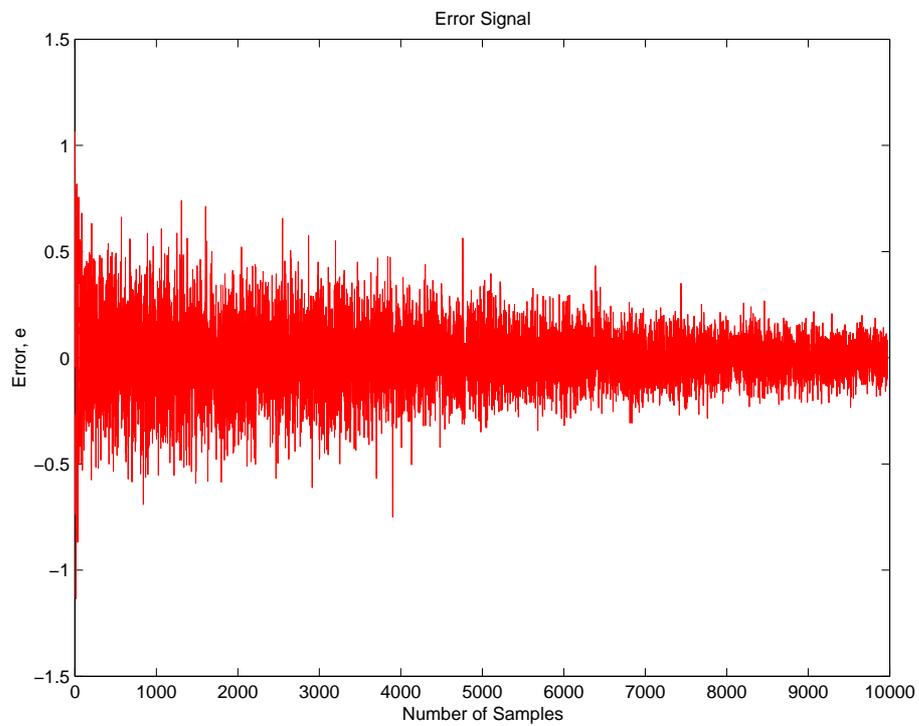


Figure 4.25: Error Signal for Harmonic + Noise Input, Filter Length = 30,  $\mu = 0.001$

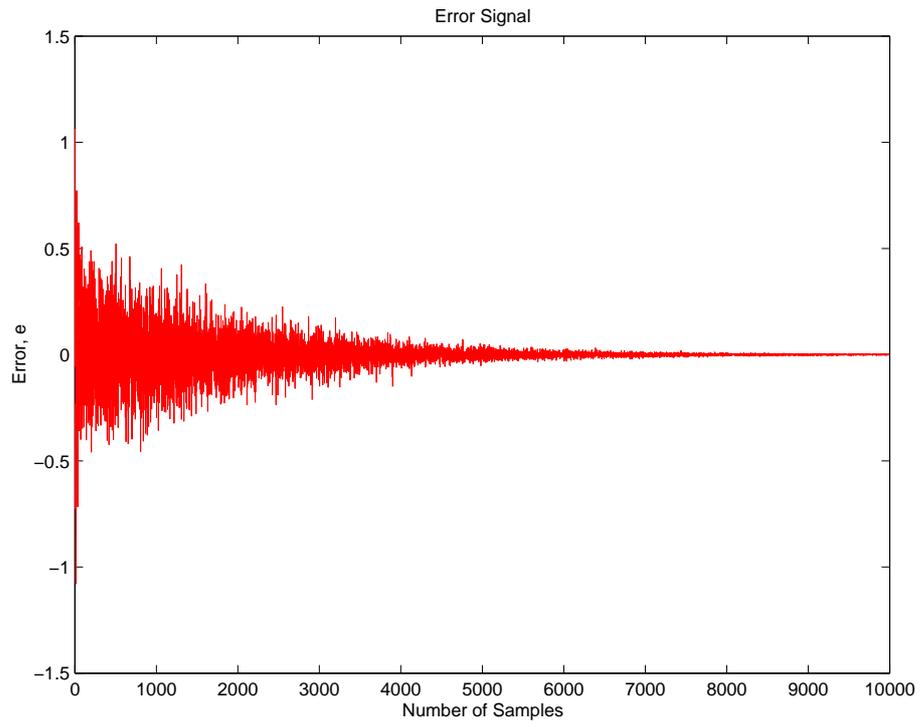


Figure 4.26: Error Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.004$

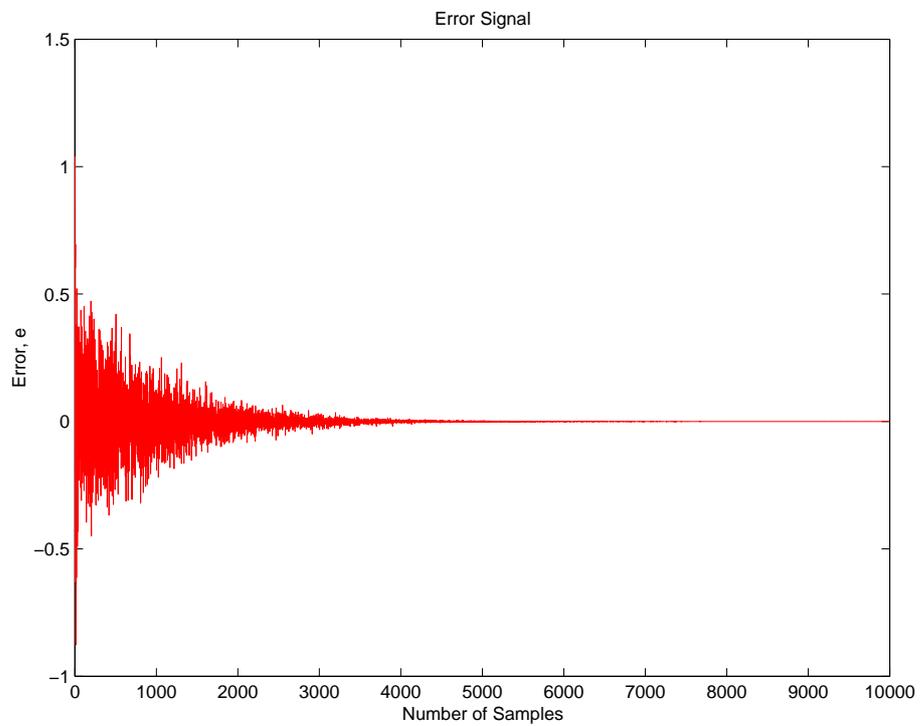


Figure 4.27: Error Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.008$

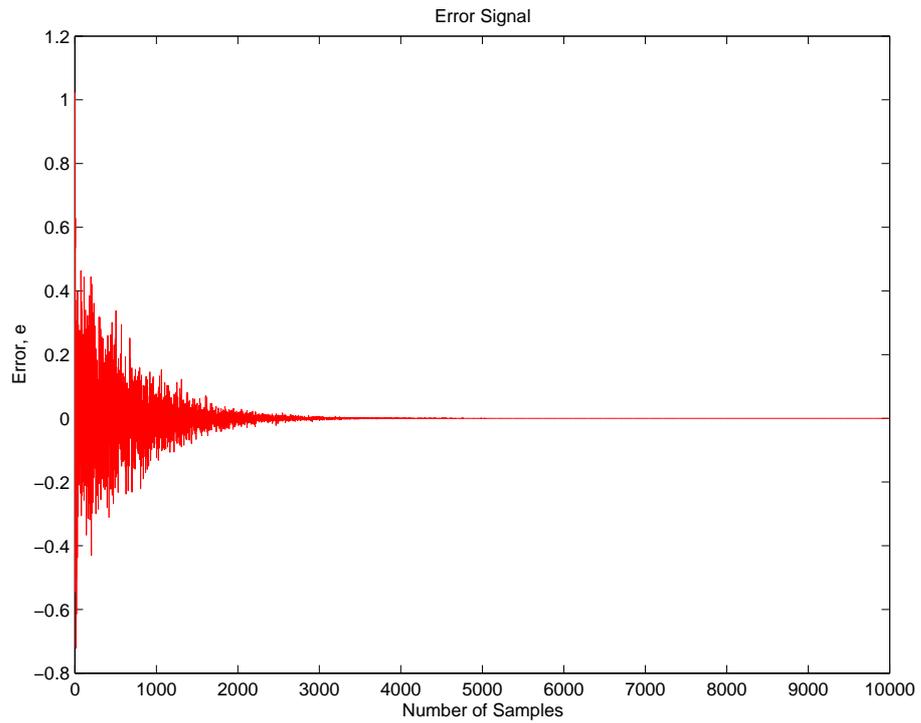


Figure 4.28: Error Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.012$

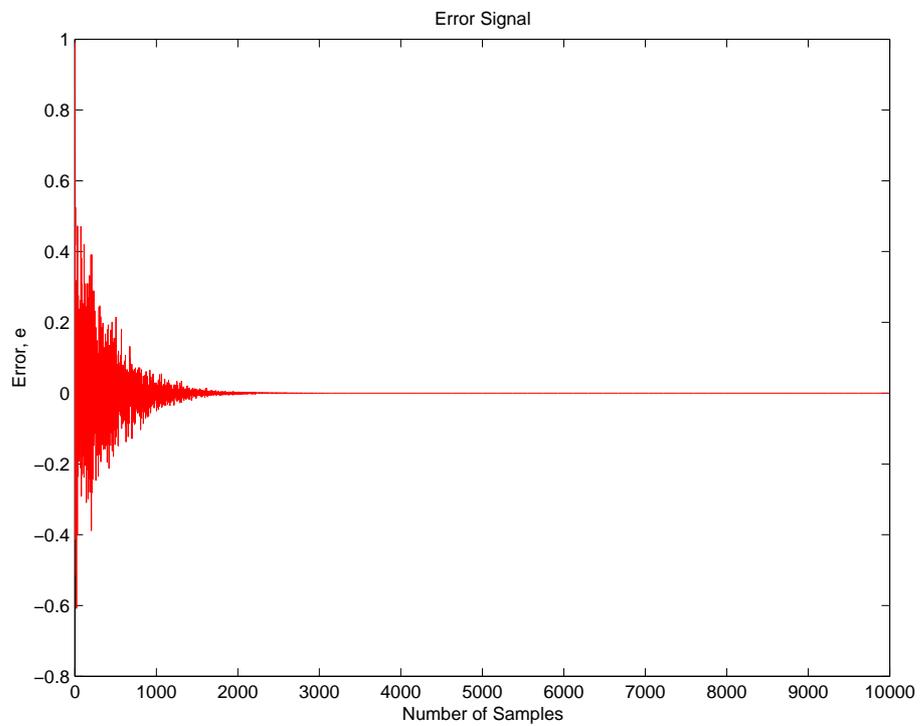


Figure 4.29: Error Signal for Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.020$

Initially, the filter length was increased to twenty in (a) and thirty in (b) while  $\mu$  remained at 0.001. The performance of the algorithm degraded slightly with the increase in filter length. For the remaining tests, (c) through (f), the convergence coefficient was gradually increased and the filter length was kept constant at ten. This increase in  $\mu$  allowed the filter to converge over 2500 samples.

It was then discovered that the filter would become unstable if the convergence coefficient was too large. This is shown in the following plots where  $\mu$  is 0.5.

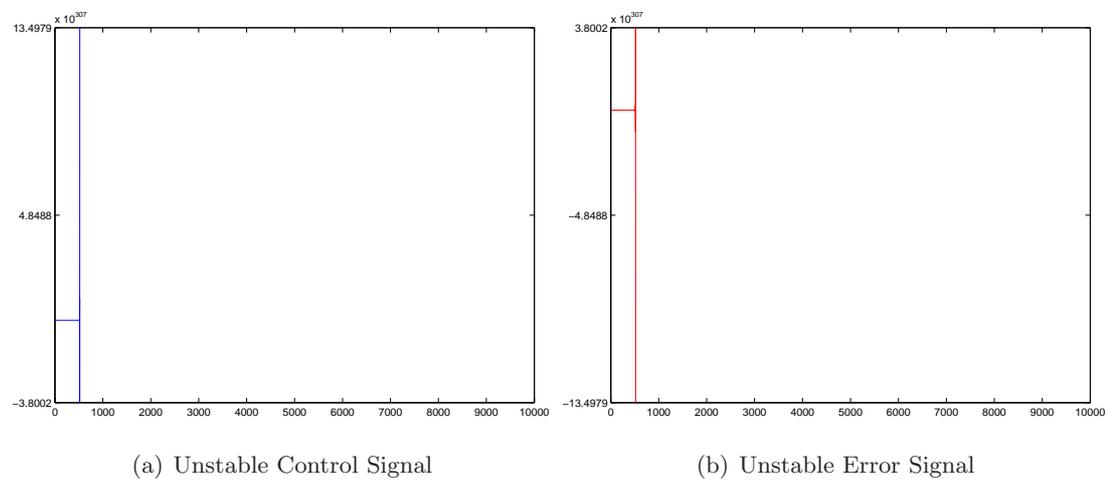


Figure 4.30: Harmonic + Noise Input, Filter Length = 10,  $\mu = 0.5$

### 4.5.3 Conclusion

The performance of the LMS algorithm varied substantially as the filter length and convergence coefficient were changed. A large filter length caused the reference signal to be over-modeled and thus decreased the performance of the algorithm. A high convergence rate allowed the error to be minimised quickly but instability occurred if  $\mu$  was too large.

Considering that, it should be noted that these simulations were performed for an ideal situation as there were no phase changes and no noise was added to the reference signal

after sampling. In practice, an optimum filter length would have to be found for the particular application. The filter length would have to be long enough to allow the filter to model any phase changes and added noise, however, it should not be too long as to cause over-modeling or cause the filter to become computationally expensive. As well as that, the convergence coefficient,  $\mu$  should be set high enough to allow the filter to converge quickly but not too high as to induce unstable behavior.

## Chapter 5

# Final Design and Real-Time Testing

## 5.1 Introduction

This chapter details the final control system design, the testing procedure and the results obtained. Initially, the physical test system is discussed and a considerable design change is presented. This is followed by an analysis of the C language programming where a program flowchart is presented as a means of portraying the final LMS filter design. The testing procedure is then described and a solution to an unexpected design flaw is portrayed. Finally, the chapter concludes with the results obtained from the real-time testing.

## 5.2 The Physical System

Initially, the control system was constructed as detailed in the previous chapter. In this design, the copper plates were fixed at 90 degrees to the beam and an electromagnetic shaker was used to move the permanent magnet. This setup would have been optimal, however, the electromagnetic shaker failed during the initial testing stage. This was probably due to off-centered vibration caused by the weight of the magnet. This failure led to a modified system design in which a car speaker was adapted so as to provide the permanent magnet motion. Because of this modification, the orientation of the conducting plates had to be changed. The first of the following figures shows the design modifications and the remaining two show the completed construction.

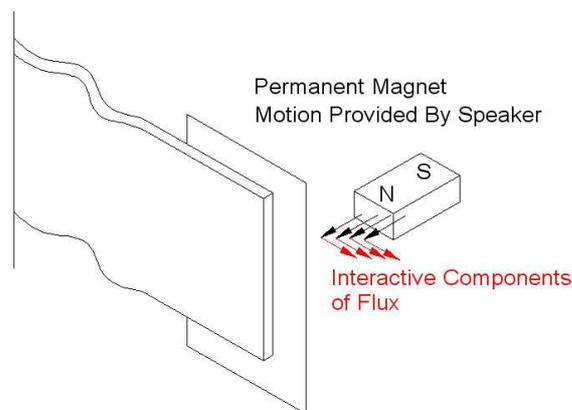


Figure 5.1: The Design Change

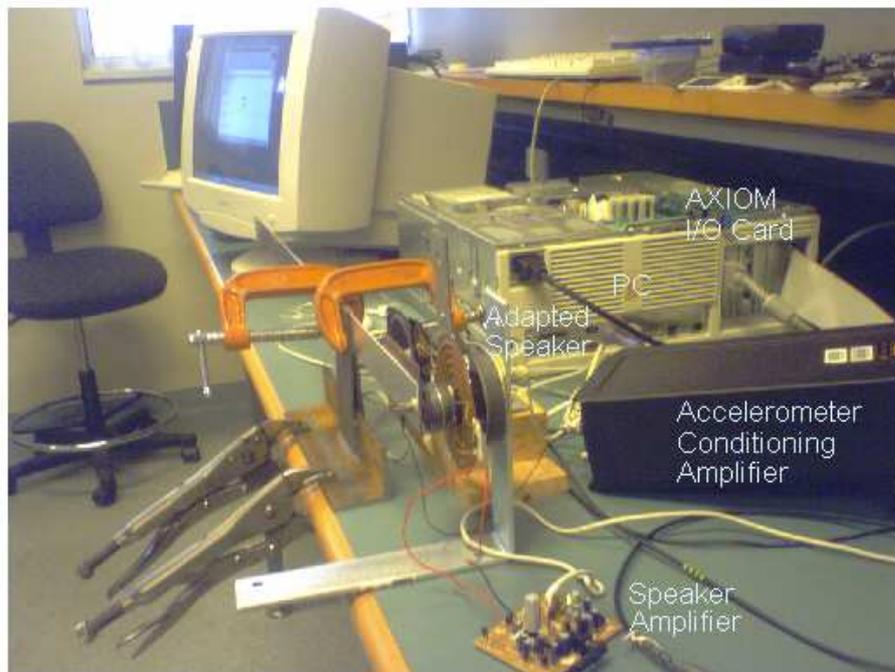


Figure 5.2: The Whole System

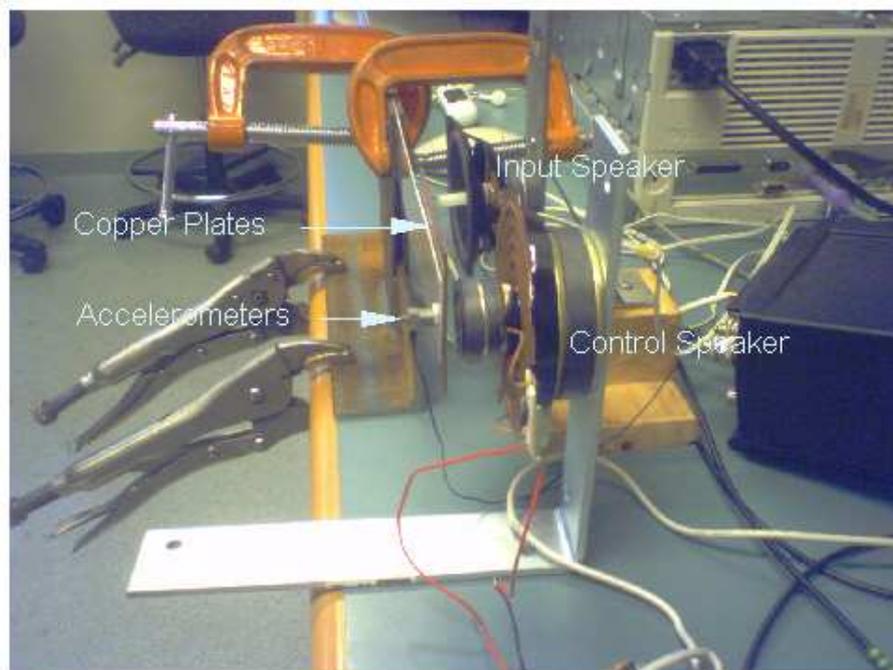


Figure 5.3: The Eddy Current Damper

## 5.3 Program Development

The LMS algorithm was developed during the construction of the test system. C was chosen as the programming language because it was well suited to the application. This is because it is a fairly low-level language that allows for efficient coding and simple I/O routines. The AXIOM data acquisition card was set up prior to programming. This involved setting several switches and jumpers on the card. A base address of 0x300 was chosen, differential mode sampling was selected so as to reduce the open-loop noise and finally, the card was set to operate in bipolar mode so as to allow for negative voltages.

Initially, relevant register addresses were defined for the card. These included the ADCs result register, the DACs conversion register, a status register and a gain register. A read function was then written. This allowed the reference and error signals to be sampled and converted into digital format. This was followed by a write function that was used to output the control signal to a DAC for conversion. These read and write functions utilised four simple 8-bit functions for reading and writing a byte of data from/to a specific memory location. These functions were located in a pre-written assembly I/O library, `ioportlib.asm`.

After the I/O routines, the LMS filter was written. The filter was based on an eternal loop and did not use interrupts for sampling and outputting signals. It was decided not to use interrupts on the basis that the mechanical frequencies were relatively low and thus did not require extremely accurate sampling. The flowchart shown below details the LMS filter. A full code listing of the C program and I/O library can be viewed in Appendix C.

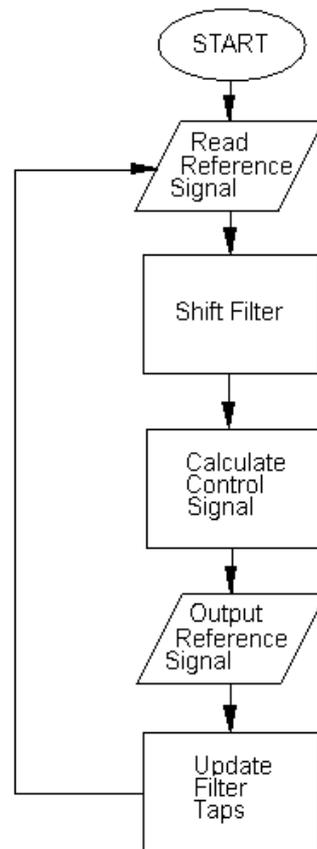


Figure 5.4: The LMS Filter

## 5.4 Testing Procedure

As explained previously, the intended testing procedure was to input the beam's first ten resonant frequencies above 50Hz. The error signal was to be recorded with and without control. This would have indicated the effectiveness of the control strategy at its worst case conditions.

Initially, an input speaker was modified to provide this input. A perpendicular arm was attached to the base of the speaker and to the base of the beam. It was intended to output the pure sinusoids to the speaker from the PC by means of a tone generator. This setup is depicted in the following figure.

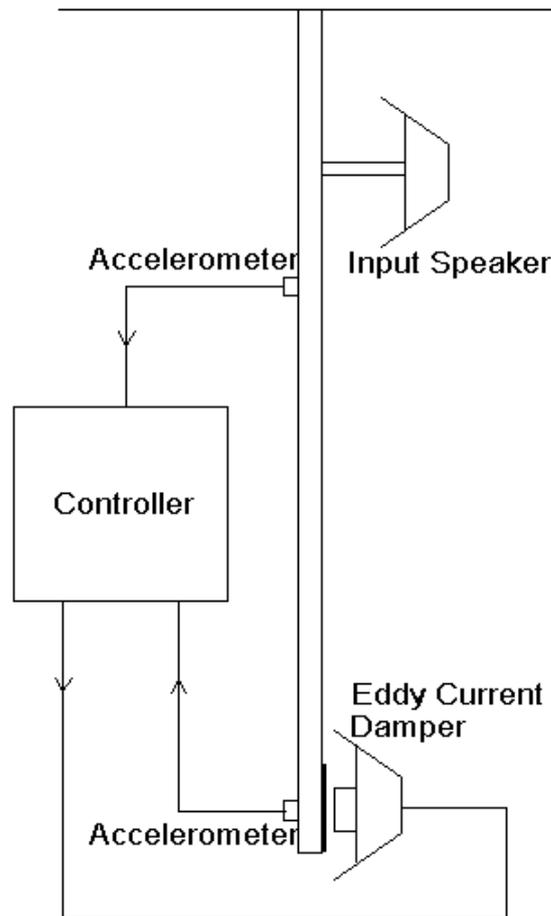


Figure 5.5: The Intended Input

Preliminary testing proved this input method to be ineffective. The speaker was not able to induce vibrations of sufficient amplitude. A larger speaker was then tested but it also lacked sufficient output power. This led to a new testing method which involved tip displacements. The tip of the beam was displaced 4mm and allowed to vibrate naturally whilst the error signal was recorded with a digital oscilloscope.

It was also discovered that the distance between the magnet and conducting plates had a direct impact on the performance of the system. Repeated testing revealed that the optimum gap was about 4mm. This distance was employed for the recorded tests.

Initial testing also involved the manual optimisation of the adaptive filter. The filter length and convergence coefficient were varied until an optimum configuration was found. Thirteen tests were performed in total, five with semi-active control, five without control and three with passive control.

## 5.5 Results

The error signals were stored as comma separated files by the oscilloscope. These files were then examined so as to accurately determine the settling time of the beam. The error signals were also plotted so the response the beam could be viewed. The average settling times and error plots are shown below. It should be noted that the steady state areas of the plots depict oscilloscope noise, not beam vibration.

### 5.5.1 Average Settling Times

Average settling time for no control = **12.1s**

Average settling time for passive damping = **9.5s**

Average settling time for semi-active control = **6.9s**

## 5.5.2 Error Plots

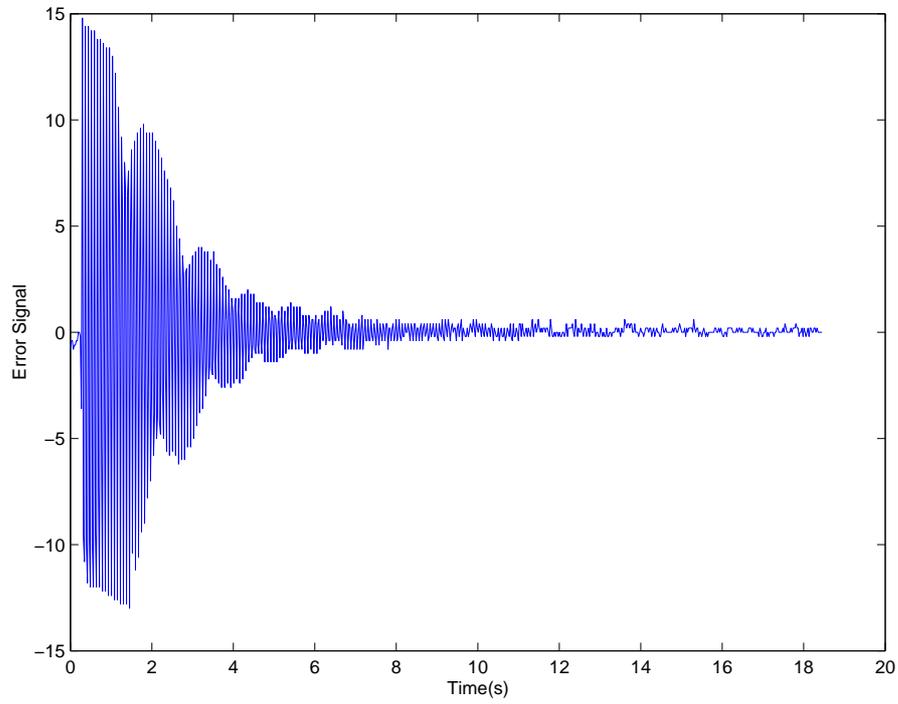


Figure 5.6: Test 1 - No Control

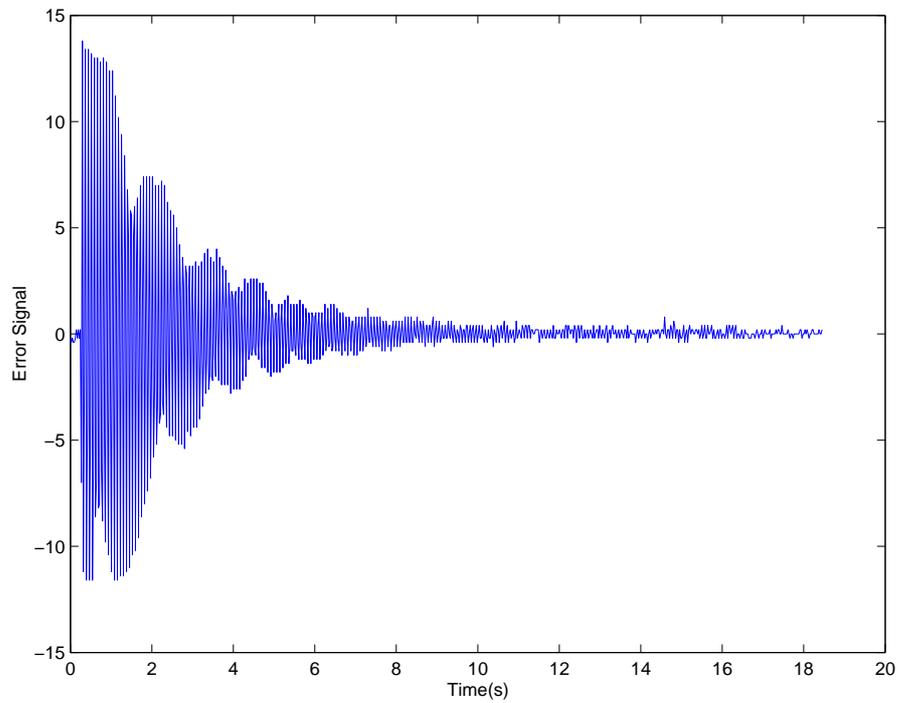


Figure 5.7: Test 2 - No Control

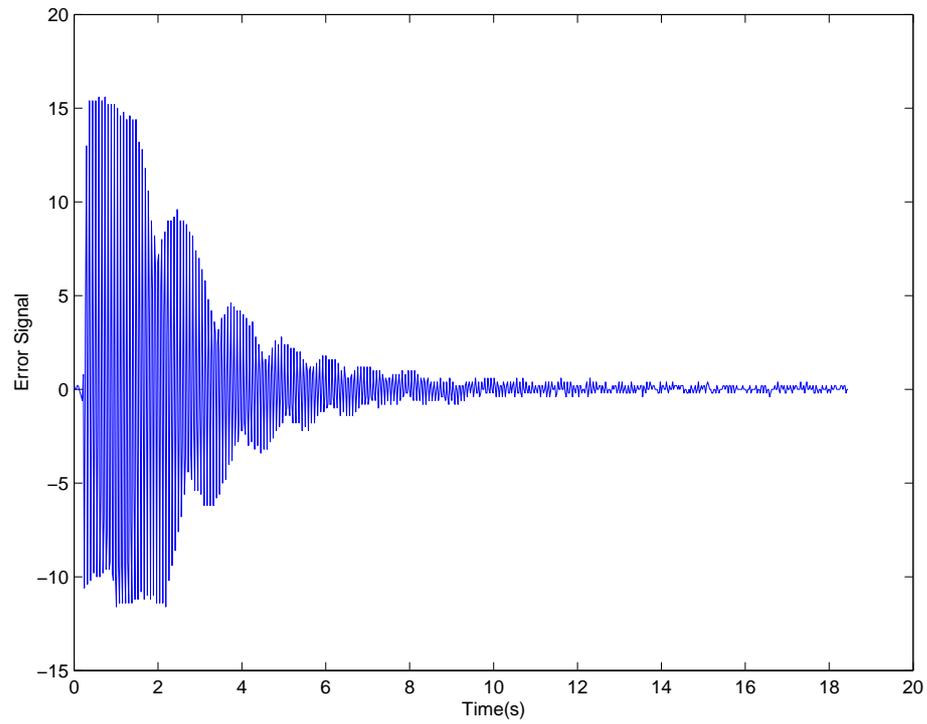


Figure 5.8: Test 3 - No Control

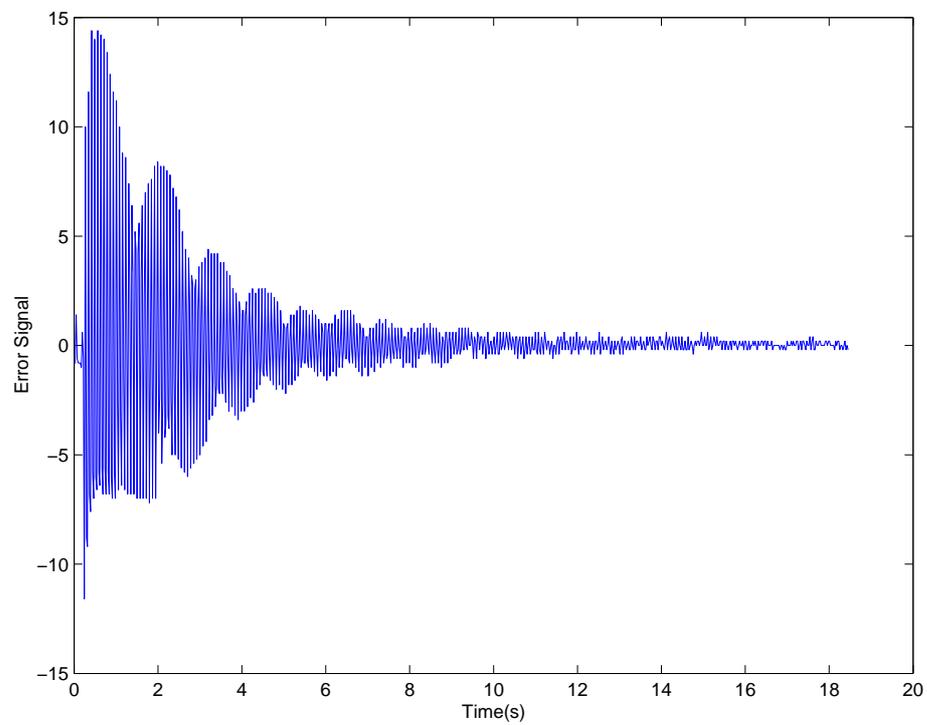


Figure 5.9: Test 4 - No Control

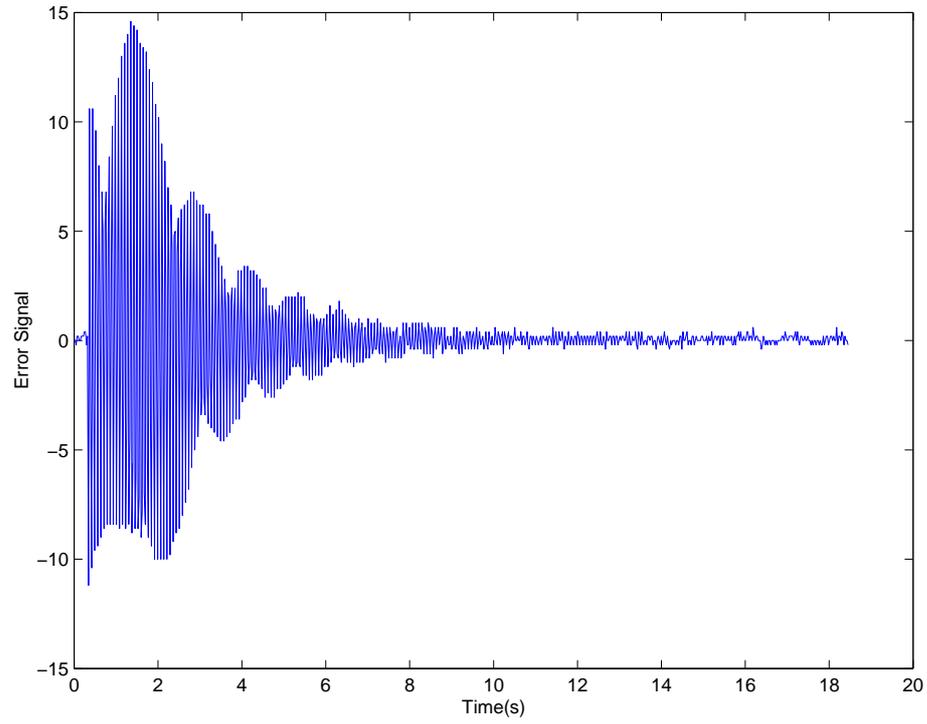


Figure 5.10: Test 5 - No Control

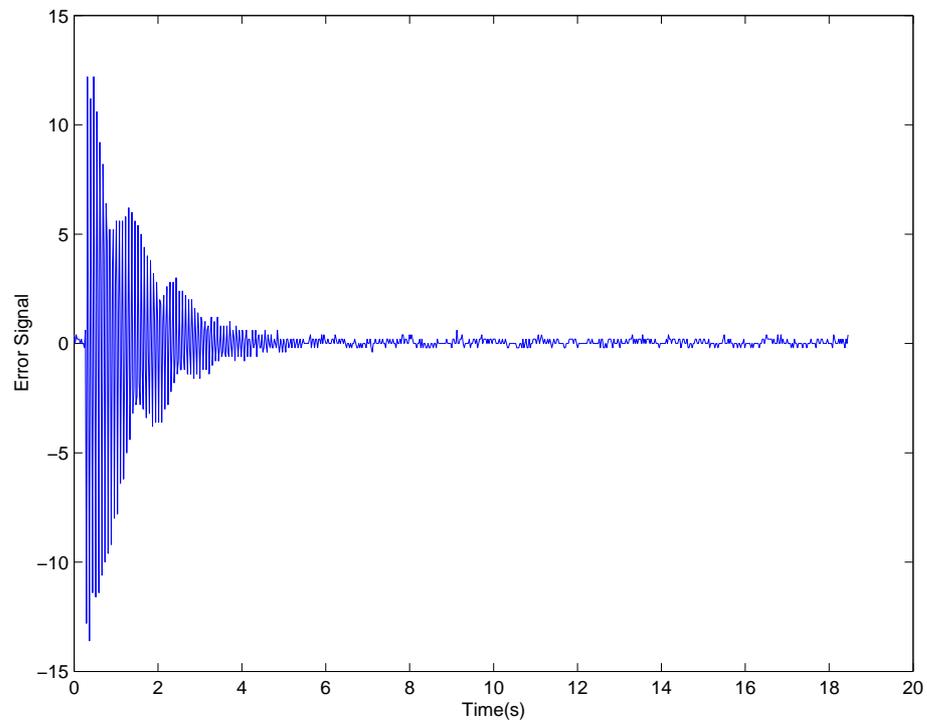


Figure 5.11: Test 6 - Semi-Active Control

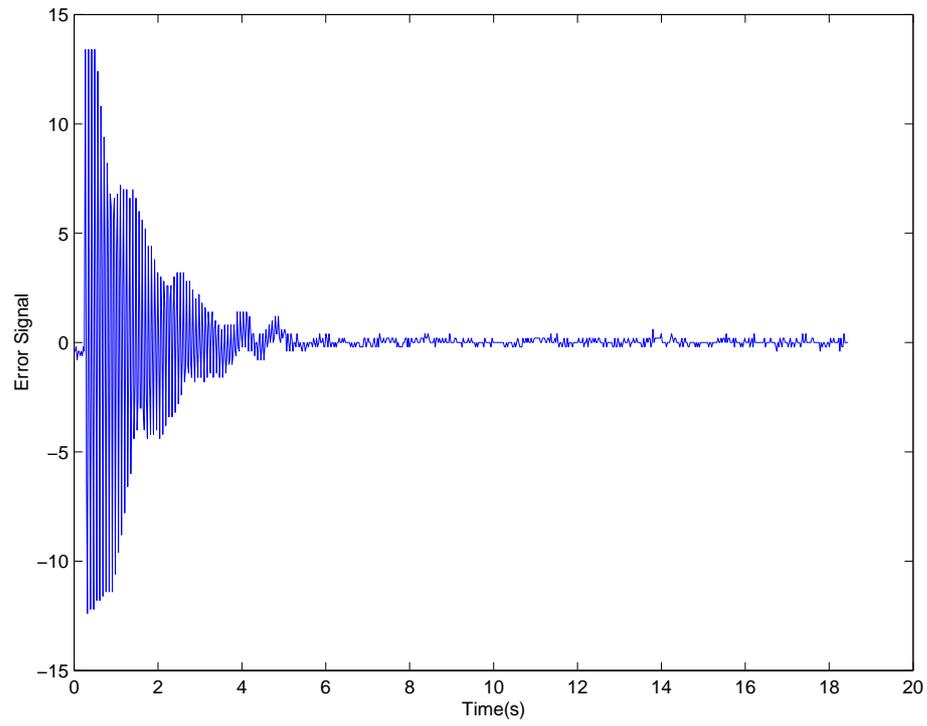


Figure 5.12: Test 7 - Semi-Active Control

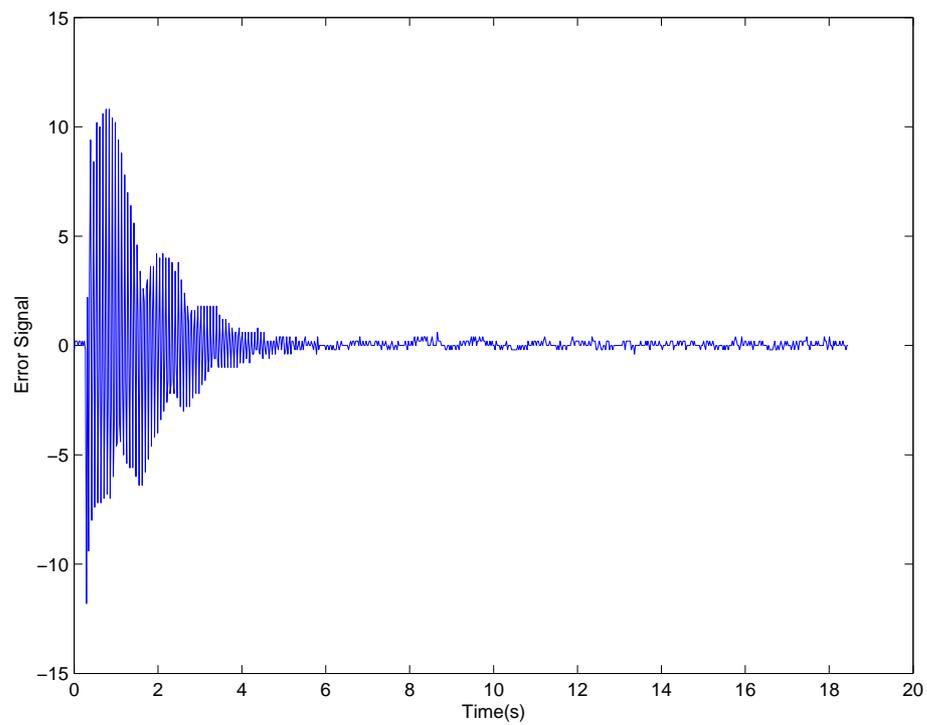


Figure 5.13: Test 8 - Semi-Active Control

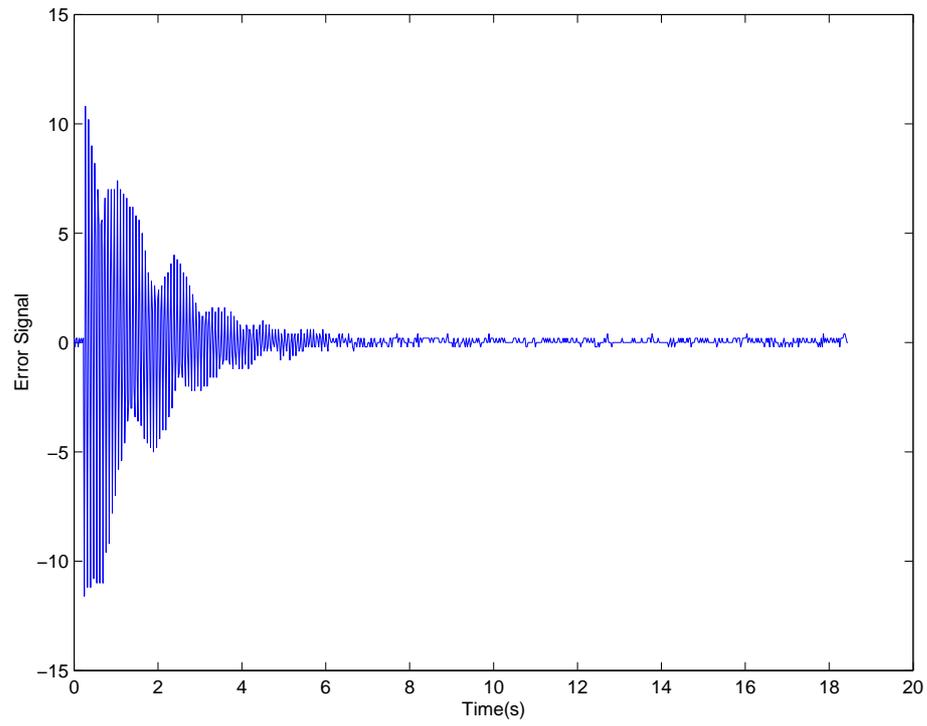


Figure 5.14: Test 9 - Semi-Active Control

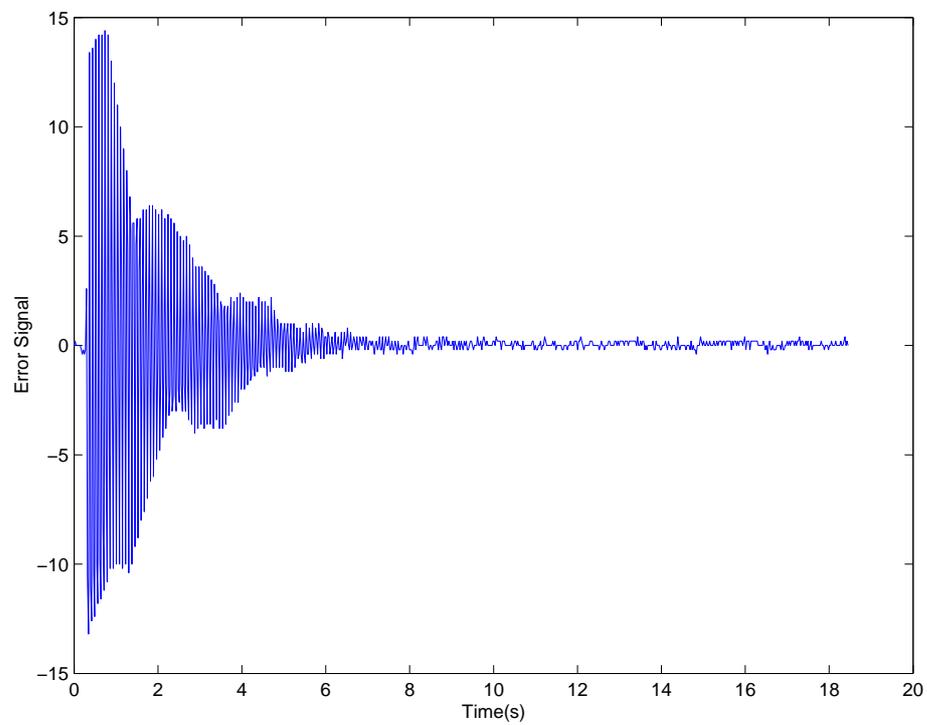


Figure 5.15: Test 10 - Semi-Active Control

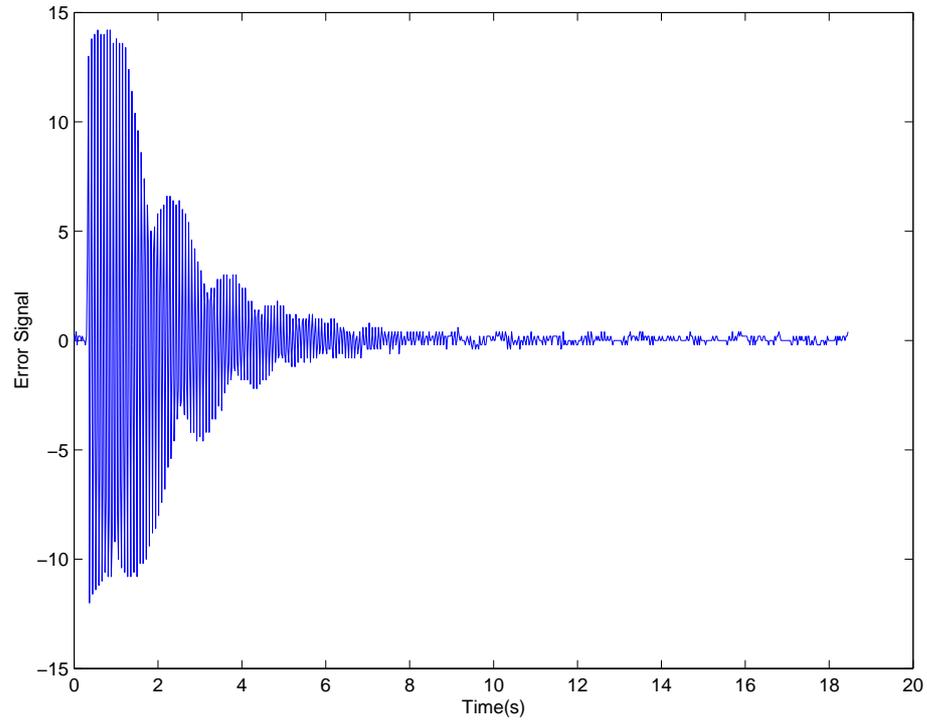


Figure 5.16: Test 11 - Passive Control

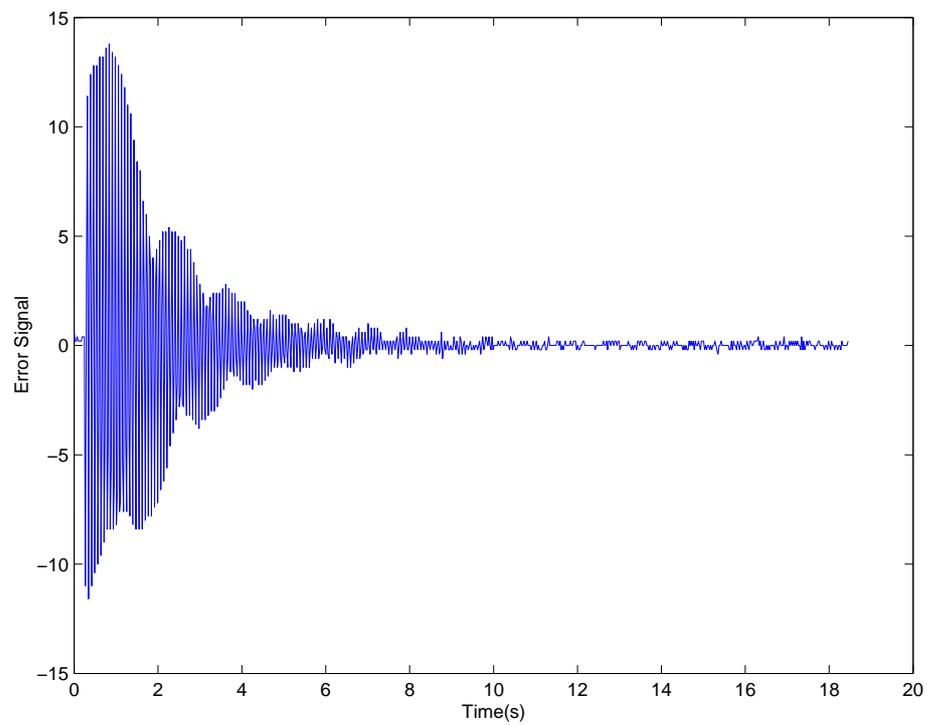


Figure 5.17: Test 12 - Passive Control

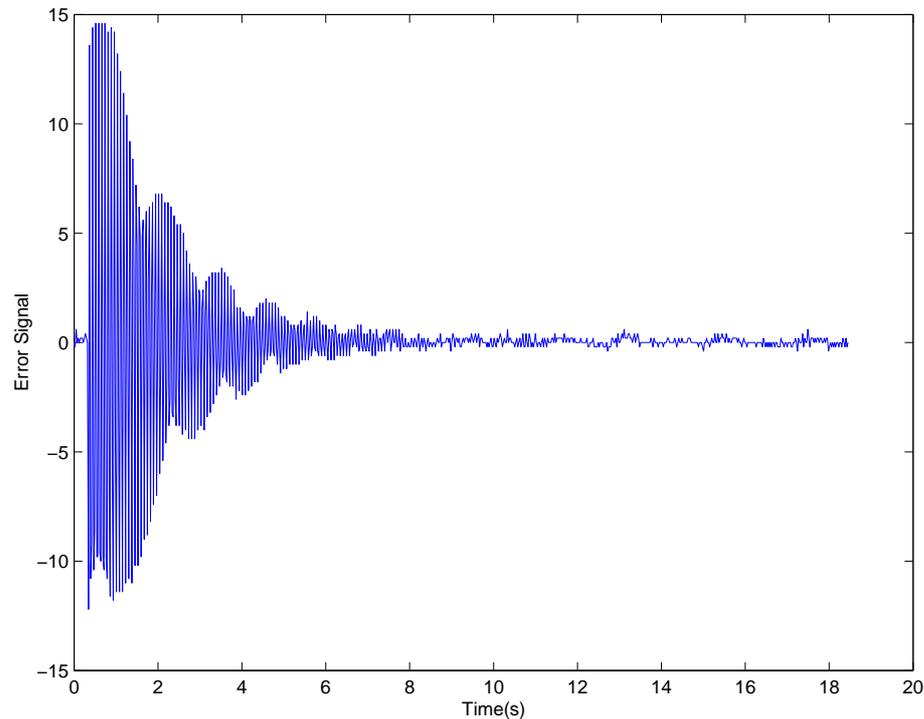


Figure 5.18: Test 13 - Passive Control

## 5.6 Conclusion

In a general sense, the test system was successful in suppressing the vibration of the beam. With full semi-active control, the settling time of the beam was almost halved. This is a fairly impressive result considering the simple nature of the equipment employed in the test system.

It should be noted that the testing method of tip displacements may have underestimated the performance of the control system. This is because the system relies on an *adaptive* controller which requires a certain amount of time to learn or train. The performance of the system would have probably improved if the input was constant or fairly steady. This is one prominent limitation of adaptive feed-forward control strategies. Although this is the case, it should be remembered that the underlying motive of this project was to develop a control system for machinery panels and enclosures.

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These structures are usually subject to constant or slowly varying vibrations, such as those induced by rotating machinery.

There are two main factors which may have reduced the overall performance of the system. The first of these involves the control device being a simple speaker. This speaker may have lacked the accuracy for the precise motion required by such a system. The damping would probably have been greater if the initial high precision electromagnetic shaker had been employed. The other factor which may have had an impact on the performance is the method of sampling the reference and error signals. At first it was thought that interrupts were not required for the low frequency sampling, however, the endless loop approach may have degraded the accuracy of the sampling and thus reduced the performance of the controller.

## **Chapter 6**

# **Conclusions and Further Work**

## 6.1 Achievement of the Project Objectives

The first objective of the project was to investigate current approaches of active vibration control, with a specific focus on adaptive feed-forward systems. This was the research aspect of the project, detailed in the first three chapters of the dissertation. It was difficult to make any direct comparisons between the many methods of actively controlling a vibrating structure. This is because the test conditions and physical structures varied from test to test. However this may be, this research was not conducted in vain as a wealth of knowledge and insight into the field of vibration control was gained. One prominent point to note is that the domain of vibration control is gaining interest internationally. This has led to the emergence of a wide variety of control strategies that are extremely effective in vibration suppression.

The next major goal of the project was to develop and simulate one particular adaptive algorithm in MATLAB. The important point of conclusion is that the performance of the LMS algorithm was found to vary with the filter length and convergence coefficient,  $\mu$ . This is consistent with current literature concerning the LMS algorithm. This insight led to the manual optimisation of these parameters during the real-time testing of the prototype system.

The final requirement of the project was to build and test a vibration suppression system for a simple cantilever beam. This system was to utilise an adaptive feed-forward control scheme. If time permitted, the research was to encompass more complex adaptive algorithms and physical structures. A prototype system was constructed and tested, however, time restrictions prevented any extension of the research. The test system worked fairly well but several aspects of the design could have been improved. The most important areas of concern were the sampling method and the accuracy of the control speaker.

## 6.2 Further Work and Recommendations

There are many avenues of investigation a future student could adopt in the field of active vibration control, however, these are far too broad to mention. This section includes this project and prototype system only. The future work of this project can be divided onto three areas:

### **Physical Test System and Testing Procedure:**

- Expand the research to panels and enclosures.
- Test the prototype system at resonant frequencies.

### **Hardware Design:**

- Test the system with a high precision shaker as the control device.
- Implement the system on a digital signal processor.

### **Algorithm and Software Design:**

- Investigate and implement more complex adaptive algorithms.
- Use interrupts as basis for the sampling of signals

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**Appendix A**

**Project Specification**

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

**ENG4111/4112 Research Project**  
**PROJECT SPECIFICATION**

STUDENT: GREGORY ROBERT LISTER

TITLE: ACTIVE VIBRATION CONTROL

SUPERVISOR(S): C. SNOOK & DR. J. LEIS

ENROLMENT: ENG4111 – S1, ONC, 2006  
ENG4112 – S2, ONC, 2006

PROJECT AIM: The general aim of the project is to investigate low cost sound/vibration attenuation systems for machinery panels and enclosures.

**PROGRAMME:** **Issue 2, 14<sup>th</sup> May 2006**

- 1) Investigate current approaches taken to minimise structural vibration using active control techniques.
- 2) Specifically analyse adaptive feed forward control systems.
- 3) Develop one adaptive control algorithm for simulation in MATLAB.
- 4) Simulate adaptive control algorithm with hypothetical vibration inputs to ascertain performance characteristics.
- 5) If resources permit, construct a vibration control system and test it on a simple beam.
- 6) If time permits, extend the research to control of more complex structures and more complex control algorithms.

**Signed:**

\_\_\_\_\_  
Student

\_\_\_\_\_  
Supervisor(s)

\_\_\_\_\_/\_\_\_\_\_/\_\_\_\_\_, \_\_\_\_/\_\_\_\_/\_\_\_\_\_  
\_\_\_\_\_/\_\_\_\_\_/\_\_\_\_\_, \_\_\_\_/\_\_\_\_/\_\_\_\_\_

## **Appendix B**

### **MATLAB Source Code**

## B.1 The lms.m MATLAB Function

```

% MATLAB Simulation
% The LMS Filter
% Gregory Lister, USQ

function [e] = lms(mu, filtlen)
format long;

% Set filter length limits
if filtlen <= 0 | filtlen > 30
    error('Filter length must be greater
    .....than 0 and less than or equal to 30');
end

% Set convergence coefficient limit
if mu <= 0 | mu > 0.02
    error('Step size, mu, should be greater
    .....than 0 and less than or equal to 0.02');
end

% Choose primary reference
sig = menu('Choose Reference Signal', 'Harmonic',
    'Gaussian Noise', 'Noise + Harmonic');

% Create/read chosen reference
if sig == 1
    s1 = 1:10000;
    x = (sin(s1*pi/10));
elseif sig == 2
    load noise.dat;
    x = noise';
elseif sig == 3
    s1 = 1:10000;
    load noise.dat;
    s2 = noise';
    x = (sin(s1*pi/10)+0.25*s2);
end

% Initialise filter taps
w = zeros(1, filtlen);

% Implement LMS algorithm
for n = 1:10000-(filtlen-1)
    xfilt = x(n:n+(filtlen-1)); % Shift x
    y(n) = sum(w.*xfilt); % Calculate y
    e(n) = x(n)-y(n); % Calculate e
    w = w+(2*mu*e(n).*xfilt); % Update filter weights
end

```

```
% Plot the control signal
T1=0:1:(10000 - filflen);
plot(T1,y)
xlabel 'Number_of_Samples'
ylabel 'Output,_y'
title 'Control_Signal'
pause
% Plot the error signal
T2=0:1:(10000 - filflen);
plot(T2,e,'r')
xlabel 'Number_of_Samples'
ylabel 'Error,_e'
title 'Error_Signal'
clear;clc
```

## Appendix C

# Assembly and C Source Code

## C.1 The test.c Test Function

```
/*
*****
Active Noise Control Implementation
LMS Feedforward Algorithm

Gregory Lister, USQ
Adapted from contec.c (J. Leis)
Requires ioportlib.asm (J. Leis)
*****
*/

/*
This program does not use interrupts and
should be loaded via DOS boot to reduce
processor multitasking for efficient sampling
*/

// Include libraries
#include <stdio.h>
#include <math.h>
#include <stdlib.h>
#pragma pack(1)
#pragma check_pointer(off)
#pragma check_stack(off)

// I/O routines in ioportlib.asm
extern int inportb(int portaddr);
extern void outportb(int portaddr, int portval);
extern int inportw(int portaddr);
extern void outportw(int portaddr);

// AXIOM register addresses
#define AXIOMBASE 0x300
#define ADPORTLOW (AXIOMBASE+0x0)
#define ADPORTHIGH (AXIOMBASE+0x1)
#define DAPORTLOW (AXIOMBASE+0x6)
#define DAPORTHIGH (AXIOMBASE+0x7)
#define STATUS (AXIOMBASE+0x8)
#define GAIN (AXIOMBASE+0x11)
```

```

/*****
                                READ AD
*****/
unsigned short ReadAD(unsigned short chan){
    outportb( ADPORTLOW, 0x00 + chan);
    while(inportb(STATUS) >= 0x80){
        printf("Sampling in progress\n");
    }
    int sample = inportw(ADPORTLOW);
    sample = sample>>4;
    return(sample & 0x0fff);
}

/*****
                                WRITE DA
*****/
void WriteDA(int value){
    value = value<<4;
    value = value & 0xfff0;
    //printf("value = %i\n", value);
    outportb(DAPORTHIGH, value);
    return;
}

/*****
                                MAIN PROGRAM
*****/
int main(){
    double mu=0.00000001;//Mu
    double w[]={0,0,0,0,0,0,0,0,0,0};//Filter taps
    int xfilt[]={0,0,0,0,0,0,0,0,0,0};//Sampled input
    int x=0;//Current sample
    int y=0;//Filter output
    int e=0;//Error
    int y1;//Temporary variable
    int k,i,j,m,n;//For loop variables

    WriteDA(2048);//Set DAC to zero
    outportb(GAIN,0x03);//Set gain range

```

```
while (1){
    // Sample x
    x=ReadAD(0);
    // printf("x = %i\n",x);

    // Shift delay line and input new sample x
    for (k=9;k>=1;k--){
        xfilt [k]=xfilt [k-1];
    }
    xfilt [0]=x;

    // Calculate y
    for (i=0;i<10;i++){
        y1=xfilt [i]+w[i];
        y=y+y1;
    }
    // printf("y = %i\n",y);

    //Scale and shift y
    y=y/100;
    if (y>=0){
        y=y+2048;
    }
    else y=2048-abs(y);

    // Output y
    WriteDA(y);

    //Clear y
    y=0;

    // Sample e
    e=ReadAD(0);
    // printf("e = %i\n",e);

    //Update filter weights
    for (j=0;j<10;j++){
        w[j]=w[j]+mu*e*xfilt [j];
    }
}
}
```



```

    pushl    %ebp
    movl    %esp,%ebp

    ; always save edi
    pushl    %edi

    ; integer arg portaddr is 8(%ebp)
    movl    8(%ebp), %edx

    ; local2 = inval2;
    ; integer arg inval2 is 12(%ebp)
    ; movl    12(%ebp),%edi

    mov $0, %eax
    in %dx, %al

    ; test
    ;mov %dx, %ax

    ; test return value
    ;movl    $0x1234, %eax

    ; epilog
    popl    %edi
    leave
    ret
;
;
;
; void outportb(int portaddr, int portval)
_outportb:
    ; prolog
    ; save base ptr
    ; base ptr to stack base
    pushl    %ebp
    movl    %esp,%ebp

    ; always saves edi
    pushl    %edi

    ; integer arg portaddr is 8(%ebp)
    movl    8(%ebp), %edx

    ; integer arg portval is 12(%ebp)
    movl    12(%ebp), %eax

    out %al, %dx

    ; test return value
    ; movl    $0x0056, %eax

    ; epilog
    popl    %edi
    leave
    ret
;
;
;
; int inportw(int portaddr)
_inportw:
    ; prolog

```

```

; save base ptr
; base ptr to stack base
pushl  %ebp
movl   %esp,%ebp

; always save edi
pushl  %edi

; integer arg portaddr is 8(%ebp)
movl   8(%ebp), %edx

; local2 = inval2;
; integer arg inval2 is 12(%ebp)
; movl   12(%ebp),%edi

mov $0, %eax
; in %dx, %al
in %dx, %ax

; test
; mov %dx, %ax

; test return value
; movl   $0x1234, %eax

; epilog
popl   %edi
leave
ret
;
;
;
; void outportw(int portaddr, int portval)
_outportw:
; prolog
; save base ptr
; base ptr to stack base
pushl  %ebp
movl   %esp,%ebp

; always saves edi
pushl  %edi

; integer arg portaddr is 8(%ebp)
movl   8(%ebp), %edx

; integer arg portval is 12(%ebp)
movl   12(%ebp), %eax

; out %al, %dx
out %ax, %dx

; test return value
; movl   $0x0056, %eax

; epilog
popl   %edi
leave
ret
;
;

```

## Appendix D

# **AXIOM AX5412 Specifications**

## GENERAL DESCRIPTION

### Introduction

The AX5412 high speed data acquisition board is designed to interface directly with the IBM PC/XT/AT or compatible computers and compatible with Kethley Metrabyte (U.S.A Company) DAS-16.

The following features are implemented on AX5412 board.

1. 12 bit A/D resolution:  
AX5412 board, using 12 bit A/D converter, has maximum throughput rate up to 90KHz in DMA mode.
2. Programmable gain:  
There are low gain and high gain version of AX5412 board. In low gain version, the programmable gains are 1, 2, 4, 8, the high input impedance ranges of +10V, +5V, +2.5V, +1.25V unipolar and  $\pm 10V$ ,  $\pm 5V$ ,  $\pm 2.5V$ ,  $\pm 1.25V$  bipolar. In high gain version, the programmable gains are 1, 10, 100 and 500, the high input impedance ranges of +10V, +1V, +.1V, +.02V unipolar and  $\pm 10V$ ,  $\pm 1V$ ,  $\pm .1V$ ,  $\pm .02V$  bipolar.
3. Switch selectable 16 channels single-ended or 8 channel differential analog input:  
To have a good noise immunity, higher accurate A/D conversion, select differential mode.
4. 3 speed A/D conversion:  
Low speed A/D conversion is initiated by software trigger. Middle speed A/D conversion about up to 3KHz throughput rate is initiated by interrupt including timer trigger and external trigger.  
High speed A/D conversion about up to 90KHz throughput rate is initiated by DMA including timer trigger and external trigger.
5. 2 channels of 12 bit D/A output:  
The D/A converter may be operated with internal reference voltage or external reference voltage.
6. 8 bits general purpose TTL/DTL compatible digital output and digital input.

**Accessories Available****AX750**

Low cost screw terminal panel accommodates all analog input connections. Shipped complete with 3.3 foot (1 meter) cable and 50 pin connector.

**AX751**

Universal screw terminal panel conceived for 50 pin connector wiring conveniently.

**AX752**

Table top screw terminal panel designed for thermocouple and low level inputs. Provides differential input for common mode rejection (+/-10V); Gains of 1,10,50,100 and 1000; Voltage offset and cold-junction compensation; and 8Hz low pass filter.

**AX753**

4 channels simultaneous sample and hold with fully differential input accessory panel board.

- \* 4 channels with fully differential input, input impedance more than 1 G $\Omega$  and 95dB CMR.
- \* 1 to 1000 available input gain range with switch selectable.
- \* Daisy chained up to 16 channels sampled simultaneously
- \* Less than 10  $\mu$ s acquisition time
- \* Up to 90KHz sampling rates with AX5412 synchronous mode.
- \* Under 40 ns aperture uncertainty between channels

Shipped complete with 3.3 foot (1 meter) cable and 50 pin connector.

## Specifications

### Analog Input

	AX5412-L	AX5412-H
Number of inputs	16SE or 8DI	16SE or 8DI
Resolution	12 bits	12 bits
Gain	1,2,4,8	1,10,100,500
Input Range		
Unipolar	0-1.25,2.5,5,10V	0-.02,.1,1,10V
Bipolar	±1.25,2.5,5,10V	±.02,.1,1,10V
A/D Throughput	90KHz	90KHz
System Accuracy (Gain=1)	±.03% FSR	±.03% FSR
Software Calibration	Yes	Yes
Channel Acquisition Time to ± 1/2 LSB		
Gain=1,2,4,8,10	9μs	9μs
Gain=100,500		15μs
A/D Conversion /Time	9μs	9μs
Input Impedance		
Off Channel	100Mohm, 20pF	100Mohm, 20pF
On Channel	100Mohm, 20pF	100Mohm, 20pF
Maximum Input Voltage Without Damage		
Power On	+/-35V	+/-35V
Power Off	+/-20V	+/-20V
Bias Current	20nA	20nA
Sample and Hold Aperture Uncertainty	0.15ns	0.15ns
Sample and Hold Aperture Delay	15ns	15ns
IMD	-90dB, 100kHz	-90dB, 100kHz

Sample and Hold Slowing Rate	45V/ $\mu$ s	45V/ $\mu$ s
Sample and Hold Acquisition Time	1.0 $\mu$ s	1.0 $\mu$ s
Output Coding Unipolar Bipolar	Binary Offset binary	Binary Offset binary
Common Mode Rejection Ratio Gain = 1	90dB	90dB
Integral Nonlinearity	$\pm$ 1LSB	$\pm$ 1LSB
Differential Non- Linearity	12 bits	12 bits
Gain Drift	20ppm/ $^{\circ}$ C	20ppm/ $^{\circ}$ C

#### Analog Output

Number of channels	2
Output Ranges	0 to 5V with -5VR, internal reference supply (+10Vdc&Vac with external reference supply)
Input data coding	Straight binary
Current Output, Voltage range	+5mA max.
Source Impedance	0.02 $\Omega$
Capacitive Driver Capability Protection	Short circuit to common for voltage ranges, current outputs are short circuit, and reverse polarity protected.

**Accuracy**

Resolution	12bits
Nonlinearity	$\pm \frac{1}{2}$ LSB
Gain Error	Adjustable to zero
Zero Error	Adjustable to zero
System Accuracy	+/-0.025% FSR (voltage out)

**Digital I/O**

Digital Input Lines	8
Digital Output Lines	8
Logic Family	LSTTL
Input/Output Level	TTL/DTL Compatible

**Electrical Characteristics**

Logic High Input Voltage (VIH)	2V (min.)
Logic Low Input Voltage (VIL)	0.8V (max.)
Logic High Input Current (IIH)	20 $\mu$ A (max.) at VI=2.7V
Logic Low Input Current (IIL)	-0.2mA (max.) at VIL=0.4V
Logic High Output Voltage (VOH)	2.7V (min.) at IOH=-0.4mA
Logic Low Output Voltage (VOL)	0.5V (max.) at IOL=8mA
Logic High Output Current (IOH)	-0.4mA (max.)
Logic Low Output Current (IOL)	8mA (max.)

**External A/D Triggers**

Number of Trigger Lines	1
Functions	Initiate A/D, synchronizes start of conversion with outside event.
Input Type	Schmitt trigger, edge sensitive, clock on rising edge.
Logic Family	LSTTL

Logic Load	1 LSTTL load
Input Termination	10Kohm pull up to +5V
Minimum Pulse Width	
Clock high	40ns
Clock low	40ns
<b>Built-in Frequency Source</b>	
Base Frequency	1MHz or 10MHz
<b>Counter/Timer (8254-2)</b>	
Type	8254-2 programmable interval timer Counters. Three 16 bit down counters. 2 permanently connected to 1 or 10MHz clock as programmable timer. 1 counter free.
Clock Input	D.C. to 10MHz
Input Level	TTL, DTL, CMOS compatible
Output Range	2.5MHz - 72 minutes/pls
<b>Interface Characteristic</b>	
Compatible Bus	IBM Personal Computer XT & AT Bus
Interface Type	I/O mapped with 10-bit addressing
Number of Locations Occupied	16 bytes
Data Path	8 bits
Factory-Assigned Base Address	300hex
Number of Interrupts	1
Interrupt Level	2, 3, 4, 5, 6, or 7
Interrupt Source	A/D Done, A/D DMA Done

Number of DMA	1
Channels of DMA	1 or 3
DMA Source	A/D Done
Connector	All user connections are made to a single 50-pin connector (CN1); mating connector is 50-pin 3M type.

**Power Requirements**

+12Vdc	120 mA typ.
-12Vdc	20 mA typ.
+5Vdc	600 mA typ.
±15Vdc	Generated by on board DC/DC power converter; low noise, regulated

**Physical/Environmental**

Dimensions	(99H X 334W) mm
Weight	285gms
Operating Temperature Range	0 to 50°C
Storage Temperature Range	-25 to 85°C
Relative Humidity	To 90%, non-condensing