AN INVESTIGATION OF THE FEASIBILITY OF MICROSCALE ADAPTIVE PASSIVE VIBRATION NEUTRALIZERS

by

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

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(ABSTRACT)

This thesis concerns the control of an adaptive passive vibration neutralizer and the feasibility of miniaturizing this type of tunable vibration neutralizer for small-scale applications.

An analytical model for the adaptive passive vibration neutralizer is derived and compared to experimental results. A tuning algorithm is derived from a curve-fit of experimental tests on the specific neutralizer. A more generic tuning algorithm is also developed, which does not require testing of the neutralizer for optimal control. Both tuning algorithms are tested using a chirp forcing function to simulate drift in the excitation frequency of a host structure. Computer simulation and experimental results are given for these tests.

A novel low-cost, small-scale vibration neutralizer is constructed from packing bubble-wrap. Analytical models for the stiffness are calculated, and experimental data is used to derive a damped mass-spring model.

Miniaturization of vibration neutralizers is described, and many of the pitfalls in design are discussed. Theoretical tuning frequencies of possible adaptive passive vibration neutralizers at different scales are included. The goal for these miniaturized vibration neutralizers is vibration control in computer hard drives.

A hard drive is analyzed for vibration problems. Included are plots of the velocities of the read-write head and spindle. Limitations of the measurement equipment are discussed, and directions for future work on small-scale tunable vibration neutralizers are outlined.
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Chapter 1: Introduction

Vibration neutralizers are devices designed to counteract unwanted vibrations in a host structure [1]. These host structures have been as diverse as ships [2] and large machines [3]. More recently, vibration neutralizers have been used in portable mechanical devices [4] and in aerospace systems [5]. These more recent small-scale and light-weight mechanical systems require smaller vibration neutralizers.

In a talk given in 1959, Richard Feynman explained a vision of miniaturized mechanical and electrical systems [6]. His vision and subsequent calculations showed how military and consumer products could be constructed with nanoscale electronic and machine parts. (Nanoscale refers to devices and components being from 1nm to just under 1μm on a side.) He ended his talk with a challenge to researchers: to build a working microscale engine. (Microscale refers to devices and components being from 1μm to just under 1mm on a side.) The prize for this challenge was awarded the year after his talk. Even with no experimental background on the topic, many of Feynman’s guidelines for the design of nanoscale systems are still used today.

Miniaturizing vibration neutralizers and tuning them are now key issues with microsystems and nanotechnological applications expanding almost daily, especially in the realm of electronics and electromechanics.
1.1 The Hard Drive Vibration Problem

The hard drive is arguably the noisiest and most frequently used mechanical component in modern computers. Some of today’s fastest CD-ROM drives are very noisy, but these are typically accessed much less frequently than hard drives in personal computers. The design constraints for current CD-ROM drives are also not as severe because compact-discs are not shrinking in size for more portable applications whereas laptop hard drives are. A neutralizer used to reduce hard drive vibrations could be adapted for use in CD-ROM drives. These considerations show why the hard drive vibration problem is examined in this thesis.

A hard drive is a set of magnetized rotating discs that stores data using read-write heads which float above these discs. Over the years, storage space and speed of hard drives have increased exponentially as physical size has decreased [7]. Every increase in areal density and speed came only with innovative new technologies to overcome each new restriction. (Areal density is the number of bits of data per unit area of the disc’s surface.) One of the recurring problems throughout this process of innovation was vibration [8]. Vibrations in hard drives can cause track misregistration errors, which are misreads of the data by the read-write head, or even head crashes, in which the read-write head strikes the surface of the disc. Predicting track misregistration errors and finding ways to minimize their occurrences are described in [9], [10], [11], and [12], though this list is not nearly exhaustive of the applicable research. As hard drives have been reduced in size, new vibration issues have arisen; therefore, new solutions are constantly required.

1.2 Miniaturization

As stated earlier, nanoscale systems are systems on the order of nanometers on a side, while microscale systems are on the order of micrometers on a side. Mesoscopic systems are systems that are on the border of the two scales, usually having nanoscale components and a microscale total form-factor [13]. Guidelines have been discussed over the past decade, but use of these terms, explicitly “mesoscopic”, is still indefinite [14].

According to Reed and Kirk [15], one can design any structure as precisely as can be imagined, down to placement of individual atoms, using one of many lithography
methods. Some of the many machine parts that can be built at the nanoscale are cams, gears, bearings, rollers, belts, rods, clutches, and ratcheting surfaces. These mechanical systems function in the same manner between the nanoscopic and macroscopic worlds. Some of these structures are so small that inter-atomic and quantum effects can play a significant role in the performance of these devices, adding to the complexity of modeling and the difficulty of design [14].

1.2.1 Nano- and Microscale Transducers

All control systems require sensors and/or actuators, and for small scales electrostatic and piezoelectric materials are prime candidates. These materials provide a voltage when strained allowing them to inform control systems of deformation. They also strain with an applied voltage allowing control systems to deform the materials [14]. This relationship between electric field and strain is reciprocal in nature [16]. Electrostatic materials tend to have larger displacements but lower force per volt, while piezoelectric materials apply larger forces but do not extend as far [14]. “Considerable effort is undertaken to fabricate piezoelectric actuators” in lieu of electrostatic actuators for microsystem design. Actuators made of lead-zircon-titanate (PZT) are favored among piezoelectric materials due to its high dielectric constant [17], and some applications of piezoelectric materials’ uses are given by [18] and [19].

An early text on the basics of piezoelectric ceramic materials is Jaffe, et al. [20], which also includes many tables of material properties. Table 1.1 shows the properties of some materials used in the neutralizer prototypes of this thesis, in particular PZT-5A.
<table>
<thead>
<tr>
<th>Material Name</th>
<th>Density $\rho$ (kg/m$^3$)</th>
<th>Elastic Modulus $E$ (GPa)</th>
<th>Piezoelectric Coefficient</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum$^{[21]}$</td>
<td>2700</td>
<td>70</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Cellulose Acetate$^{[21]}$</td>
<td>1270</td>
<td>1.8</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Copper$^{[21]}$</td>
<td>8960</td>
<td>110</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>EVA Film Grade$^{[21]}$</td>
<td>926</td>
<td>0.19</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Iron or Steel$^{[21]}$</td>
<td>7870</td>
<td>200</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Lead$^{[21]}$</td>
<td>11340</td>
<td>14</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>PVDF$^{[21]}$</td>
<td>1790</td>
<td>2.3</td>
<td>18 V/m</td>
<td>Dielectric Strength</td>
</tr>
<tr>
<td>PZT (ACX™)$^{[22]}$</td>
<td>7700</td>
<td>69</td>
<td>-179E-12 m/V</td>
<td>Strain Coefficient (d$_{31}$)</td>
</tr>
<tr>
<td>PZT 5A$^{[23]}$</td>
<td>7800</td>
<td>66</td>
<td>-190E-12 m/V</td>
<td>Strain Coefficient (d$_{31}$)</td>
</tr>
</tbody>
</table>

Table 1.1: Physical Properties of Some Materials

Modeling miniaturized systems, typically with computer simulations, allows for preliminary design optimization without spending time or money on numbers of prototypes. Yu$^{[24]}$ and Rogacheva$^{[25]}$ have recent nonlinear modeling techniques for piezoelectric layers and shells, and a finite element mathematical method for piezoelectric vibrations is explained in$^{[26]}$.

Piezoelectric elements are often used in small-scale applications because of their “simplicity and ruggedness”, and, even at the microscale, the voltage applied is proportional to the force exhibited by the element$^{[27]}$. Goto, et al.$^{[28]}$, explain some of the intricacies of using piezoelectric thin film actuators in microstructures. Leveraging techniques to increase the stroke of PZT actuators by an order of magnitude are discussed by Perrette, et al.$^{[29]}$.

Numazawa, et al., have developed a plasma treatment to control the dielectric constants of 1-3 piezoelectric composites made from micro-scale PZT rods. These composites were built for use as ultrasonic probes for medical applications such as blood-flow measurement$^{[30]}$. In biological nanotechnology, deoxyribonucleic acid (DNA) is being used not only to aid in construction of nanoactuators but also to power nanoactuators$^{[31]}$. Many more DNA based applications can be found in$^{[32]}$, $^{[33]}$, and $^{[34]}$. 
1.2.2 Fabrication of Nano- and Microscale Devices

Many nanoscopic fabrication methods exist, and each has its own cost and resolution limitations that need to be considered before one is chosen. Often chemical processes are used to add layers to and remove layers from a structure, allowing for relatively complex structures that can have gaps, holes, and tunnels. Sputtering and inert-gas evaporation are also used to create intricacies in nano- and microscale structures. See [35] for more details on these methods.

For more information on nanoscale precision, [36] has descriptions of many of the methods used when creating microstructures. Included are etching and lithography for creating features and surfaces, plus several bonding methods. Several different experimental papers based on lithography methods are provided in [37], [38], [39], [40], and [41]. Other methods have found varying success for different applications and are discussed in [42] on plasmas, [43] on low-temperature compaction of powders, and [44] on sol-gel processing methods. [45] is a study of electron microscopy of nanomaterials with a fascinating section examining the structure of ancient American materials, namely an extraordinarily stable blue-colored clay.

With these nearly atom-by-atom construction techniques and the use of piezoelectric materials, creating microscopic vibration neutralizers for hard drives has become possible. Cost has been prohibitive, but these expenses will decrease considerably as manufacturing processes for miniaturized devices improve [46]. Consumers are also willing to spend more money on smaller, lighter-weight versions of electronic devices [6]. Laptop computers with smaller and lighter hard drives allow for further reductions in weight and form-factor of the computer itself, which has warranted higher consumer costs in the past.

1.2.3 Current Trends in Miniaturization

A small sampling of the up and coming technologies, including some sources for further reading, are listed below. They were chosen for their applicability in future work relating to this thesis.

Recently, ultraprecision trapezoidal microgrooves have been tested in microscale devices, in lieu of adhesives to join structures, with success [47]. Yao, et al. [48], developed a microscopic electromechanical actuator that accomplished the work that a
2\text{mm} \text{ commercial piezoelectric actuator could accomplish in a beam of only } 400\text{nm}. \text{ Rows of tiny threshold acceleration sensors, each measured only several micrometers on a side, were tested by Selvakumar, et al. [49].}

High-aspect-ratio (HAR) structures can be on the order of one millimeter in height while having a considerably smaller width and depth [50]. This general shape uses a minimum surface area on the main structure while allowing for much greater power storage capacity. Fischer suggested two methods to build HAR structures: the single layer method and the multiple layer stacking method. The multiple layer stacking method requires creating many shorter elements that are assembled into a HAR structure. These elements can be restacked to different heights without building a completely new structure. The single layer method uses lithography to create the structure at once, which does not have the alignment, stability, and durability issues that multiple layer stacking assemblies have. However, precise lithography methods (like the use of low energy synchrotrons) cannot be used with this single layer method, and resolution limitations per cost can become an important issue in design [51]. One application of the multiple layer stacking method is a linear magnetic actuator, which achieved heights over one millimeter, or a 100:1 aspect ratio in height to base width [52]. More recently, Kondo, et al. [53], explained the effect that high aspect ratios can have on high power electrostatic microactuators.

As devices like acceleration sensors and actuators continue to be developed on smaller and smaller scales, consumers will continue to expect the advantages that such miniaturization brings. More funding for research and lower costs to manufacture miniaturized mechanical devices will result. Small-scale sensing equipment also reduces the weight requirements for vibration reduction systems, which will allow for a wider range of vibration reduction applications. See Chapter 5 for a more thorough discussion of modeling and fabricating miniaturized devices.

1.3 Passive Vibration Neutralizer

The first passive vibration absorber was in the H.M.S. Inflexible, described in 1883 by Watts [2]. This absorber was tuned by changing the amount of water in a specially shaped tank until the minimum pitch amplitude was achieved. The pitch amplitude was reduced by over a third in certain sea conditions without loss of steering
control or speed with a nearly negligible increase in total mass of the ship. While this was the first published vibration reduction system for a large structure, the first patented dynamic vibration absorber was published in a U.S. Patent in 1909 by Frahm [54]. Eason [3] applied the idea of absorbing vibrations from machinery on upper floors of buildings by placing the offending machinery on springs.

In 1928, Ormondroyd, et al. [55], published one of the first widely distributed theoretical papers on vibration absorbers. Their vibration absorber theories were the inverse of Eason’s vibration absorber, attaching a mass-spring system to the vibrating machine, which is more practical in most cases.

In his paper, “Vibration Control,” Mead [1] coined the phrase “vibration neutralizer.” Vibration neutralizers reduce vibrations by adding impedance to a host structure. See Figure 1.1 for a diagram of this system.

![Figure 1.1: Vibration Neutralizer on a Host Structure](image)

With the neutralizer mass moving ninety-degrees out-of-phase with the motion of the host structure, the added impedance keeps the host structure from responding to vibrations at that frequency [56]. The amount of possible attenuation when a neutralizer is ninety-degrees out-of-phase, or tuned, is related to the damping in the neutralizer. While higher damping provides a wider range of frequencies that get attenuated, less vibration attenuation occurs at the tuned frequency [57].

Strasburg, et al., explain, through theory and experimentation, that several vibration dampers mounted on a host structure act as a unified impedance. This impedance reduces vibrations in the host structure at the tuned frequencies of each of the
vibration dampers [58]. Brennan [59] uses beams of different lengths affixed at a
common point to create a wideband vibration neutralizer. This method is simpler than
using an adaptive system, (see section 1.4,) because no control system is needed to tune
the neutralizer to reduce vibrations over a range of frequencies. This broadband method
would be most effective in cases where cost and space prohibit computational electronics
or in systems where the drift in excitation frequency is too rapid for control systems.

1.4 Active/Adaptive Vibration Neutralizer

According to Hunt [60], a vibration neutralizer is dynamic if it has a tuning
feedback control loop that keeps the neutralizer tuned to the excitation frequency of the
host structure. A passive vibration neutralizer has no feedback control system to allow
for tuning, which reduces the neutralizer’s effectiveness against vibrations other than at
its manufactured resonant frequency. With a lightly damped vibration neutralizer, the
feedback control system allows for the lightest and smallest vibration neutralizer for the
required level of attenuation.

Brennan [56] states that, in controlling the frequency of a neutralizer, there is a
slight time delay while the control system adjusts the neutralizer to the new frequency.
Three key parameters are explained in this article about how tunable vibration
neutralizers act. Increasing the mass ratio, \( \mu = \frac{m_n}{m_s} \), where \( m_n \) is the neutralizer mass
and \( m_s \) is the host structure mass, increases the potential attenuation when the neutralizer
is tuned to the excitation frequency of the host structure. The bandwidth is proportional
to the damping of the neutralizer, so an increase in damping widens the frequency range
in which the neutralizer attenuates but reduces the maximum attenuation. The third
parameter defined is the critical frequency, which is the point where the host structure
vibrates exactly the amount that it would if no neutralizer were attached. Above this
critical frequency, the neutralizer actually has the negative effect of producing more
vibrations in the host structure. The difference between the tuned frequency and this
critical frequency is directly proportional to the mass ratio. Therefore, an increase in the
mass ratio increases the frequency range between the tuned frequency and the frequency
above which the neutralizer begins detrimentally affecting the vibrations in the host
structure. See Figure 1.2 for a plot of the response of a vibrating structure with a neutralizer attached.

![Figure 1.2: Change in Displacement of a Machine Fitted with a Neutralizer](image)

The resonant frequency of the neutralizer is shown as $\omega_r$, and the upper frequency of the half-power band-width is shown as $\omega_b$.

In short, a control system should be quick as well as accurate to ensure that the neutralizer is reducing vibrations in the system. The mass ratio should be increased as much as possible without impeding on the host structure’s intended function. And, the damping should be high enough to allow the control algorithm to keep up with the changes in frequency without adding vibrations to the system. A higher damping ratio also allows more attenuation of transients in the host structure, but if the damping is too high, attenuation at the tuned frequency may become unacceptably low [56].

Long, et al. [61], tuned a vibration neutralizer by bringing the phase difference between vibrations in a host structure and its neutralizer’s hanging mass to zero. One method to change the resonant frequency of a neutralizer is by using pneumatic bellows to change the amount of air in reservoirs, thereby changing the stiffness of the neutralizer [62], [63]. Another active vibration control method was proposed by Wang and Lai, which, through removal of the near-steady-state vibrations and of the transients, achieved a 98% reduction in vibrational energy in a host structure [64].
Francheck, et al., created an adaptive passive vibration neutralizer that actively tuned by using a helical spring in a spring collar. A DC motor changed the number of coils on the far side of the spring collar, thereby changing the stiffness and the resonant frequency of the neutralizer. A digital feedback tuning method provided a non-directional DC offset for the phase difference requiring a fix to find the correct tuning direction [65]. Work on large beam-style adaptive passive neutralizers is presented in [66] and [67], while A. von Flotow, et al. [68], review examples of the basic types of vibration neutralizers and damping absorbers.

Another vibration control method uses AC electrical signals which drive piezoelectric materials as actuators to actively reduce vibrations in structures. Several examples of active vibration neutralizers using piezoelectric elements are found in [69], [70], and [71], while in [72], a single piezoelectric plate was used for simultaneous measurement and actuation.

1.5 Objectives and Scope

There is an untested adaptive passive vibration neutralizer, and tests will be conducted to see if it will tune. A model of the neutralizer will be developed and compared to experimental trials. Design curves will be included for future development of this style of adaptive passive vibration neutralizer.

A computer control algorithm will developed to automatically tune the neutralizer to the excitation frequency of a host structure. A second tuning algorithm will allow computer control without prior testing of the specific neutralizer being controlled. These tuning algorithms will be compared to the neutralizer with no computer control. Experimental data and a computer simulation will be compared for each of these tuning situations. Not speed, but accuracy in tuning, is the aim of these tests.

Novel low-cost, light-weight vibration neutralizers made of plastic air bubbles will be built and tested. A model for these neutralizers will be developed and compared to experimental results.

Miniaturizing mechanical systems, especially vibration neutralizers will be presented. Future implementation of these neutralizers in computer hard drives will be discussed.
1.6 Thesis Structure

Immediately following this introduction is Chapter 2, which contains the analytical and experimental analyses of two solid-state adaptive passive vibration neutralizers. The tuning of the first vibration neutralizer using two different tuning algorithms is discussed in Chapter 3, including computer simulations to predict its response. In Chapter 4, novel vibration neutralizers using individual bubble-wrap bubbles are modeled and compared to experimental data. Chapter 5 evaluates the analytical model derived in Chapter 2 for applicability to the microscale and nanoscale. This analysis includes an order of magnitude prediction for this neutralizer design at smaller scales. Chapter 6 analyzes a computer hard drive for vibration problems, and conclusions with a summary of future work is in Chapter 7.
Chapter 2: Solid-State Adaptive Passive Vibration Neutralizer

Adaptive passive vibration neutralizers are tunable devices designed to reduce harmonic shaking in a structure through passive, as opposed to active, means. The neutralizers discussed here were designed with piezoelectric actuators that adjust the tension in a membrane. Hence the device can be considered solid-state. See Figure 2.1 below for a diagram of the solid-state adaptive passive vibration neutralizer.

![Solid-State Adaptive Passive Vibration Neutralizer Diagram](image)

The membrane has a mass hanging from it that vibrates when excited by a host structure. A voltage across the piezoelectric plates cause a strain which, since they are
constrained by metal plates, causes a beam-tip deflection at the top of the composite plates. This changes the tension in the membrane, thereby changing the resonant frequency of the hanging mass.

2.1 Natural Frequency of a String

Before delving into the design and testing of the adaptive vibration neutralizer, a review of the basic equations for the elements is presented. This section reviews the natural frequency of a string. Since the neutralizer is of uniform depth, the natural frequency of the membrane with a mass in the center (of the same uniform depth) can be simplified to a string with a mass in the center. See Figure 2.2 for a diagram of this string model.

![String-mass diagram](image)

The restoring force for the strings is

\[ F = -2T_s \sin \theta = -2T_s \theta \]  \hspace{1cm} (2.1)

where:

- \(T_s\) is the total tensile force in the string, and
- \(\theta\) is the angle between the stretched string and its horizontal resting situation, assuming small angles.

The total tensile force is

\[ T_t = T_0 + T \]  \hspace{1cm} (2.2)

where:

- \(T_0\) is the initial tensile force in the string, and
- \(T\) is the tensile force in the string due to the strain caused by the voltage.

The restoring force for the mass is
\[ F = m\ddot{w} = -2T\theta \]  
(2.3)

where:

- \( m \) is the mass between the strings, and
- \( w \) is the displacement of the hanging mass in the z-direction.

The displacement of the mass is

\[ w = l_x \sin \theta = l_x \theta \]  
(2.4)

where:

- \( l_x \) is the distance from the hanging mass to the beam edges,

again, assuming small angles.

Solving for the angles and substituting (2.4) into (2.3) provides the equation of motion

\[ m\ddot{w} + \frac{2T_f}{l_x} w = 0 \]  
(2.5)

Therefore, the stiffness can be given as \( 2T_f/l_x \). The tension is assumed to be independent of the motion of the string in this model.

According to Kinsler and Frey, [73], the natural frequency of a string is

\[ \omega_n = \sqrt{\frac{2T_f}{l_x m}} = 2\pi f_n \]  
(2.6)

where:

- \( \omega_n \) is the natural frequency in radians-per-second, and
- \( f_n \) is the natural frequency in cycles-per-second.

Substituting (2.2) into (2.6) and rearranging the result provides

\[ f_n = \frac{1}{\pi} \sqrt{\frac{T_0 + T}{2l_x m}} \]  
(2.7)

### 2.2 Second Moment of Area of the Composite Plate

The plates attached to the ends of the membrane are of the same uniform depth. These plates are attached together so that a voltage in the piezoelectric plate will strain against the metal plate and they will bend. The second moment of area, for this composite plate can be calculated using the parallel axis theorem and assuming that the
bending is about the central axis of the metal plate. See Figure 2.3 for a diagram of this composite plate.

![Figure 2.3: Piezoelectric/Metal Plate Showing the Induced Moment of Bending](image)

The direction of the bending is dependant on whether the voltage input is positive or negative.

This second moment of area around the neutral axis is given by [74] as

$$I_c = I_p + A_p d_p^2 + n \left( I_b + A_b d_b^2 \right)$$

(2.8)

where:

- $I_p$ is the second moment of area of the piezoelectric plate,
- $A_p$ is the surface area of the bending side of the piezoelectric plate,
- $d_p$ is the distance from the neutral axis and the centroid of the piezoelectric plate,
- $n$ is the ratio of Young’s Moduli of the plates,
- $I_b$ is the second moment of area of the metal plate,
- $A_b$ is the surface area of the bending side of the metal plate, and
- $d_b$ is the distance between the neutral axis and the centroid of the metal plate.

The second moment of area for the piezoelectric plate is

$$I_p = \frac{1}{12} l_p h_p^3$$

(2.9)

where:

- $h_p$ is the thickness of the piezoelectric plate.
The area, \( A_p \), is given by

\[
A_p = h_p l_y
\]  
(2.10)

where:

\( l_y \) is the length of the membrane in the y-direction, or the uniform depth of the neutralizer.

The distance, \( d_p \), is given by

\[
d_p = \bar{x} - \frac{1}{2} h_p
\]  
(2.11)

where:

\( \bar{x} \) is the distance to the neutral axis from the outside edge of the piezoelectric plate.

The ratio of Young’s Moduli is

\[
n = \frac{E_b}{E_p}
\]  
(2.12)

where \( n > 1 \).

The second moment of area for the metal plate is

\[
I_b = \frac{1}{12} l_y (2h_b)^3 = \frac{2}{3} l_y h_b^3
\]  
(2.13)

where:

\( h_b \) is the half-thickness of the metal plate.

The area, \( A_b \), is given by

\[
A_b = 2h_b l_y
\]  
(2.14)

This distance, \( d_b \), is given by

\[
d_b = h_p + h_b - \bar{x}
\]  
(2.15)

All of these equations came from [74].

The distance to the neutral axis from the outside edge of the piezoelectric plate is
Combining all of these equations into (2.8) provides the second moment of area of the composite plate around the neutral axis, which is

\[ I_c = \frac{\left(16E_b^2h_b^4 + E_p^2h_p^4 + 8E_bE_ph_bh_p\left(4h_b^2 + 3h_bh_p + h_p^2\right)\right)l_y}{12E_p\left(2E_bh_b + E_ph_p\right)} \]  

(2.17)

### 2.3 Tensile Force in Membrane Due to Tip Displacement

The composite plate induces a tension in the membrane and the membrane displaces an amount. The tension in the membrane is increased or decreased accordingly, and this section examines the tension in the membrane as a function of the beam-tip displacement without an attached membrane. See Figure 2.4 for a diagram showing the displacement variables.

![Beam-Tip Displacement Diagram](image)

Figure 2.4: Beam-Tip Displacement Diagram

The following equation uses the definition of Young’s Modulus to find the force caused by the membrane

\[ E_m = \frac{\sigma_m}{\varepsilon_m} = \frac{T}{A_m} \frac{l_s}{\Delta l_s} = \frac{T}{h_m l_y} \frac{l_s}{\Delta l_s} \]  

(2.18)

where:
\(E_m\) is the Young’s Modulus of the membrane,

\(\sigma_m\) is the stress in the membrane,

\(\varepsilon_m\) is the strain in the membrane,

\(A_m\) is the area of one side of the membrane,

\(\Delta l_x\) is the change in length of the membrane in the x-direction, and

\(h_m\) is the thickness of the membrane.

This equation can be solved for the force

\[
T = \frac{E_m h_m l_y \Delta l_x}{l_x} = (\Delta x - \Delta l_x) k
\]

where:

\(\Delta x\) is the displacement of the beam-tip without a membrane attached, and

\(k\) is the bending stiffness of the beam.

According to Inman [75], the bending stiffness of the beam is

\[
k = \frac{3E_c I_c}{l_z^3}
\]

where:

\(E_c\) is the effective Young’s Modulus of the composite plate, and

\(l_z\) is the height of the composite plate.

The effective Young’s Modulus of the composite plate is the same as the Young’s Modulus of the piezoelectric plate since the neutral axis was calculated with respect to the piezoelectric plate.

Rearranging (2.19) and substituting in (2.20) provides

\[
\Delta l_x \left( \frac{E_m h_m l_y}{l_x} + \frac{3E_c I_c}{l_z^3} \right) = \Delta x \frac{3E_c I_c}{l_z^3}
\]

which simplifies to

\[
\Delta l_x = \left( \frac{E_m h_m l_y l_z^3}{l_x 3E_c I_c} + 1 \right)^{-1} \Delta x
\]

The tensile force as a function of the beam-tip displacement without an attached membrane is
\[ T(\Delta x) = \frac{E_m l_y}{l_x} \left( \frac{E_m l_y l^3}{l_x 3E_c l_c} + 1 \right)^{-1} \Delta x \]  

(2.23)

### 2.4 Beam-Tip Displacement

This section derives the beam-tip displacement of these composite plates as a function of the strain and the geometric constants of the composite plate.

The slope with respect to the strain in the piezoelectric plate from Fuller, [76], is

\[ \frac{\partial^2 x}{\partial z^2} = K' \varepsilon_p \]  

(2.24)

where:

- \( x \) is the displacement of the plate in the x-direction,
- \( z \) is the height of the measurement point in the z-direction,
- \( K' \) is the geometric constant of the composite plate, and
- \( \varepsilon_p \) is the induced strain in the piezoelectric plate.

The strain induced by the piezoelectric plate is given by [76] as

\[ \varepsilon_p = \frac{d_{31} V}{h_p} \]  

(2.25)

where:

- \( d_{31} \) is the piezoelectric strain coefficient of the piezoelectric plate, and
- \( V \) is the voltage across the piezoelectric elements.

The material geometric constant was calculated by using the moment equilibrium about the neutral axis, \( \bar{x} \), and the force equilibrium as described in [76] and is given by

\[ K' = \frac{12E_b E_p h_b h_p (2h_b + h_p)}{\left(16E_b^2 h_b^4 + E_b E_p (32h_b^3 h_p + 24h_b^2 h_p^2 + 8h_b^2 h_p^3) + E_p^2 h_p^4 \right)} \]  

(2.26)

where:

- \( E_b \) is the Young’s Modulus of the piezoelectric plate.

Double integrating equation (2.24) results in

\[ x(z) = K' \varepsilon_p \frac{1}{2} z^2 + C_f \]  

(2.27)

where:
$C_I$ is the constant of integration.

The boundary conditions can be applied to get

$$x(0) = K' \varepsilon_{p} \frac{1}{2} 0^2 + C_I = 0 \Rightarrow C_I = 0$$

which solves for the constant of integration, and

$$x'(0) = K' \varepsilon_{p} 0 = 0$$

which proves that the solution is valid.

The displacement at the tip of the beam (i.e. $z = l_z$) is

$$\Delta x = \frac{1}{2} K' \varepsilon_{p} l_z^2$$

### 2.5 Analytical Model

This section shows the analytical natural frequency as a function of the voltage across the piezoelectric plates followed by a plot of this function.

Equations and (2.22) can be substituted into (2.19) to get

$$T = \frac{3E_x E_m h_m l_x l_y l_z^2 K' d_{31} V}{h_p \left(6E_x l_x l_z + 2E_m h_m l_z^3\right)}$$

which is the tensile force as a function of the voltage across the piezoelectric plates.

Substituting this tensile force into equation (2.7) provides

$$f_n = \sqrt{\frac{1}{2\pi^2 l_m} \left[ T_0 + \frac{3E_p E_m h_m l_x l_y l_z^2 K' d_{31} V}{h_p \left(6E_p l_x l_z + 2E_m h_m l_z^3\right)} \right]}$$

which is the analytical natural frequency of the adaptive passive vibration neutralizer.

Figure 2.5 shows a plot of this analytical model for the second vibration neutralizer prototype.
Though this plot appears to be linear, it is only a relatively linear portion of an exponential curve. The second prototype’s physical properties and experimental voltage range were used in this model. See Appendix A for the MATLAB® code used to create this figure, and see Figure 2.12 for a plot of the experimental data and the analytical model within this voltage range.

### 2.6 Design Curves

Predominant in the analytical model are two related sets of attributes. These attributes are the half-thickness of the metal plate, the thickness of the piezoelectric plate, and the Young’s Moduli of each plate. These attributes are described together as the bending stiffness of the plates in the figures below.

Figure 2.6 and Figure 2.7 show the resonant frequency and change in resonant frequency per volt as functions of the bending stiffness at +200 and -200V levels. All constants used in the analytical model for these curves were taken from the second prototype.
Figure 2.6: Resonant Frequency vs. Normalized Bending Stiffness

Figure 2.7: Change in Resonant Frequency per Volt vs. Normalized Bending Stiffness
In the first figure, the resonant frequency against the bending stiffness ratio, $10\log_{10}(E_p I_p/E_c I_c)$, shows the predicted frequency range between the two voltage values. The bending stiffness curves show a clear design goal, providing the largest tunable frequency range and the largest change in resonant frequency per volt given by the proportion $E_p I_p \equiv 0.020E_c I_c$.

Above this design goal, the piezoelectric plate bending stiffness is much greater than the metal plate bending stiffness and begins to dominate the composite plate. The reverse is true below this design goal. When the piezoelectric plate is much stiffer, then the composite plate will just stretch instead of bending. When the piezoelectric plate is much less stiff, then the composite plate will not deform. The resulting systems cannot achieve as wide a range of tunable frequencies as at the design goal. The curves also show a maximum possible normalized bending stiffness at 1, since the piezoelectric plate can never have a larger bending stiffness than the composite plate.

The previous two figures were calculated at –200 and +200V instead of –420 and +420V since the second prototype had arcing problems at higher voltages, explained later in this chapter. See Appendix A for the MATLAB® code used to create these three figures. The remaining variables in the analytical model simply scale the frequencies produced and are not discussed individually. Chapter 5 contains details on how this model reacts at smaller scales.

2.7 Experimental Setup

The test equipment used in this chapter and in Chapter 3 follows. See Figure 2.8 for a photograph of this setup. An Intel® Pentium® class PC with Visual Basic® 6.0, and two NI™ PCI data acquisition (DAQ) cards, type 6024 and 6714, was used throughout experimentation. The NI™-6024 ran the input channels and the DC output voltage through a NI™-2110 breakout box. The NI™-6713 ran the forcing function through a NI™-2120 breakout box. A second DAQ card and breakout box were necessary since no simple solution could be found to allow multiple DMA channels to be assigned to one DAQ card using ActiveX® in Visual Basic®.
The forcing function signal from the NI™-6713 DAQ card was smoothed to an analog signal by a DL Instruments model 4302 set as a low-pass filter at 500Hz. This was faster than creating ten times as many data points for the forcing function before each test run. Then this filtered signal was amplified by a Rane Corporation® MA6 power amplifier and sent to a Ling Dynamic Systems™ model V208 shaker. A B&K Type 8001 impedance head was mounted on the shaker, and the vibration neutralizer was mounted on top of that with beeswax and some small plastic sheets to electrically insulate the impedance head. A B&K Type 4374 accelerometer was mounted on top of the neutralizer’s hanging mass with beeswax. The sum of these two masses is used as the mass in the computer simulation.

The accelerometer signal and the acceleration signal from the impedance head were channeled through B&K Type 2635 charge amplifiers powered by internal batteries. The signals then went to frequency filters of the type mentioned in the previous paragraph set as high pass filters at 10Hz to remove low frequency noise. From the filters the signals were then sent to the NI™-6024 DAQ card.

The DC voltage output from the DAQ card was split and one side was inverted using an inverting op-amp based on a TL072 integrated circuit (IC). Four 10kΩ resistors, +/- 5% were used in this op-amp: one from the source to the input, one from the input to the negative output, and two in parallel from the positive output to ground. See Figure 2.9 for a schematic and photograph of this inverting op-amp; the two resistors in parallel to the ground are not shown in the schematic.
The IC was powered by an Agilent™ triple output DC voltage power source, model E3630A, set to +/- 15V. The output from the inverter and the original DC signals were then sent to a pair of DC power amplifiers. These power amplifiers were 790 Series amplifiers by AVC™ Instrumentation, a division of PCB® Piezotronics. The signals were then sent to the positive and negative leads of the vibration neutralizer’s composite plates.

2.8 Results

Several different vibration neutralizers were constructed, and two prototypes are discussed in this section.

2.8.1 Prototype Designs

The first prototype was constructed long before this research was conducted, but several design alterations resulted in a second prototype. Table 2.1 shows the physical properties of the two prototypes.
<table>
<thead>
<tr>
<th>Property</th>
<th>Units</th>
<th>Prototype 1</th>
<th>Prototype 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$l_x$</td>
<td>(m)</td>
<td>1.0E-02</td>
<td>1.2E-02</td>
</tr>
<tr>
<td>$l_y$</td>
<td>(m)</td>
<td>1.3E-02</td>
<td>1.3E-02</td>
</tr>
<tr>
<td>$l_z$</td>
<td>(m)</td>
<td>1.5E-02</td>
<td>2.4E-02</td>
</tr>
<tr>
<td>$h_p$</td>
<td>(m)</td>
<td>1.0E-03</td>
<td>5.1E-04</td>
</tr>
<tr>
<td>$h_b$</td>
<td>(m)</td>
<td>1.3E-04</td>
<td>1.3E-04</td>
</tr>
<tr>
<td>$h_m$</td>
<td>(m)</td>
<td>2.8E-04</td>
<td>1.5E-04</td>
</tr>
<tr>
<td>$E_p$</td>
<td>(Pa)</td>
<td>6.6E+10</td>
<td>6.6E+10</td>
</tr>
<tr>
<td>$E_b$</td>
<td>(Pa)</td>
<td>6.9E+10</td>
<td>6.9E+10</td>
</tr>
<tr>
<td>$E_m$</td>
<td>(Pa)</td>
<td>1E+10</td>
<td>1E+08</td>
</tr>
<tr>
<td>$m$</td>
<td>(kg)</td>
<td>5.13E-03</td>
<td>5.61E-03</td>
</tr>
<tr>
<td>$d_{31}$</td>
<td>(m/V)</td>
<td>-1.66E-10</td>
<td>-1.90E-10</td>
</tr>
</tbody>
</table>

Table 2.1: Neutralizer Prototype Physical Properties

The primary difference between the prototypes was that the base of the second prototype was bisected down the y-axis. Each half of the base was screwed into a track to allow for an adjustable pre-tension in the membrane. See Figure 2.10 for a photograph of these two neutralizer prototypes.

![Figure 2.10: a.) The First and b.) Second Vibration Neutralizer Prototypes](image)

In front of the neutralizers is a centimeter-scale ruler to serve as a size reference. A different PZT element thickness and membrane material were used in constructing the second prototype due to their availability.

### 2.8.2 Analytical Model and Experimental Data

Experimental data for both prototypes was collected, testing from −420 to +420V across the PZT plates. The data for the first prototype from −420 to +200V was not
reproducible from trial to trial. See Figure 2.11 for two of the more similar data sets from these trials.

![Figure 2.11: Example of Reproducibility Problem Below +200V in the First Prototype](image)

The two data sets in this figure show the lack of reproducibility up to about +170V. The two sets of data begin to converge above this point and are much more consistent from trial to trial. Since this prototype is apparently not in tension until almost +200V, a trial from +200 to +420V is used in the analysis of this prototype. This lack of repeatability in a large portion of the voltage range was a primary reason for the construction of the second prototype.

The analytical model uses the physical properties given in Table 2.1 in conjunction the analytical model to find the resonant frequency of the neutralizer per voltage value. See Figure 2.12 and Figure 2.13 for plots of the experimental results and analytical predictions for both prototypes.
Figure 2.12: Frequency vs. Voltage for the First Prototype

Figure 2.13: Frequency vs. Voltage for the Second Prototype
For each data set, a first-order curve-fit, “in a least-squares sense” according to MATLAB®, is shown as a dashed line. The curve-fit of the analytical models are nearly imperceptible since the model in this voltage range is nearly linear. The pre-tensile force, $T_0$, was arbitrarily chosen since equipment necessary to measure the strain in the membrane was unavailable. The variance lines, denoted by the dashed lines, are the y-axis +/- delta values calculated by MATLAB® from the curve-fit. See Appendix A for the MATLAB® code used to create these plots.

### 2.8.3 Discussion

Table 2.2 shows the slope of each curve-fit of the experimental results and the analytical predictions.

<table>
<thead>
<tr>
<th>Prototype</th>
<th>Data From</th>
<th>Gradient (Hz/V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>Model</td>
<td>0.0139</td>
</tr>
<tr>
<td>First</td>
<td>Experiment</td>
<td>0.0394</td>
</tr>
<tr>
<td>Second</td>
<td>Model</td>
<td>0.0044</td>
</tr>
<tr>
<td>Second</td>
<td>Experiment</td>
<td>0.0046</td>
</tr>
</tbody>
</table>

Table 2.2: Comparison of Curve-Fits for Near-Linear Data

The gradient comparison shows a good match between the model and the experimental data for the second prototype. The model and experimental data of the first prototype, as the plot suggests, did not agree very well.

Though the second prototype’s model seemed very close to the experimental data. However, the Young’s Modulus of the membrane and the pre-tensile force in the membrane were not known, so testing involving measurement of both of these unknown values would provide further credence to the analytical model.

The second prototype’s frequency range per the voltage input was disappointingly small compared to that of the first prototype. The thinness of the aluminum plates compared to the piezoelectric plates was probably the cause, though this model proved inadequate at predicting the resonant frequencies of neutralizers with much thinner constraining plates. The first prototype, as stated earlier, was built before research began by an unknown party, so the physical properties of the membrane were unknown and the two prototypes cannot be compared further.
2.9 Conclusion

An almost linear trend in frequency against voltage was exhibited in the second prototype by allowing for a pre-tension in the membrane, while a promising frequency range was seen in the first prototype. Arcing occurred at $-420\,V$ during testing of the second prototype, and damage was done to the DC power amplifiers. The thinner PZT elements used in the second prototype should be kept to a $-200$ to $+200\,V$ range in the future to minimize this risk. The resonant frequency range and centered frequency could be shifted by altering the materials or dimensions of the components as per the analytical model.

This chapter clearly showed that the inch-scale adaptive passive vibration neutralizer has a narrow frequency range that can be varied by changing the voltage across the piezoelectric plates in the neutralizer. It is hoped that the bandwidth will be larger at smaller scales.
Chapter 3: Control of a Solid-State Adaptive Passive Vibration Neutralizer

To broaden the number of applications to which the solid-state vibration neutralizer design from the previous chapter may be applied, an adaptive control algorithm is required. This algorithm varies the resonant frequency of the neutralizer in response to a change in excitation frequency in a host structure. Due to the limited frequency range of the second prototype from section 2.8, the first vibration neutralizer prototype is used throughout this chapter.

3.1 Computer Implementation of the Controller

Of the many programming environments available for this project, Visual Basic® 6.0 was chosen. Visual Basic® allows for speedy development of clean program interfaces, and the National Instruments™ ActiveX® library contained all of the necessary input and output components to control DAQ cards. See Appendix B for this Visual Basic® code, and see Figure 3.1 for a screen shot of the program.
The circled controls in Figure 3.1 are explained in section 3.1.1, section 3.1.2, and section 3.1.3; all other controls are explained herein. The Filename field contains the name of the file into which all of the input velocity and output voltage data is stored. The Sampling Frequency (Hz) field is the data collection sampling frequency in Hertz, and the Total Time (s) field holds the amount of time, in seconds, during which data is collected. The Total Points field shows the total number of data points saved to the data file, which is the sampling frequency multiplied by the total time. The Points Between Tuning field is the number of points averaged in each tuning attempt, and the Current Control Voltage (V) field shows the current output voltage.

The radio buttons allow the user to conduct an experiment with the real neutralizer or to run a computer simulation of the experiment. The button to start the program is entitled Start Timer, and the textbox to the right shows changes in the velocity error. Also shown in this textbox is the output voltage and a resonant frequency estimation at this voltage, given by a curve-fit from experimental data. This estimation would have to be calibrated for each neutralizer and is superfluous to the function of the tuning algorithm. However, it was not removed because it will prove useful to the user in certain situations, such as during development of new tuning algorithms.
3.1.1 The Simulation

As explained in section 2.8, the neutralizer is not in tension throughout the entire voltage range, so the analytical model is not always an accurate predictor of the experimental resonant frequency. A third order curve-fit of the experimental resonant frequency against input voltage was used to find the stiffness of the neutralizer membrane for the computer simulation. The analytical model will prove useful in future experimentation with neutralizers of this design but will not be used for the computer simulation in this thesis.

Rao [77] derived a very useful set of equations for use in computer simulations of 2DOF systems, and these equations follow. The equation to find the stiffness of the neutralizer from the aforementioned curve-fit is given as

\[ k_2 = m_2 \left( 2\pi \left( 0.000000472V^3 - 0.000402V^2 + 0.13V + 40.7 \right) \right)^2 \]

(3.1)

where:

- \( m_2 \) is the mass of the neutralizer, and
- \( V \) is the voltage across the PZT elements.

The first term of this equation appears insignificantly small, but it is necessary to include since this term at +/- 400 \( V \) has significant impact on the stiffness of the neutralizer.

The damping constant of the neutralizer is

\[ c_2 = \eta_2 \sqrt{m_2 k_2} \]

(3.2)

where:

- \( \eta_2 \) is the loss factor of the neutralizer.

The acceleration of the host structure is

\[ \ddot{x}_1 = \frac{F - (c_1 + c_2) \dot{x}_{1_{n-1}} + c_2 \dot{x}_{2_{n-1}} - (k_1 + k_2) x_{1_{n-1}} + k_2 x_{2_{n-1}}}{m_1} \]

(3.3)

where:

- \( F \) is the current force being applied to the system,
- \( c_1 \) is the damping constant of the host structure,
- \( \dot{x}_{1_{n-1}} \) is the previous iteration’s velocity of the host structure,
- \( \dot{x}_{2_{n-1}} \) is the previous iteration’s velocity of the neutralizer,
\( k_1 \) is the stiffness of the host structure,
\( x_{1,n-1} \) is the previous iteration’s displacement of the host structure,
\( x_{2,n-1} \) is the previous iteration’s displacement of the neutralizer, and
\( m_1 \) is the apparent mass of the host structure.

The acceleration of the neutralizer mass is
\[
\ddot{x}_2 = \frac{c_2 (\dot{x}_{1,n-1} - \dot{x}_{2,n-1}) + k_2 (x_{1,n-1} - x_{2,n-1})}{m_2} \tag{3.4}
\]

The change in the forcing function from the previous iteration is
\[
\Delta F_1 = F_{n-1} - F \tag{3.5}
\]

The effective change in forcing function in the host structure is
\[
\Delta \tilde{F}_1 = \Delta F_1 + m_1 \left( \frac{6 \dot{x}_{1,n-1}}{\Delta t} + 3 \ddot{x}_1 \right) + 3 (c_1 + c_2) \dot{x}_{1,n-1} - c_2 \ddot{x}_{2,n-1}) + \frac{\Delta t}{2} ((c_1 + c_2) \ddot{x}_1 - c_2 \ddot{x}_2) \tag{3.6}
\]

where:
\( \Delta t \) is the change in time between iterations.

The effective change in forcing in the neutralizer is
\[
\Delta \tilde{F}_2 = m_2 \left( \frac{6 \dot{x}_{2,n-1}}{\Delta t} + 3 \ddot{x}_2 \right) + 3 (c_2 (\ddot{x}_{2,n-1} - \dot{x}_{1,n-1})) + \frac{\Delta t}{2} (c_2 (\ddot{x}_2 - \dot{x}_1)) \tag{3.7}
\]

The denominator of the inverse effective stiffness matrix is
\[
\tilde{K}_{\text{den}}^{-1} = \frac{\Delta t (3c_2 + k_2 \Delta t) (3c_1 \Delta t + \Delta t^2 k_1 + 6m_1) + 6 (\Delta t^3 c_1 + c_2) + \Delta t^2 (k_1 + k_2) + 6m_1 \, m_2}{\Delta t^2} \tag{3.8}
\]

The inverse effective stiffness matrix equations are given by
\[
\tilde{K}_1^{-1} = \frac{3c_2 \Delta t + \Delta t^2 k_2 + 6m_2}{\tilde{K}_{\text{den}}^{-1}} \tag{3.9}
\]
\[
\tilde{K}_2^{-1} = \frac{3c_2 \Delta t + \Delta t^2 k_2}{\tilde{K}_{\text{den}}^{-1}} \tag{3.10}
\]
\[
\tilde{K}_3^{-1} = \frac{3c_2 \Delta t + \Delta t^2 k_2}{\tilde{K}_{\text{den}}^{-1}}, \text{ and} \tag{3.11}
\]
\[
\tilde{K}_4^{-1} = \frac{3 (c_1 + c_2) \Delta t + \Delta t^2 (k_1 + k_2) + 6m_1}{\tilde{K}_{\text{den}}^{-1}} \tag{3.12}
\]

The change in displacement of the host structure is
\[ \Delta x_{1n} = \tilde{K}_1^{-1} \Delta \tilde{F}_1 + \tilde{K}_2^{-1} \Delta \tilde{F}_2 \]  
(3.13)

The change in displacement of the neutralizer is

\[ \Delta x_{2n} = \tilde{K}_3^{-1} \Delta \tilde{F}_1 + \tilde{K}_4^{-1} \Delta \tilde{F}_2 \]  
(3.14)

The change in velocity of the host structure is

\[ \Delta \dot{x}_1 = \frac{3 \Delta x_1}{\Delta t} - \frac{\dot{x}_{1,n-1}}{2} \frac{\dot{x}_1 \Delta t}{2} \]  
(3.15)

The change in velocity of the neutralizer is

\[ \Delta \dot{x}_2 = \frac{3 \Delta x_2}{\Delta t} - \frac{\dot{x}_{2,n-1}}{2} \frac{\dot{x}_2 \Delta t}{2} \]  
(3.16)

The previous equations are executed in this order in the computer simulation.

This simulation does not run in real-time, but it has a factor, seen in the Factor field in Figure 3.1, which specifies the number of iterations per single data collection point. A factor of 10 means that only one of 10 iterations is collected for tuning. This allows a sampling rate of 1000Hz to give 1000 points per simulated second, though 10,000 iterations of the simulation are executed. This factor is necessary since the damping of the neutralizer is low, and small time intervals are required per iteration to keep the simulation stable.

The tuning algorithm executes in the same manner in real-time as the simulation, the only difference being the actual execution time. On the test computer, the simulation run-time was always many times greater than the real-time execution using the physical neutralizer.

### 3.1.2 The Control Algorithms

The control algorithms found in the “Control Method” drop-down box, shown in Figure 3.1, are used in both the physical neutralizer experimentation and the computer simulation. The voltage range for the PZT elements in the first prototype is +200 to +420V, and this voltage range was explained in section 2.8. Three tuning algorithms are discussed below.

### 3.1.2.1 Voltage Slide “Tuning” Algorithm

To optimize the data-acquisition parameters in the program, a direct, neutralizer specific, tuning algorithm is needed. The first step in creating this direct tuning algorithm
is collecting data with a voltage-slide “tuning” algorithm. This “tuning” method starts at the lowest voltage setting and slowly increments the voltage to the maximum voltage by the end of data collection. This algorithm should not be confused with the other two self-tuning algorithms which automatically tune to a drifting excitation frequency.

### 3.1.2.2 Poly Tuning Algorithm

With a sine wave set to a frequency in the center of the neutralizer’s tuning range, the voltage slide algorithm provides a table of error values and their corresponding voltages. These voltages give an estimate of the voltage changes required to tune the neutralizer. This data is curve-fitted to create the Poly tuning method, shown as “Poly” in Figure 3.1.

### 3.1.2.3 Halving Tuning Algorithm

A robust tuning algorithm is desired to allow for uniform tuning performance of the same types of neutralizers. The Poly method requires calibration for each individual neutralizer for its specific tuning characteristics, which is not possible in all applications. This robust Halving tuning method tunes by halving and doubling a change in voltage where the maximum change in voltage is set at 25 percent of the tunable range of the neutralizer. This tuning algorithm was designed and tested using only the computer simulation due to hardware problems; see section 3.3 for details.

### 3.1.3 The Forcing Functions

The selected forcing function is fully generated before the data acquisition is started during the experimental trials. This is done so that only one call to the forcing function DAQ card is necessary, ensuring continuity in the forcing function throughout each trial. The computer simulation requests each piece of the forcing function as it is needed, though the same code segment is executed with either method.

#### 3.1.3.1 Impulse Forcing Function

The impulse forcing function was originally used to find the resonant frequency of the neutralizer but fell to disuse when later forcing functions proved better at this task. The only variable used by this forcing function is the amplitude of the spike. See Figure
3.1 for the interface to select this “Input Amplitude” and the variables used in the following forcing functions.

### 3.1.3.2 Random Noise Forcing Function

A white-noise random forcing function was used in preliminary tests to find the response of the shaker and to debug the program. Other forcing functions proved more useful to the research, and this forcing function was not used once the program was complete and debugged. This forcing function requires only a maximum input amplitude.

### 3.1.3.3 Sine Wave Forcing Function

The sine wave forcing function produces a sine wave and requires an amplitude and a frequency, listed as “Start Frequency” in the figure. The predominant use for this forcing function is in collecting data to calibrate the Poly tuning method.

### 3.1.3.4 Periodic Chirp Forcing Function

The chirp excites the shaker linearly through a range of frequencies. This algorithm begins by oscillating at the lowest stated frequency until the chirp-start-time, “Start Time,” is reached. The frequency linearly increases to the highest stated frequency, “End Frequency,” by the chirp-end-time, “End Time.” Then the input signal oscillates at that final frequency through the end of the test. This algorithm also keeps the phase information between the frequency shifts to keep the forcing function continuous throughout the entire data set. This forcing function is used to simulate a drifting excitation frequency of a host structure through the tuning tests.

### 3.2 Experimental Setup

The exact same equipment was used in this chapter as in Chapter 2. See section 2.7 for the specific pieces of equipment and the photographs of the setup.

All data collected by the Visual Basic® program in this chapter used the following settings: a sampling frequency of 1000Hz, a total time of 30s, 100 points between tuning (which allowed for tuning every 0.2s) and all simulations used a factor of 25. The simulation results did not change with larger factor values, but instability in the simulation came with smaller factor values. Since all data in this chapter was taken from
the first prototype, only voltages between +200 and +420V were used, as stated earlier. Therefore, a minimum of +200V was across the PZT elements even with no tuning. Simply changing the constants for the maximum and minimum allowable voltage would allow this program to accommodate any similarly designed vibration neutralizer.

3.3 Results

Three tests were conducted to present the tuning capabilities of the neutralizer and the accuracy of the computer simulation.

3.3.1 No Tuning Algorithm

The first test has a chirp signal running from 50 to 65Hz with no tuning algorithm. This chirp frequency range was selected to completely encompass the neutralizer’s tunable frequency range. This chirp started at 4s and the frequency slid from 50 to 65Hz by 26s. As stated earlier, the neutralizer was set to its lowest voltage to keep the membrane in tension throughout the test. See Figure 3.2 for this plot.

Figure 3.2: Periodic Chirp with No Tuning, Experimental and Simulated Data
The computer simulation is close to the experimental data and proves that the simulation is an adequate predictor of the base structure response. The experimental data actually shows better attenuation of the base structure vibrations than the simulation. The chirp input signal shown below the velocity data is the same signal used for all three figures from this Results section.

3.3.2 Poly Tuning Algorithm

The second test runs the chirp using the Poly tuning method, which is a polynomial curve-fit of the RMS error against voltage. A first order curve-fit of the error curve provided from the Voltage Slide was used to simplify computer calculations. See Figure 3.3 for the experimental and simulated data for this tuning test.

![Figure 3.3: Periodic Chirp with the Poly Tuning, Experimental and Simulated Data](image)

The RMS error, shown below the velocity data in Figure 3.3, depicts how the tuning algorithm performed compared to the computer prediction of its performance. The comparison between the computer simulation and the experimental velocities is quite good, and the minor discrepancy in the RMS error plots shows that the simulated
neutralizer has a slightly higher frequency range than the experimental data. The amplitude of vibrations is also higher in the experimental data than in the simulation, as it was with no tuning algorithm.

### 3.3.3 Halving Tuning Algorithm

The Halving tuning method simulation is shown in Figure 3.4.

![Figure 3.4: Periodic Chirp with Halving Tuning, Simulated Data](image)

Due to hardware problems and time limitations, experimental data for the Halving method is not available for comparison. The simulation and experimental data corroborated well in the previous two tests, so it is reasonable to assume that the Halving tuning method would continue this trend. This hardware problem, then, does not invalidate analysis of the tuning algorithms.

### 3.3.4 Summary and Discussion

The vibrational energy of the host structure for each tuning method is a clear way to compare these tuning algorithms. This vibrational energy is given by
where:

\[ T \] is the total time, and

\[ f_s \] is the sampling frequency.

See Table 3.1 for the values given by this equation from the data shown in the previous three plots.

<table>
<thead>
<tr>
<th>Tuning Method</th>
<th>Experimental Work</th>
<th>Simulated Work</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(J)</td>
<td>(dB)</td>
</tr>
<tr>
<td>No Tuning</td>
<td>101.8</td>
<td>20.08</td>
</tr>
<tr>
<td>Poly Tuning</td>
<td>63.39</td>
<td>18.02</td>
</tr>
<tr>
<td>Halving Tuning</td>
<td>N/A</td>
<td>75.33</td>
</tr>
</tbody>
</table>

Table 3.1: Experimental and Simulated Vibrational Energy of the First Prototype

These numbers corroborate the visual inspection that the physical experiment had more attenuation due to tuning than the simulation predicted. The table shows 2.06\(dB\) of experimental energy reduction compared to 1.47\(dB\) of simulated energy reduction. The close energy values of the Poly and Halving tuning methods during simulation also suggest that the experimental Halving and Poly methods would be quite similar.

The Poly tuning method works using a curve-fit of the experimental RMS error against the voltage to get an error of zero at a frequency in the middle of the neutralizer’s tunable range. The Halving method changes the voltage in the correct direction until it slides over the zero-error point, and then the change in voltage halves and doubles back. The Poly tuning method should easily outperform the Halving tuning method in a speed test just by the way that it works. If the Poly tuning method was not well calibrated to the specific neutralizer being tuned, then the Halving method would outperform the Poly method. Also, when calibrating the control algorithms to the individual neutralizers is not feasible, the Halving method is the tuning algorithm of choice.

### 3.4 Conclusion

This chapter showed that this adaptive passive vibration neutralizer could be automatically tuned to a host structure’s drifting excitation frequency. The Halving tuning method was developed so that calibration to a specific neutralizer was no longer
required. The computer simulation also proved a good predictor of the experimental data. This will allow for development of faster and more robust tuning algorithms without risking the experimental hardware through exhaustive testing.
Chapter 4: Air Bubble-Wrap Neutralizer

Before moving on to applications and miniaturization of adaptive passive vibration neutralizers, a novel low cost small-scale vibration neutralizer is discussed. This novel neutralizer is constructed from a piece of packing bubble-wrap and was proposed by Dr. Fuller during a meeting about miniaturization of vibration neutralizers.

Two different sizes of bubble wrap were used to create these neutralizers. The small bubbles are approximately one centimeter in diameter, while the big bubble neutralizers are three centimeters across. The mass plates for each bubble neutralizer is a square metal plate of approximately the same length on a side as their respective bubbles’ diameters.

4.1 Analytical Modeling

To predict the response of bubble neutralizers, an analytical model is required.

4.1.1 Single-Degree-of-Freedom Mobility

A simple model of the air bubble neutralizer would be the mass-spring single-degree-of-freedom (SDOF) system. See Figure 4.1.
Figure 4.1: Single-Degree-of-Freedom Damped Mass-Spring System

The equation of motion for a single degree of freedom damped mass-spring system is

\[ m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f(t) \]  \hspace{1cm} (4.1)

where:
- \( m \) is the mass on the spring,
- \( c \) is the damping constant,
- \( k \) is the spring constant,
- \( x \) is the displacement of the mass,
- \( f_0 \) is the forcing of the host structure, and
- \( t \) is the time.

Assuming that the solution for the response is harmonic, \( x = x_0 e^{i\omega t} \) where \( \omega \) is the driving frequency, then equation (4.1) can be written as

\[ f(t) = \left( i\omega m + c + \frac{k}{i\omega} \right) \dot{x}(t) \]  \hspace{1cm} (4.2)

Transforming to the frequency domain, the mobility is given by

\[ \frac{\dot{X}}{F} = \frac{1}{(im\omega + c + k/i\omega)} \]  \hspace{1cm} (4.3)

Below are two analytical approaches to predicting the stiffness values of the different neutralizers. See Table 4.1 for a summary of these calculations compared with the stiffness values obtained experimentally.

### 4.1.2 Stiffness of a Volume

From Kinsler and Frey [73] came the stiffness estimation using the Helmholtz resonator method, or the volume of air in a cylinder, and is
\[ k_{helm} = \rho_0 c^2 \frac{S^2}{V} = \rho_0 c^2 \frac{(\pi r^2)}{2} = 1.21(343) \cdot \frac{\pi (3.13 \times 10^{-2})}{2} \cdot 1.26 \times 10^{-2} = 5.55 \times 10^5 \text{Nm}^{-1} \quad (4.4) \]

where:
- \( \rho_0 \) is the density of air,
- \( c \) is the speed of sound in air,
- \( h \) is the height of the bubble,
- \( S \) is the “neck” surface area in the resonator model, and
- \( V \) is the volume of air inside the bubble.

See Figure 4.2 for a diagram of this model.

![Diagram of Stiffness of a Volume Model](image)

This model describes the stiffness as a volume of a cylinder of air, ignoring the stiffness of the plastic bubble. The cylinder of air is compressed with a force in the vertical direction, as shown in the figure, and the stiffness can be seen as this downward force divided by the displacement of the air due to this force. It also assumes that the volume of the cylinder does not change. This model overestimates the stiffness since the stiffness of the plastic is not taken into account.

### 4.1.3 Stiffness Using the Definition of Young’s Modulus

The previous method utilized the compression of air as the stiffness mechanism. The Young’s Modulus method assumes that stretching of the plastic causes the neutralizer stiffness.
The definition of Young’s Modulus is

\[ E = \frac{\sigma}{\varepsilon} = \frac{F}{A\varepsilon} = \frac{Fx}{yd\Delta x} \]  

(4.5)

where:

\( \sigma \) is the stress,
\( \varepsilon \) is the strain,
\( F \) is the force being applied over the area which produces the stress,
\( A \) is this area,
\( d \) is the thickness of the membrane of the bubble,
\( y \) is the circumference around the cylinder,
\( x \) is the height of the cylinder, and
\( \Delta x \) is the change in this height.

Hence, the stiffness can be written

\[ k = \frac{F}{\Delta x} = \frac{Eyd}{x} \]  

(4.6)

Substituting physical properties into equation (4.6) gives the big neutralizer’s stiffness,

\[ k_x = \frac{10^8 \pi (3.13 \times 10^{-2}) 1.3 \times 10^{-4}}{1.26 \times 10^{-2}} = 1.0 \times 10^5 \text{Nm}^{-1} \]  

(4.7)

and a small neutralizer’s stiffness,

\[ k_s = \frac{1.8 \times 10^6 \pi (1.15 \times 10^{-2}) 1.3 \times 10^{-4}}{0.38 \times 10^{-2}} = 1.2 \times 10^5 \text{Nm}^{-1} \]  

(4.8)

Since the Young’s Modulus of the plastic was not known precisely, the value of \( 10^8 \text{ Pa} \) was used.

See Figure 4.3 for a diagram of the Young’s Modulus model.
As stated earlier, the definition of Young’s Modulus model for stiffness uses the elasticity of the bubble plastic as the stiffness mechanism. As shown in equation (4.6), the stiffness is the force on the walls of the cylinder over the change in height of the cylinder. This model ignores the compression of air inside the bubble which causes it to overestimate the stiffness of the bubble.

4.1.4 Combination of Models

Since these stiffness mechanisms will act as springs in the final mass-spring model, they can be combined by adding their stiffness values in parallel. The equation to combine two parallel springs is

\[ k_{\text{parallel}} = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2}} \]  

(4.9)

Combining the volume of a cylinder method with the Young’s Modulus method for a large bubble would provide

\[ k_{\text{parallel}} = \frac{1}{\frac{1}{5.55 \times 10^5} + \frac{1}{1.0 \times 10^5}} = 8.5 \times 10^4 \text{Nm}^{-1} \]  

(4.10)

With the preliminary model calculations completed, the next step is to create and test some actual neutralizers to see if they perform as the predictions suggest.

4.2 Construction of the Neutralizers

The neutralizers themselves were simple to construct. Two sizes of bubble-wrap were used, as stated in section 4.1.
4.2.1 General

A transparent plastic bubble-wrap was used, with a diameter of just over one centimeter (1.15 cm), which is referred to as a small neutralizer throughout the rest of this chapter. The other bubble-wrap neutralizer was created from a three-centimeter (3.13 cm) diameter translucent pink plastic bubble-wrap. This is referred to as a big neutralizer. See Figure 4.4 for a photograph of these two neutralizers.

![Figure 4.4: The Small and Big Bubble-Neutralizers](image)

Each mass was attached to a single bubble with 3M™ spray-glue. This glue was also used to attach the bubble neutralizers to the shaker-plate during testing. (However, packing tape was used in the photograph.) See Figure 4.5 for a photograph of the shaker with a small neutralizer mounted to the top.

![Figure 4.5: A Small Bubble-Neutralizer Mounted on a Shaker](image)

4.2.2 Big Bubble Neutralizers

A 50.39 g lead mass was used on a big bubble that was 1.26 cm high, more resembling a cylinder than a bubble after construction. This was anticipated in the stiffness predictions in section 4.1. An accelerometer was used to test this neutralizer;
therefore the total weight in the following sections is listed as 51.04g. Using the laser vibrometer, a lighter aluminum mass of 1.97g was also tested on a big bubble.

4.2.3 Small Bubble Neutralizers
The small neutralizers were created with square aluminum plates glued to their tops. The length of each of these square pieces of aluminum was roughly equal to the diameter of the small bubbles. The three masses were 0.135g, 0.152g, and 0.162g, each with a cylindrical height of about 0.38cm.

4.3 Experimental Setup
After creating the bubble neutralizers, an experiment to extract the resonant frequencies was required. As stated earlier, two different methods were employed during the testing process.

The two testing setups used to gather data from all of the air bubble-wrap neutralizers were based on a B&K accelerometer and a Polytec PI, Inc. laser vibrometer. The data acquisition equipment and shaker were common between the two setups.

4.3.1 Common Setup
The data acquisition system consisted of a Gateway™ Pentium® class PC, running LabVIEW™ version 5.0, connected to a National Instruments™ SCXI™ system. The SCXI™ system had one model 1001 power and computer connector component, two model 1305 input arrays, and two model 1140 sample and hold amplifiers. Only the first two channels of the first array were used throughout this experiment. See Figure 4.6 for a photograph of this setup.

Figure 4.6: PC and DAQ System Used in Data Collection
The input forcing function was delivered to the neutralizers using a Ling Dynamic Systems™ model V208 shaker. To excite the shaker, a B&K Type 1405 noise generator was used with a Rane Corporation® MA6 power amplifier. A DL Instruments electronic filter, model 4302, was used to filter out the frequencies beyond the scope of this neutralizer study, below $30\text{Hz}$ and beyond $2000\text{Hz}$. This reduced unnecessary input to the shaker and the data acquisition system to improve the signal-to-noise ratio (SNR). Figure 4.7 shows the noise generator, frequency filter, and shaker connected to the data acquisition system.

![Figure 4.7: Noise Generator, Frequency Filter, and Neutralizer on a Shaker](image)

The reference signal used to generate the frequency response function (FRF) was the output from the white noise generator to the shaker before filtering or amplification. The input of the FRF came from the accelerometer or laser vibrometer, as described below.

### 4.3.2 Accelerometer Setup

The accelerometer-based setup used a B&K Type 4374 accelerometer and a B&K Type 2635 charge amplifier running on internal battery power. The accelerometer was attached on top of the lead mass near the center using beeswax. This setup was much simpler and quicker than that of the laser vibrometer, but the mass of the accelerometer posed limitations when testing the masses on the small bubble neutralizers. See Figure 4.8 for a photograph of this setup.
4.3.3 Laser Vibrometer Setup

To allow for neutralizer masses smaller than an accelerometer’s mass, a Polytec PI, Inc. laser vibrometer was used. This system consisted of a Polytec OFV 501 fiber interferometer and an OFV 2600 vibrometer controller, set at $5\text{mm/s}/V$ resolution. The input channel to the DAQ system received velocity data instead of acceleration data as from the accelerometer. The main advantage in using this system was that the minimum mass on the neutralizer bubble was no longer restricted by the mass of the accelerometer.

Mounting the laser vibrometer was the first issue with which to contend. Conveniently, the inner diameter of a standard roll of duct-tape is just smaller than the rim of the shaker and the diameter of the laser housing. Using a duct-tape roll and a drafting-tape roll on top of one another, the laser had a cost-effective structure to hold it steady above the shaker. See Figure 4.9 for a photograph of this setup.
A more complex structure than a pair of tape rolls would allow for a clearer view to aim the laser at the small circles of reflective tape on the top of the neutralizers, but the tape rolls completely blocked this view. To alleviate this problem, the neutralizers were mounted in the very center of the shaker-plate, and the pieces of reflective tape were attached in the very centers of the neutralizers. The LED displays on the vibrometer controller and fiber interferometer, which show the intensity levels of the returning laser beam, then allowed for an accurate line-up of the laser on the reflective tape.

4.4 Results

This section discusses the experimental results obtained from the setup described in section 4.3.

4.4.1 Stiffness of the Neutralizers

The experimental stiffness values were calculated from the resonant frequency, \( \omega_r \), and mass, \( m \), and is given by

\[
k_{b,\text{exp}} = \omega_r^2 m = (117(2\pi))^2 \times 0.05104 = 2.76 \times 10^5 \text{Nm}^{-1}
\]

See Table 4.1 for a comparison of the tested neutralizers and their experimental and analytical stiffness values.
<table>
<thead>
<tr>
<th>Mass (kg)</th>
<th>Reso. Freq. (Hz)</th>
<th>Nat. Freq. (Hz)</th>
<th>Modal Damping Ratio</th>
<th>Experim. Stiffness (Nm$^{-1}$)</th>
<th>Analytical Stiffness Volume (Nm$^{-1}$)</th>
<th>Elasticity (Nm$^{-1}$)</th>
<th>Elasticity &amp; Volume (Nm$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small bubbles</td>
<td>1.35E-04</td>
<td>1590</td>
<td>1640</td>
<td>0.08889</td>
<td>1.43E+04</td>
<td>6.77E+05</td>
<td>1.2E+05</td>
</tr>
<tr>
<td>1.52E-04</td>
<td>1480</td>
<td>1571</td>
<td>0.09002</td>
<td>1.48E+04</td>
<td>6.77E+05</td>
<td>1.2E+05</td>
<td>1.0E+05</td>
</tr>
<tr>
<td>1.62E-04</td>
<td>1620</td>
<td>1692</td>
<td>0.08998</td>
<td>1.83E+04</td>
<td>6.77E+05</td>
<td>1.2E+05</td>
<td>1.0E+05</td>
</tr>
<tr>
<td>Big bubbles</td>
<td>1.97E-03</td>
<td>529</td>
<td>535</td>
<td>0.08047</td>
<td>2.23E+04</td>
<td>5.56E+05</td>
<td>1.0E+05</td>
</tr>
<tr>
<td>5.10E-02</td>
<td>110</td>
<td>117</td>
<td>0.19228</td>
<td>2.76E+04</td>
<td>5.56E+05</td>
<td>1.0E+05</td>
<td>8.5E+04</td>
</tr>
</tbody>
</table>

Table 4.1: Summary of the Analytical and Experimental Bubble Neutralizers

The natural frequencies and modal damping ratios came from the RFP mode summation approximation explained in section 4.4.3. The resonant frequencies were found using the peaks of the experimental FRFs. The resonant frequencies are the damped natural frequencies, and without damping the resonant frequency is the same as the natural frequency. The experimental stiffness values were calculated from equation (4.11) using the natural frequencies. The analytical stiffness values for each neutralizer came from equations (4.4) and (4.7). The combined stiffness models were calculated using equation (4.9) assuming that the stiffness values were springs in parallel.

### 4.4.2 Damping of the Neutralizers

With the stiffness calculations completed, only the damping constant remains to obtain a damped mass-spring analytical model.

To find the damping of the neutralizer from its FRF the half power bandwidth must be determined. The ratio of the bandwidth to the resonant frequency is equal to the loss factor, according to Nashif [78],

$$
\eta = \frac{\omega_2 - \omega_1}{\omega_n}
$$

(4.12)

Figure 4.10 shows part of a FRF for a small bubble neutralizer from which the damping coefficient was obtained.
The difference between the y-axis using $X'/V$ and using the mobility, $X'/F$, is that this voltage, $V$, is the input to the shaker which is directly converted to force, $F$, in the shaker. Therefore, the response of the shaker is included in the experimental data throughout this chapter. All experimental plots show the y-axis as mobility, $X'/V$, due to the direct correlation between the force and this voltage. This also turns the analytical mobility into a 2DOF system, including the shaker mass, damping, and stiffness. This will be discussed in section 4.4.5.

The frequency values used in the calculations below were obtained using MATLAB® plots of the data collected from the laser vibrometer. The damping coefficients for the big and small bubble neutralizers were calculated to be

$$
\eta_b = 2\zeta_b \approx \frac{\Delta f_{-3dB}}{f_n} \approx \frac{558 - 486}{529} \approx 0.136
$$

and

Figure 4.10: Plot Showing the FRF of Small Bubble Neutralizer
\[ \eta_s = 2\zeta_s = \frac{\Delta f_{-3dB}}{f_n} = \frac{1770 - 1395}{1590} \approx 0.236 \]  

respectively, where \( \zeta_b \) and \( \zeta_s \) are the damping ratios.

The damping is smaller in the large bubble with the heavier mass. This may mean that the damping is an inverse function of the size of the bubble, or that the modes may not be separated well enough to measure the damping in this manner.

### 4.4.3 Damping Ratios and Natural Frequencies by Modal Methods

Nashif’s method for calculating damping, explained in section 4.4.2, requires that the modes in the FRFs be well separated. Since they are not, modal methods are used to more accurately find the natural frequencies and damping ratios. According to Iglesias [79], “in general, the Rational Fraction Polynomial Method (RFP) is a more reliable method than the other methods,” meaning the Complex Exponential Method, the Ibrahim Time Domain Method, and the Hilbert Envelope Method. This is why the RFP modal method was selected in this analysis.

The RFP method is a curve-fit of the FRF using the assumption that the form of the FRF is a sum of modes. The RFP will yield the natural frequencies, the damping ratios, and the mode vectors, referred to as the residues by Iglesias. The MATLAB® code used to run this test came from Iglesias as well [79]. Figure 4.11 and Figure 4.12 show the magnitude and phase of the 3DOF RFP mode summation approximation of the experimental FRF.
Figure 4.11: Magnitude of the Experimental and RFP FRFs

Figure 4.12: Phase of the Experimental and RFP FRFs
Some systems are too complex and too nonlinear to be expressed as a summation of modes, but a 2DOF mass-spring system is simple enough that this is not an issue.

As is customary, a 3DOF RFP mode summation is used to find the two degrees of freedom under scrutiny and to allow for a mode beyond the highest frequency in the data set. When the lowest frequency in the data set is not zero, another degree-of-freedom is typically added, as well. The RFP method revealed a set of imaginary modes to give the best fit of the experimental data. These imaginary modes take energy from the system, but are not expressed as peaks, as real modes are, and cannot be modeled in a 2DOF mass-spring system. Also, these imaginary modes are not useful in this analysis and will not be discussed further.

Table 4.2 shows the residues, poles, natural frequencies, and damping ratios of the two real modes given by the RFP method.

<table>
<thead>
<tr>
<th></th>
<th>Residues</th>
<th>Poles</th>
<th>Natural Frequency</th>
<th>Damping Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>First Real Mode</td>
<td>1.088 +/- 2.015i</td>
<td>- 4.615 +/- 2.648i</td>
<td>53.71 Hz</td>
<td>0.1717</td>
</tr>
<tr>
<td>Second Real Mode</td>
<td>- 1.904 +/- 3.289i</td>
<td>- 7.077 +/- 7.830i</td>
<td>1571 Hz</td>
<td>0.09002</td>
</tr>
</tbody>
</table>

Table 4.2: Results of RFP Method on the Bubble Neutralizer

The poles and residues can be used to reconstruct the mode shapes but are not necessary in this research.

4.4.4 Summary

The combined analytical stiffness model provided stiffness values that were four to six times the experimental stiffness values. This may seem to be a large discrepancy, but the Young’s Modulus value was an order of magnitude approximation. With more accurate values for the Young’s Modulus of the plastic, smaller discrepancies would be observed. See Table 4.1 for the values explained in this section.

The damping ratios given by the RFP algorithm were consistent among the smaller masses. However, the higher damping ratio of the heavier mass, on the larger bubble, may be due to the nonlinear dynamics seen in the experimental FRF. The RFP algorithm did not fit that set of data with the accuracy seen while fitting the other data sets. This non-linearity and lack of close curve-fitting means that the RFP mode
summation is inadequate, since this is not a simple enough system to be modeled as a summation of modes.

The bubble loss factors calculated using Nashif’s method, from section 4.4.2, should be twice the values of the modal damping ratios. The damping ratio of the big bubble with the heavier mass, using Nashif’s method, is approximately one-third of the modal damping ratio. Because of the non-linearity of the FRF, stated above, Nashif’s method is more accurate than the modal method for calculating damping in the big bubble with the heavy mass. However, the modal damping ratio of the big bubble with the lighter mass was 1.18 times the damping ratio given by Nashif’s method. This bubble’s FRF was clearly separated, and the RFP method had a good curve-fit, so the modal damping ratio should be used to model this bubble.

The small bubble’s calculated damping ratio was only 1.31 times the modal damping ratio from the RFP algorithm. The height of the resonance peak of these small bubbles is quite low compared to the shaker’s resonance peak, so, even with 1400 Hz of separation between these peaks, the shaker’s resonance affects the bubble’s apparent response. Also, seen on every small bubble FRF is some noise just lower in frequency than the resonant frequency. These two problems caused the damping values of the bubbles with small masses, given by Nashif’s method, to be slightly higher than those given by the RFP algorithm. The modal damping ratio, from the RFP algorithm, is probably a more accurate value to use for the damping in the model for the small bubble neutralizers.

4.4.5 The Two-Degree-of-Freedom Mass-Spring Model

As explained in section 4.4.2, the experimental FRFs were collected as velocity over voltage, $X'/V$. This requires that the system be modeled as a 2DOF system since the shaker and bubble neutralizer responses are both expressed.

The stiffness values and damping values have been collected, and an analytical model for the damped mass-spring system can be constructed and compared to the experimental data.

The equation of motion for a 2DOF system is given by
\[
\begin{bmatrix}
F_1 \\
F_2
\end{bmatrix} = \begin{bmatrix}
A
\end{bmatrix} \begin{bmatrix}
x_1 \\
x_2
\end{bmatrix} \quad (4.15)
\]

The mobility including the shaker response is
\[
\begin{bmatrix}
x_1 \\
x_2
\end{bmatrix} \begin{bmatrix}
F_1 \\
F_2
\end{bmatrix}^{-1} = [A]^{-1} \quad (4.16)
\]

where the matrix, \([A]\), is given by
\[
[A] = \begin{bmatrix}
io m_1 + \frac{k_1 - k_2}{io} + c_1 - c_2 & c_2 + \frac{k_2}{io} \\
-(c_2 + \frac{k_2}{io}) & io m_2 + \frac{k_2}{io} + c_2
\end{bmatrix} \quad (4.17)
\]

Note that these forces, \(F_1\) and \(F_2\), are proportional to the voltage forcing the shaker. The unknowns of the above equation for one of the small bubble neutralizers are a shaker mass, given as
\[
m_1 = 0.21 kg \quad (4.18)
\]
a neutralizer mass, given as
\[
m_2 = 0.152 g \quad (4.19)
a shaker stiffness, given as
\[
k_1 = 23,913 Nm^{-1} \quad (4.20)
a neutralizer stiffness, given as
\[
k_2 = 14,809 Nm^{-1} \quad (4.21)
a shaker damping constant, given as
\[
c_1 = \eta_1 \sqrt{k_1 m_1} = 2 (0.1717) \sqrt{23,913 Nm^{-1} 0.21 kg} = 24.3 kg / s \quad (4.22)
\]
and a neutralizer damping constant, given as
\[
c_2 = \eta_2 \sqrt{k_2 m_2} = 2 (0.09002) \sqrt{14,809 Nm^{-1} 0.152 (10)^{-3} kg} = 0.270 kg / s \quad (4.23)
\]

where the loss factors, \(\eta_1\) and \(\eta_2\), used to calculate these damping constants were given in Table 4.2 from the RFP mode summation.

See Figure 4.13 and Figure 4.14 to see the experimental FRF and the analytical mobility given by equation (4.16) using the constants from the preceding equations. As
stated earlier, the laser vibrometer measured the response of the mass on top of the bubble neutralizer for this experimental FRF.

Figure 4.13: Magnitude of Experimental vs. Analytical Mobility
These plots show that the bubble-wrap neutralizers can be modeled as damped 2DOF mass-spring systems. The first degree-of-freedom is the shaker response since the input channel to the DAQ system was the input voltage to the shaker. The second degree-of-freedom is the vertical motion of the neutralizer mass. Appendix C contains the MATLAB® code used to calculate and plot these FRFs.

In these plots, the analytical mobility seems to be affected by a frequency filter set too close to the testing range. No frequency filter was between the laser vibrometer and the DAQ system, and frequency filters in computer NI™ DAQ cards tend to have a much higher decibel-per-octave drop than the couple of decibels-per-octave shown in Figure 4.13. Therefore, this un-modeled drop in response seen in the experimental data was most likely caused by the low-pass filter set at 2000Hz between the white noise generator and the shaker. As previously stated, the unfiltered signal from the white noise generator was the second channel to the DAQ system. This filtering problem may have been responsible for the two imaginary modes found by the RFP algorithm.
4.5 Future Testing

Their low cost is an attraction, but designing adaptive passive bubble neutralizers would expand their range of applications, thus making them more viable in vibration control. One such tuning method is based on heating the air in the bubble to change the stiffness of the bubble. Nonlinearities in the plastic with changing heat could become an issue, and the melting point of the plastic would be the upper limit of tuning. Heat escaping the bubble and heating the air outside may cause other nonlinearities. Another method is to change the amount of air in the air bubble to increase or decrease the stiffness. Damping would also increase and decrease with this change, but the bubble plastic strength will be a limiting factor on the frequency range. If space would permit, a framed larger bubble could be fitted around the smaller bubble, without one touching the other. The pressure could be increased or decreased in the larger bubble, changing the resonant frequency of the smaller bubble. This would allow for more precise tuning with less accurate or larger pressure sensors if this becomes a problem. The framed bubble system could allow for air to surround the air bubble neutralizer in space or underwater applications.

The proposed tuning methods all have form-factor and/or reliability issues that would make them un-useable in computer hard drives. These neutralizers could find use in space applications where very lightweight and relatively small vibration neutralizers are the key. Also, using several bubble neutralizers on a host structure with slightly varying resonant frequencies would be useful for an application with a small range of running frequencies as a wideband passive vibration neutralizer. Tuning them is also not impossible; though stronger materials for the bubble may be necessary to have a useful tunable frequency range.

Most future work with these bubble neutralizers will likely be based on large arrays of neutralizers, since a single bubble neutralizer is unlikely to attenuate vibrations substantially.

4.6 Conclusion

These bubble vibration neutralizers are very inexpensive to manufacture. The 2DOF model shown in the previous section was a fair approximation of the experimental
FRF and, with more stringent experimental methods, would show much closer approximations. More funding will allow tuning methods to be developed to expand the range of applicability of these low cost, light-weight neutralizers which should prove most useful for space applications.
Chapter 5: Microsystems and Nanotechnology

This chapter describes miniaturization of mechanical devices, with an emphasis on the vibration neutralizers from Chapter 2.

5.1 Macrosystem to Microsystem Model Considerations

According to Drexler [14], a first approximation of a macroscale design as a microsystem is directly scaleable. Changes to this directly scaled-down model are made when components interact differently at smaller scales. Computer simulations to help find these changes are powerful tools in micromachining.

5.1.1 Computer Modeling to Solve Small-Scale Problems

In 1992, researchers regularly simulated molecular mechanical models consisting of more than ten-thousand atoms using computers. Computing power today allows the number of atoms in a typical simulation to be several orders of magnitude greater.

A molecular mechanical model requires that atomic bonds be well defined and unchanging. The atom types available in these models were limited to a small fraction of the different atoms on the periodic table in 1992, though newer molecular mechanical modeling programs have a much wider selection [14].

London dispersion forces, or the forces attracting electrons to a different atom’s nucleus, can affect the function of nanoscale structures. When these intermolecular...
forces cannot be ignored, a surface continuum approximation is used. Allowing for the limitations of the specific simulations, many nanoscopic systems can be accurately modeled using computer simulations [14]. Some of the specific characteristics not found in macroscale systems that computer simulations can handle are given below.

Simple electrical systems accurately scale from the macroscale when components of the nanoscale structure are large compared to the atomic effect radii of individual atoms. For example, a single-atomic-thickness rod would express different electrical properties between the experimental results and a scaled-down macrosystem model [14].

Small movements in nanomechanical systems can have surprisingly high temperature gradients, which can be detrimental to nanomachines. Changes in molecular properties and permanent damage to the nanostructure can result from excessive thermochemical activity. In addition to thermal energy caused by movement in the structure, photochemical processes from a high intensity photon source, like the sun, can be just as destructive to the life of a nanostructure. When using any photochemically unstable molecules in sunlight, an opaque coating is used over the unstable areas to prevent damage. Most metals, especially aluminum, are good at blocking UV light in this manner, but a large microscopic or even a macroscopic thickness is required in some cases to shield from high energy photons. Sample calculations for these problems are given in [14].

5.1.2 Material Property Changes and Wear at Small-Scales

At the macroscopic level, non-brittle metals, unlike brittle materials, deform elastically even with microscopic discontinuities. Discontinuities can be eliminated with atom-by-atom positional control, and, in general, brittle materials have stronger atomic bonds than metals. Therefore, brittle materials are actually much more useful in molecular manufacturing than metals. Several carbon-based molecules are among the strongest molecules to use as the backbone of a nanostructure, which bond covalently to several neighboring molecules at the same time. Some other elements that are frequently used in small-scale structures are oxygen, nitrogen, silicon, phosphorus, sulfur, and boron. All of these atoms have multiple valence electrons to share with neighboring atoms [14].
The tensile fracture strength in structures is enhanced further through the use of multilayering films. In Cammarata [80], equal thickness layers of aluminum and copper were attached to one another. When the total thickness was decreased from 2000\(nm\) to 140\(nm\), the tensile fracture strength was “significantly enhanced”.

Contaminants that increase wear are removed at production and do not reappear until the device wears out. So, wear in a nanoscopic device is not cumulative; the first signs of wear usually signal the end of the device’s usefulness. This precision in construction also allows nanosystems to deform well beyond the breaking point of their macrosystem equivalents. In macroscale systems, the number of parts increases the cost and decreases the reliability. However, in nanosystems the number of molecules in the structure affect the cost and reliability, and part count has little effect [14].

5.2 Miniaturizing the Piezoelectric Neutralizer

A brief qualitative discussion of the effects of miniaturization on the performance of the piezoelectric neutralizer modeled in Chapter 2 is presented in this section. Purely mechanical designs scale well from the macroscopic to the nanoscopic. No design changes are necessary in these designs until the smallest dimension of the structure approaches several times the diameter of a constituent molecule [14]. Typically, this threshold is in the low nanometer scale since most molecules are smaller than a few nanometers.

5.2.1 Problems with Scaling

There are many sources of problems in scaling down macrosystems to describe nanosystems, as discussed in the previous section. To scale down the adaptive passive vibration neutralizer from Chapter 2, each of these sources for problems must be addressed. Changes to the neutralizer’s design may be required if one of the many computer simulation types finds a problem at a particular scale. Since this structure uses no magnetic fields and there are no complex electronics, problems from these areas will not be a factor in scaling down the neutralizer itself. However, computer simulations to test these effects may be necessary for the wiring leading to the neutralizers or for small-scale control systems.
Since there are no sliding parts in this design and the beam-tip deflections are very small, little thermal friction can be introduced to the system. Brittle materials may be more effective than using aluminum from the macroscale design, and the design goal of $E_b \approx 6.7E_p$ from Chapter 2 should still be valid. Finding specific materials for the job will require research, including computer simulations of material performance at the required scales. With no new concerns resulting from molecular interaction simulations, the direct scaling of the macrosystem can be assumed to be valid as an approximation of the nano- or microscale system [14].

### 5.2.2 Calculations for Resonant Frequencies of Miniaturized Neutralizers

Given in Table 5.1 are a few approximations to show the estimated resonant frequencies using the neutralizer model from Chapter 2. The first column tells the relative scale, followed by the rough dimensions used in the calculations, shown as Length on a Side (mm). The Membrane Thickness (mm) and the Tensile Force, $T_0$ (N) are next, followed by the Hanging Mass (kg) to be attached to the membrane. In the last column is the predicted Resonant Frequency (Hz) of the new neutralizer.

<table>
<thead>
<tr>
<th></th>
<th>Length on a Side (mm)</th>
<th>Membrane Thickness (mm)</th>
<th>Tensile Force, $T_0$ (N)</th>
<th>Hanging Mass (kg)</th>
<th>Resonant Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Macroscale</td>
<td>20</td>
<td>1.5x10$^{-1}$</td>
<td>5.0x10$^9$</td>
<td>5.1x10$^{-3}$</td>
<td>7.1x10$^4$</td>
</tr>
<tr>
<td>Microscale</td>
<td>1x10$^{-1}$</td>
<td>7.5x10$^{-2}$</td>
<td>2.5x10$^{-1}$</td>
<td>6.4x10$^{-7}$</td>
<td>6.3x10$^{-1}$</td>
</tr>
<tr>
<td>Microscale</td>
<td>1x10$^{-2}$</td>
<td>7.5x10$^{-4}$</td>
<td>2.5x10$^{-2}$</td>
<td>6.4x10$^{-10}$</td>
<td>2.0x10$^{-3}$</td>
</tr>
<tr>
<td>Microscale</td>
<td>1x10$^{-3}$</td>
<td>7.5x10$^{-5}$</td>
<td>2.5x10$^{-3}$</td>
<td>6.4x10$^{-13}$</td>
<td>6.3x10$^{-6}$</td>
</tr>
<tr>
<td>Microscale</td>
<td>1x10$^{-4}$</td>
<td>7.5x10$^{-7}$</td>
<td>2.5x10$^{-5}$</td>
<td>6.4x10$^{-19}$</td>
<td>6.3x10$^{-7}$</td>
</tr>
</tbody>
</table>

Table 5.1: Predicted Approximate Resonant Frequencies at Different Sizes

See Appendix D for the MATLAB$^\text{®}$ code used to calculate this table. These values for the resonant frequency were calculated using equation (2.32). To simplify calculations the lengths of the sides of the neutralizer were set to the same value. The pre-tension in the membrane, all of the geometric dimensions, and the voltage across the piezoelectric plate were scaled down at the same rate as the lengths of the sides of the neutralizer. The hanging mass was scaled down as the cube of the change in the length of a side.
5.2.3 Scaling of Control Electronics

The size and placement of the controlling system is also an issue at small scales, and solutions depend upon the exact design complications of each application. Using a series of tunable miniaturized neutralizers, as opposed to the parallel system described in section 1.4, would be preferable if broadband coverage is desired. Depending on space and power restrictions of the application, either a common control system or individual controlling systems would be used. If a single controller design is required to control the array, then each neutralizer in the series would be pre-tuned across a range of frequencies. The single controller could tune them through its frequency range to achieve maximum vibration reduction giving both broadband and tunable coverage. This would ensure that the neutralizer system is always removing vibrations from the host structure, even during the lag time in the control system.

5.2.3.1 Transducers

Accurate transducers on the neutralizer base and the hanging mass are required for tuning control. Threshold accelerometers are not suitable; even large arrays of threshold accelerometers do not have the sensitivity required to note small phase differences between the two input signals. Less massive accelerometers are also not very sensitive. So, the accelerometers may have to be built into the neutralizer to keep excess mass to a minimum while allowing as much sensitivity as the control system requires.

5.2.3.2 Control Logic

Follower electronics can be built at almost any scale, though cost will become a serious issue at smaller scales. To keep costs low, the integrated circuits used to perform the tuning control are likely to be larger than the neutralizers being controlled. Hence an integrated device will have a very low mass ratio and will not have as positive an effect in reducing vibrations. Without integrating the control system, the neutralizer will no longer be an add-on device and will likely have to be designed into the structure to allow for control placement and wiring.

5.2.3.3 Wiring

For extremely small nanoscale neutralizers, electrical wires of just a few atoms in diameter may be necessary. The amount of power required to run the control system may
be too great for these extremely thin wires. Heat dissipation and signal loss may become prominent design issues when using nanowires. There may also be atomic interaction issues, and these systems should be modeled in appropriate computer simulations before being built.

If the control system is not located on the neutralizer, then wiring may have to be etched into the host structure. This will be necessary with moving parts near the nanowires to keep them from being damaged.

See section 6.5 for a description of how such a system could be implemented in a computer hard drive.

5.2.4 Discussion

In cases where component dimensions are close to a single molecule in thickness, extensive molecular modeling tests should be run. The smallest dimension shown in Table 5.1 is many times the width of a constituent molecule, so these values should be valid, as is.

In small nanoscale ranges, new methods, like the aforementioned microgrooves, are required to create the composite plates. Also, as the neutralizer becomes small, higher resonant frequencies are exhibited. For a more thorough discussion of the differences between nanoscale properties of elements and the macroscale properties, see [81].

The predicted resonant frequencies of the nanoscale neutralizers may seem too high to be useful. However, neutralizers with these properties may become quite useful as microscale and large nanoscale machines become more prevalent in consumer markets.

5.3 Conclusion

This chapter presented the scaling of a macroscale model to the micro- and nanoscales, including calculations to predict the approximate resonant frequencies at these scales. Also covered were the many pitfalls of miniaturizing macroscale systems and their design tools. Most of these tools were based on rigorous computer simulation testing. Other problems are likely to occur at these small scales and will require other
design changes to allow for optimal performance, but the construction of microscale adaptive passive vibration neutralizers is physically feasible.
Chapter 6: Vibration of a Computer Hard Drive

Computer hard drives have advanced by orders of magnitude in capacity and speed over the past four decades. Many problems continually challenged this advancement, and at the root of the problems were stability and accuracy. The settling time is the time it takes for the read-write head to jump to a track and stop vibrating enough to begin reading the requested data. This delay is one of the key factors in the average access time of a hard drive and is the factor that can be affected most directly by vibration control. Through vibration control, the actuator arm’s speed could be increased while lowering the settling time, thereby decreasing the average seek-time of the hard drive. Increased stability of the platters and spindle will also allow for more accuracy and therefore faster speeds in data transfer [8]. Through the control of these two key vibration problems, faster, more compact hard drives can be manufactured for today’s miniaturizing computer industry.

6.1 How a Hard Drive Functions

Typically, hard drives have multiple platters, which are simply the discs upon which data is stored. Each platter is partitioned into tracks and then into sectors. A simplified diagram, Figure 6.1, shows the relationship between the tracks that run around
a hard drive and the sectors in each track. These sectors are made up of a set number of bytes to which there are 8 bits per byte [7].

![Diagram Showing the Logical Layout of a Computer Hard Drive](image)

**Figure 6.1: Diagram Showing the Logical Layout of a Computer Hard Drive**

Sample specifications for a hard drive, given by a computer science course [82] at Virginia Tech in the spring of 2000, were for a 16.8 gigabyte drive. This drive had 10 platters, or physical disks, and 10 read-write heads; so, each platter had a capacity of 1.68 gigabytes. There were 13,085 tracks on each platter, 256 sectors around each track, and 512 bytes per sector. With a running speed of 5400 RPMs, the hard drive’s fundamental running frequency would be 90 Hz.

The spindle is attached to each of the platters through their centers. After the initial spin-up of the hard drive, they rotate at a constant speed together. The data is transferred by read-write heads on a voice-coil actuator arm. The arm is attached to a spring to provide negative tension to automatically park the read-write heads when the power is turned off. Parking the heads is necessary to avoid damaging data on the platters during transportation. There are usually read-write heads on both the tops and the bottoms of each arm so that each platter can hold twice the information with only a negligible increase in weight [7]. The read-write head also writes bits by placing a magnetic field over a small section of the magnetic media, setting that small section to either “on” (1), or “off” (0) [8].
The operating system and BIOS keep track of these details so that the hard drive works transparently to the end-user. Nothing more about the operation of a hard drive is necessary to understand the rest of this chapter.

6.2 The History of Hard Drive Vibrations

Computer hard drives are arguably the most important long-term storage devices in personal computers today. The computer hard drive started in the commercial world with the IBM® RAMAC™ type 350 in 1957. This drive consisted of fifty 24-inch rotating platters and one read-write head with a total capacity of 5MB, or over 5 million bytes. This system was leased to users at $130 a month [8]. Now, hard drives are household items having common capacities of up to 80GB, or over 80 billion bytes. A drive of this capacity costs as little as $250 to own, as of the first quarter of 2001, and has only four 3.5-inch rotating platters [83].

6.2.1 The Problem

Since 1961, experts have been saying, “make it smaller, make it lighter, make it more reliable,” with respect to computer hardware designs [84]. Three problems with hard drives have to be overcome to make the necessary leaps up in capacity and down in form factor that consumers require.

The first problem is reducing the flying height of the read-write head over the platter surface. To reduce disk and head wear, the read-write head no longer touches the surface of the platter, but the head needs to be kept as close to the surface as possible. With the read-write head closer to the surface, smaller magnetic fields can be used to manipulate data. This reduces disruption to the surrounding bits of data so that they can be packed in tighter on the platter. The second problem is reducing the head gap size, which is the gap between the two magnetic fields on the read-write head itself. This needs to be kept to a minimum to allow for each bit to take up less space around the track, thereby allowing for larger capacities on disks of the same size. The third problem is reducing the magnetic media thickness. Thinner media allows for smaller magnetic fields to place and detect data, thereby allowing for higher areal densities [8].

As each of these three issues is overcome, vibrations cause accuracy errors limiting the rewards of the innovations. With the flying height minimized, there is less
room for error, and head crashes occur more frequently. To decrease seek-time to the
data, the servos would be made to move faster, but more impact vibrations would be felt
throughout the hard drive. These vibrations limit the performance boosts of the faster
actuators and servos due to track misregistrations (TMRs), or misreads of data [8].

6.2.2 Some Current Solutions

One paper on vibration problems in hard drives compared ball bearings to liquid-
lubricated spiral groove bearings. The spiral groove bearings exhibited potential for
dramatic reductions in spindle vibrations [85]. In [86], a twin-drive actuator for the read-
write arm reduced vibrations to a quarter of the level expressed by single-drive actuators.

Gao, et al. [4], proposed a vibration suppression method using active PVDF
elements embedded in the read-write arms. The interactions between the read-write head
and the hard drive platter are reduced by using velocity feedback control to excite PVDF
elements. They simultaneously reduced residual vibrations from impacts and vibrations
at the hard drive’s excitation frequency.

In spite of the vibration problems, hard drives have become smaller in form-factor
and more dense with information every year. IBM® has been shipping a family of
Microdrive™ one inch hard drives for two years; their newest model has one gigabyte of
storage space [87]. Continuing work on vibration reduction in hard drives will allow for
faster transfer speeds and higher areal densities. These advancements are becoming more
necessary with smaller applications, such as palm-top computers, which do not yet come
with hard drives.

6.3 Experimental Setup

The test hard drive was a Maxtor™ model 86480D6, which is a 6,480MB drive,
built in 1998 [88]. See Figure 6.2 for a photograph of the hard drive. This hard drive
was tested using two B&K accelerometers and charge amplifiers powered on internal
batteries. The same type of accelerometers and charge amplifiers were used in Chapter 4.
A computer power-supply and motherboard were used to test the hard drive during the
self-diagnostic start-up. This diagnostic period started with the platter spin-up and was
followed by a read-write head test to ensure that no obstructions could impede normal
hard drive operations. Only with a motherboard properly connected to the power supply
would the hard drive be given power. Otherwise, the hard drive diagnostic was completely independent of the motherboard.

![Figure 6.2: Overhead View of Tested Hard Drive](image)

The aforementioned read-write head test started with a pair of actuator arm full-range motions. A set pattern of seek test movements followed to ensure that the hard drive could successfully seek several sectors as rapidly as is necessary for normal hard drive operations. The hard drive then spun at a steady speed awaiting further instructions from the hard drive controller card which was not connected to the hard drive.

The two accelerometers were placed in the same locations as the circles of reflective tape on the spindle screw and near the end of the read-write arm, as shown in Figure 6.2. The read-write arm accelerometer was later moved to the base of the read-write arm. This replacement of the accelerometer was to show how much hard drive vibration was caused by mass-loading the end of the read-write arm with the accelerometer; this is explained further in section 0.

The reflective tape was used in laser vibrometer measurements; see Chapter 4 for more details about the laser vibrometer used in this experiment. See Figure 6.3 below for the test setup using the laser vibrometer.

![Figure 6.3: Hard Drive Vibration Test Using the Laser Vibrometer](image)
The accelerometer measurements were taken with the hard drive lying horizontally. The laser vibrometer measurements were taken with the hard drive standing vertically, as shown in the photograph above. The situation of the hard drive had no effect on the results since the hard drive is capable of functioning at optimal performance from either position.

### 6.4 Uncontrolled Vibration

The data clearly shows the impact and periodic vibration issues in the tested hard drive. Figure 6.4, Figure 6.5, and Figure 6.6 are 3-D plots showing the diagnostic spin-up of the hard drive as measured from both the spindle and the read-write arm. These plots show the acceleration, as height and shading, with time on the x-axes and frequency on the y-axes. See Appendix for the MATLAB® code used to create these plots.

![Figure 6.4: Acceleration vs. Time vs. Frequency of the Spindle](image)

Figure 6.4: Acceleration vs. Time vs. Frequency of the Spindle
Figure 6.5: Acceleration vs. Time vs. Frequency of the Read-Write Arm

Figure 6.6: Acceleration vs. Time vs. Frequency of Spindle without Mass-Loading
The power to the computer was turned on at roughly two seconds, shown as a broadband impulse on both plots. Note that the power was turned on earlier in Figure 6.6, so the time-scale is shifted from the other two plots. From just above two seconds to about six seconds, the hard drive platter is spinning-up to full speed. During this period, the hard drive tests the spindle motors for speed and to ensure that no obstructions can impede normal hard drive operations. The six to seven second period on the plots show the read-write arm test. During this period, the hard drive tests the full range of motion of the read-write arm, checking for obstructions. From roughly seven seconds to ten seconds, the hard drive spins at its operating speed with the read-write arm stationary.

The first two plots show the mass-loading of the read-write arm, shown as a deeper color and higher acceleration amplitudes, during and after the read-write arm test. During preliminary testing, the accelerometer on the read-write head produced a high pitched whining sound. Unfortunately, this whining was never heard while data was being collected, making further analysis impossible. Figure 6.6 shows lower vibration levels by several decibels across the tested frequency range, which proves that the accelerometer should not be used to measure vibrations on the end of the read-write arm. Data for this third plot was taken from an accelerometer on the spindle with no second accelerometer on the hard drive. Nearly identical results to this third plot were obtained from the laser vibrometer and are not contained herein.

The laser vibrometer was discussed as an option to remove the mass-loading on the end of the read-write arm. The primary issue with this option was how to keep the laser aimed at the reflective tape near the read-write head during the brisk movement of the actuator arm. This issue could not be resolved with the available equipment, though testing the read-write arm horizontally would be a solution if a laser vibrometer with a finer beam width could be used.

On Polytec’s website, they discuss new fiber-optic vibrometers that are ideal for hard drive experimentation. These new vibrometers have greatly decreased “spot sizes” and do not require reflective tape like the vibrometer used in this experimentation. There is also a dual-fiber system which allows for the simultaneous measurement of two points on a structure. This would remove the need for repeatability of movements of the read-
write arm in laser vibrometer experiments [89]. A laser vibrometer of this precision and flexibility would be advantageous to use in any future hard drive vibration testing.

6.5 Adaptive Passive Vibration Neutralizers in Hard Drives

With no alterations, vibration neutralizers of the type discussed in Chapter 2 could be scaled to one millimeter on a side, as shown in Chapter 5, and be used in computer hard drives. A broadband neutralizer of this form factor could be mounted on the hard drive actuator arm to remove the impact vibrations caused by arm movement. These millimeter-sized neutralizers could be mounted on the edges of the 3.5-inch hard drive platters or on the spindle between the platters to provide tunable vibration reduction around the excitation frequency. See Figure 6.7 for a diagram of this system.

![Diagram of Hard Disc Edge-Mounted Vibration Neutralizer System](image)

Mounting vibration neutralizers on the edges of the platters themselves would be more efficient and require smaller masses than installing them on the spindle. This increase in platter stability would allow for lighter and thinner platters and/or for faster running speeds. The only mechanical change required for this last option would be faster spindle motors.

The immediate problem with these edge-mounted neutralizers would be providing power to tune them while the hard drive is spinning. A simple solution would be to etch a small pair of wires into the glass platter, under the magnetic film, in the radial direction. If there are interactions between the electromagnetic fields caused by these wires and
those of the disk’s magnetic medium, then a small percentage of the hard drive’s data-area would remain unused.

Regardless of any problems like this, using miniaturized neutralizers in this way would allow for small hard drives, like laptop hard drives, to be lighter without sacrificing speed, which has been noted to be a concern in the portable computing industry. Laptop hard drives also could be designed with variable speed, based on power restrictions, without worry of data loss through vibrations in the drive. Unfortunately, the excitation frequency of a hard drive, around 100\(Hz\), would be difficult to reach with an adaptive passive vibration neutralizer of the required form-factor.

Many issues would be resolved with better sensing hardware for continued hard drive testing, like the dual-fiber laser vibrometer, and with better control of the hard drive. Assembly programs could be written to allow this kind of control of a hard drive. The mass-loading on the read-write arm would also be alleviated with better sensing equipment, and a more thorough hard drive vibration analysis could take place. Testing the edge-mounted vibration neutralizers would be notably easier with these technological advantages.

This kind of work is definitely possible with a specific application and design problem to overcome. Newer materials and methods may make the edge-mounted vibration neutralizers unnecessary in the next generation of laptop computer hard drives. However, the next step is large capacity long-term storage in handheld computing devices, like personal digital assistants (PDAs). Developing hard drives for these devices at today’s standards for speed, reliability, and capacity with reasonable manufacturing costs will be quite a challenge in the coming years. Vibration reduction will play a key role in the development of such devices.

6.6 Conclusion

Hard drive vibration problems clearly exist, and researchers have been working on solutions since the first hard drives were designed. Much work will be done in the near future on hard drive vibration issues as hard drives become smaller and faster. New testing equipment is making analysis of this problem easier, but describing the problem is only the first step to solving it. This chapter clearly showed one application for
microscale adaptive passive vibration neutralizers making further research economically feasible.
Chapter 7: Conclusions and Further Work

The aim of this research and thesis was to analyze adaptive passive vibration neutralizers to examine their feasibility for small-scale vibration problems.

7.1 Summary

The analytical model of the adaptive passive vibration neutralizer described in Chapter 2 closely resembled the experimental data from the second prototype. Automatically tuning this neutralizer was the topic of Chapter 3, wherein two computer algorithms chased a drifting excitation frequency. The first method used a curve-fit of the error against voltage that required calibration with the specific neutralizer. The second method halved and doubled a change in voltage, not requiring knowledge of specific neutralizer characteristics. This proved a success, and future work on this design of adaptive passive vibration neutralizer and on the halving tuning method are suggested.

The applications for miniaturized adaptive passive vibration neutralizers are virtually endless, but in Chapter 6 the computer hard drive was proven to be a prime candidate. The bubble neutralizers of Chapter 4 proved less useful than was hoped for computer hard drive applications, but may be very useful in space applications due to their light weight. Chapter 5 showed how models of simple mechanical systems can be directly scaled down and analyzed. Construction methods and materials become
increasingly difficult to optimize as scales get smaller, and computer simulations are required to analyze more advanced physical systems and circuitry. This chapter showed that microscale adaptive passive vibration neutralizers are likely to function closely to the analytical model’s predictions.

7.2 Future Work

To further test the adaptive passive vibration neutralizer and the analytical model from Chapter 2, building and testing such a neutralizer with known pre-tension and membrane Young’s Modulus values could be conducted. And, further testing of the speed of the tuning algorithms and devising new schemes would be the next step in the work presented in Chapter 3.

In the end of Chapter 4, making adaptive passive vibration neutralizers using bubble wrap was discussed. Using changes in pressure and temperature in the bubble would allow for frequency tuning. Due to limitations discussed in section 4.5, these kinds of vibration neutralizers may find their widest range of future applications in space-structure design.

Chapter 6 discussed the hard drive vibration problem and proposed using small adaptive passive vibration neutralizers to counteract these vibrations. An edge-mounted millimeter-sized vibration neutralizer was discussed in section 6.5, explaining some of the difficulties and benefits of using such a vibration neutralizer. Section 5.2.3 discussed the control circuitry in such a system.

7.3 Closing

This research and thesis has provided an analysis of a new design of adaptive passive vibration neutralizer, including tuning algorithms for this neutralizer. Miniaturizing these neutralizers was explained, including possible limitations and the role that computer simulations will play in nanoscale versions of this neutralizer. Recent and future work on vibration reduction in hard drives was also discussed. This thesis has not only proved that adaptive passive vibration neutralizers will theoretically function at the microscale but that there are applications for such devices in today’s consumer markets. In short, microscale adaptive passive vibration neutralizers are feasible in today’s consumer devices which warrants further research.
Dr. Feynman [4] once said, “in the year 2000, when they look back at this age, they will wonder why it was not until the year 1960 that anybody began seriously to [miniaturize mechanical and electrical systems.]” Now that miniaturization is being seriously researched to provide smaller consumer products, new small-scale vibration problems will require smaller vibration reduction devices. He titled his talk “There’s Plenty of Room at the Bottom,” by which he meant that with enough innovation, one can always make something just a little bit smaller than before. This approach has only just begun to be applied to vibration control and so is a fruitful field of research with much room to expand.
Appendix A

This appendix contains the MATLAB® code used to plot the figures in Chapter 2.

**MATLAB® Code to Plot Figure 2.5**

```matlab
m = 5.61e-3; % mass of hanging mass and accelerometer (kg)
lx = 1.2e-2; % half length of membrane (m)
ly = 1.3e-2; % depth of composite plate (m)
lz = 2.4e-2; % height of composite plate (m)
hp = 0.508e-3; % thickness of PZT (m)
hb = 0.13e-3; % half-thickness of plate (m)
hm = 0.28e-3; % thickness of membrane (m)
Eb = 6.9e10; % Young's modulus of plate (Pa)
Ep = 6.6e10; % Young's modulus of PZT (Pa)
Em = 1e8; % Young's modulus of membrane (Pa)
d31 = 190e-12; % PZT dielectric constant (C/N)
V = linspace(-420, 420, 1000); % voltage across PZT elements (V)
To = 11.62; % pre-tension (N)

n = Eb / Ep;
Ab = 2*hb*ly;
Ap = hp*ly;
xbar = (Ap/2*hb+n*Ab*(hp+hb))/(Ap + n*Ab);
dp = xbar-1/2*hp;
db = hp+hb-xbar;
Ib = 2/3 * ly * hb^3 + Ab*db^2;
Ip = 1/12 * ly * hp^3 + Ap*dp^2;
Ic = Ip + n * 1b;
Ec = Ep;
Kc = 12*Eb*Ep*hb*hp*(2*hb+hp)/(16*Eb^2*hb^4+Ep^2*hp^4+... 
8*Eb*Ep*hb*hp*(4*hb^2+3*hb*hp+hp^2));
dx = 1/2*Kc*d31*V/hp*lz^2;
dlx = (Em*hm*ly*lz^3/(lx*3*Ec*Ic)+1)^-1*dx;
T = Em*hm*ly/lx*dlx;
f10 = 1/pi*abs(sqrt((To+T)/(2*lx*m)));

figure(5);
plot(V,f10,'k');
hold on;
hold off;
xlabel('Voltage (V)');
ylabel('Resonant Frequency (Hz)');
axis([V(1) V(length(V)) 91.5 95.5])
```

**MATLAB® Code to Plot Figure 2.6 and Figure 2.7**

```matlab
E = 69*10^9; % Eb 69 GPa
Ep = logspace(-2, 2, 1000)*E;
Eb = logspace(2, -2, 1000)*E;
h = 1.3*10^-4; % hb 0.13 mm
hp = logspace(-2, 2, 1000)*h;
```
hb = logspace(2, -2, 1000)*h;
To = 11.62;
V1 = +200;
V2 = -200;

m = 5.61e-3; % mass of hanging mass and accelerometer (kg)
lx = 1.2e-2; % half length of membrane (m)
ly = 1.3e-2; % depth of composite plate (m)
lz = 2.4e-2; % height of composite plate (m)
hm = 0.28e-3; % thickness of membrane (m)
Em = 1.5e7; % Young's modulus of membrane (Pa)
d31 = 190e-12; % PZT dielectric constant (C/N)

n = Eb / Ep;
Ab = 2*hb*ly;
Ap = hp*ly;
xbar = (Ap/2.*hp+n.*Ab.*(hp+hb))./(Ap + n.*Ab);
dp = xbar-1/2*hp;
db = hp+hb-xbar;
Ib = 2/3 * ly * hb.^3;
Ip = 1/12 * ly * hp.^3;
Ic = (Ip + Ap.*dp.^2) + n.*(Ib + Ab.*db.^2);
Kc = 12*(Eb.*Ep.*hb.*hp.*(2*hb+hp))./(16*Eb.^2.*hp.^4+Ep.^2.*hp.^4+...
   8*Eb.*Ep.*hb.*hp.*(4*hb.^2+3*hb.*hp+hp.^2));
dx = 1/2*Kc*d31./hp*lz^2; % actually dx / V
Ec = Ep;
dlx = (((Em*hm*ly*lz^3)./(lx*3*(Ec.*Ic))+1).^-1).*dx;
T = Em*hm*ly/lx*dlx;

ce = T;
a = 2*pi^2*lx*m;
feH1 = abs(sqrt((To + V1*ce)/a)); % resonant frequency
feH2 = abs(sqrt((To + V2*ce)/a));
dfeH1 = ce./(2*a*abs(sqrt((To + V1*ce)/a))); % change in frequency
dfeH2 = ce./(2*a*abs(sqrt((To + V2*ce)/a)));

figure(7)
semilogx(Ep(1)*hp(1)^3/(Ec(1)*(2*hb(1)+hp(1))^3),feH1(1),'ko')
hold on;
semilogx(Ep(1)*hp(1)^3/(Ec(1)*(2*hb(1)+hp(1))^3),feH2(1),'k^')
number = length(feH1)/20; % the number of array elements between highlighted data points
for blah = 1:length(feH1)/number
    semilogx(Ep(blah*number)*hp(blah*number)^3/(Ec(blah*number)*(2*hb(blah*number)+hp(blah*number))^3),feH1(blah*number),'ko')
    semilogx(Ep(blah*number)*hp(blah*number)^3/(Ec(blah*number)*(2*hb(blah*number)+hp(blah*number))^3),feH2(blah*number),'k^')
end
semilogx(Ep.*hp.^3./(Ec.*(2*hb+hp).^3),feH1,'k')
semilogx(Ep.*hp.^3./(Ec.*(2*hb+hp).^3),feH2,'k')
hold off
xlabel('Piezo Plate Bending Stiffness / Composite Plate Bending Stiffness')
ylabel('Resonant Frequency (Hz)')
legend('+200V','-200V',2)
axis([1e-7 1 92.5 94.5]);

figure(8)
MATLAB® Code to Plot Figure 2.12 and Figure 2.13

% First Prototype
m = 5.13e-3; % mass of hanging mass and accelerometer (kg)
xl = 1.0e-2; % half length of membrane (m)
ly = 1.3e-2; % depth of composite beam (m)
lz = 1.5e-2; % height of composite plate (m)
hp = 1.0e-3; % thickness of PZT (m)
hb = 0.13e-3; % half-thickness of plate (m)
hm = 0.15e-3; % thickness of membrane (m) (6/1000")
Eb = 6.9e10; % Young's modulus of plate (Pa)
Ep = 6.6e10; % Young's modulus of PZT (Pa)
Em = 1e10; % Young's modulus of membrane (Pa)
d31 = 166e-12; % PZT dielectric constant (C/N)
V = linspace(200, 420, 1000); % voltage across PZT elements (V)
To = 3.55; % pre-tension (N)

n = Eb / Ep;
Ab = 2*hb*ly;
Ap = hp*ly;
xbar = (Ap/2*hp+n*Ab*(hp+hb))/(Ap + n*Ab);
dp = xbar-1/2*hp;
db = hp+hb-xbar;
Ib = 2/3 * ly * hb^3 + Ab*db^2;
Ip = 1/12 * ly * hp^3 + Ap*dp^2;
Ic = Ip + n * Ib;
Ec = Ep;
Kc = 12*Eb*Ep*hb*hp*(2*hb+hp)/(16*Eb^2*hb^4+Ep^2*hp^4+... +8*Eb*Ep*hb*hp*(4*hb*2+3*hb*hp+hp^2));
dx = 1/2*Kc*d31*V/hp*lz^2;
dlx = (Em*hm*ly*lz^3/(lx*Ec*Ic)+1)^-1*dx;
T = Em*hm*ly/lx*dlx;
f10 = 1/pi*abs(sqrt((To+T)/(2*lx*m)));

Vp = linspace(200,420,23);
fp = [60.6814, 60.7965, 61.0269, 61.1420, 61.6027, 61.8330, 62.1785, 62.2937, 62.7543,... 62.9846, 63.3301, 63.7908, 64.2514, 64.4818, 65.1727, 65.8637, 66.4395, 67.0154...};
67.7063, 67.9367, 68.1670, 68.1670, 68.5125];
% values gathered from experimental data plots

[Pa,Sa] = POLYFIT(V,f10,1);
[Pe,Se] = POLYFIT(Vp,fp,1);
[fa,DELTAa] = POLYVAL(Pa,V,Sa);
[fe,DELTAe] = POLYVAL(Pe,Vp,Se);

figure(13);
temp = 1;
for i = 1:(length(V)/length(Vp)):length(V)
    shortV(temp) = V(round(i));
    shortf10(temp) = f10(round(i));
    temp = temp + 1;
end
plot(shortV,shortf10,'ko',Vp,fp,'k^');
hold on
plot(Vp,fe,'k',Vp,fe+DELTAe,'k:',Vp,fe-DELTAe,'k:');
plot(V,fa,'k',V,fa+DELTAa,'k:',V,fa-DELTAa,'k:');
hold off
xlabel('Voltage (V)');
ylabel('Resonant Frequency (Hz)');
legend('Analytical Model','Experimental Data','Curve-Fits','Variances',2);
axis([200 420 59.4 69.1]);
clear all;
% Second Prototype
m = 5.61e-3; % mass of hanging mass and accelerometer (kg)
lx = 1.2e-2; % half length of membrane (m)
ly = 1.3e-2; % depth of composite plate (m)
lz = 2.4e-2; % height of composite plate (m)
hp = 0.508e-3; % thickness of PZT (m)
hb = 0.13e-3; % half-thickness of plate (m)
hm = 0.28e-3; % thickness of membrane (m)
Eb = 6.9e10; % Young's modulus of plate (Pa)
Ep = 6.6e10; % Young's modulus of PZT (Pa)
Em = 1e8; % Young's modulus of membrane (Pa)
d31 = 190e-12; % PZT dielectric constant (C/N)
V = linspace(-420, 420, 1000); % voltage across PZT elements (V)
To = 11.62; % pre-tension (N)

n = Eb / Ep;
Ab = 2*hb*ly;
Ap = hp*ly;
xbar = (Ap/2*hp+n*Ab*(hp+hb))/(Ap + n*Ab);
dp = xbar-1/2*hp;
db = hp+hb-xbar;
Ib = 2/3 * ly * hb^3 + Ab*db^2;
Ip = 1/12 * ly * hp^3 + Ap*dp^2;
Ic = Ip + n * Ib;
Ec = Ep;
Kc = 12*Eb*Ep*hp*hp*(2*hb+hp)/(16*Eb^2*hp^4+Ep^2*hp^4+...
     8*Eb*Ep*hp*hp*(4*hb^2+3*hb*hp+hp^2));
dx = 1/2*Kc*d31*V/hp*lz^2;
dlx = (Em*hm*ly*lz^3/(lx*3*Ec*Ic)+1)^-1*dx;
T = Em*hm*ly/lx*dlx;
\[ f_{10} = \frac{1}{\pi} \text{abs} \left( \sqrt{\frac{(T_0+T)}{2\pi l x m}} \right) \]

\[
\text{Volts} = [-420 -380 -338 -296 -254 -212 -170 -128 -85.2 -42.6 0.0 42.6 85.2 128 170 212 254 296 338 380 420];
\]

\[
\text{Freqs} = [91.5 91.6 91.7 91.9 92.3 92.6 93.0 93.2 93.5 93.7 93.9 93.8 93.7 94.0 94.2 94.5 94.6 94.8 95.0 95.2 95.3];
\]

% also gathered from experimental data plots

\[
[\text{Pa}, \text{Sa}] = \text{POLYFIT}(\text{V}, f_{10}, 1);
\]

\[
[\text{Pe}, \text{Se}] = \text{POLYFIT}(\text{Volts}, \text{Freqs}, 1);
\]

\[
[\text{fa}, \text{DELTAa}] = \text{POLYVAL}(\text{Pa}, \text{V}, \text{Sa});
\]

\[
[\text{fe}, \text{DELTAe}] = \text{POLYVAL}(\text{Pe}, \text{Volts}, \text{Se});
\]

figure(14);

temp = 1;

for i = 1:(length(V)/length(Volts)):length(V)
    shortV(temp) = V(round(i));
    shortf10(temp) = f10(round(i));
    temp = temp + 1;
end

plot(shortV,shortf10,'ko',Volts,Freqs,'k^');

hold on

plot(Volts,fe,'k',Volts,fe+DELTAe,'k:',Volts,fe-DELTAe,'k:');
plot(V,fa,'k',V,fa+DELTAa,'k:',V,fa-DELTAa,'k:');
hold off

xlabel('Voltage (V)');
ylabel('Resonant Frequency (Hz)');
legend('Analytical Model','Experimental Data','Curve-Fits','Variances',2);
axis([-420 420 91.3 95.7])
Appendix B

This appendix contains the MATLAB® and Visual Basic® code used in Chapter 3.

MATLAB® Code to Plot Figure 3.2

% Periodic Chirp, No Tuning
fn = 'chirp_none_30s_exp'; gn = 'chirp_none_30s_sim_newc';

% Create Chirp Frequency Plot
totaltime = 30; chirpstart = 4; chirpend = 26; % (s)
freqstart = 50; freqend = 65; % (Hz)
points = 1000;
time = linspace(0, totaltime, points);
for i = 1:length(time)
    if i < (chirpstart/totaltime*points)
        freq(i) = freqstart;
    elseif i > (chirpend/totaltime*points)
        freq(i) = freqend;
    else
        freq(i) = freqstart + (freqend - freqstart) * ((i-chirpstart/totaltime*points) /...
            (points-((chirpstart)*2/totaltime*points)));
    end
end
figure(1)
subplot(2,1,1);
eval(['load 'fn,'.cfg;dt = 'fn,'(1);points = 'fn,'(2);']);
eval(['load 'fn,'.dat;x = 'fn,';']);
plot(dt*[0:length(x)-1],x(:,1),'b');
hold on;
eval(['load 'gn,'.cfg;dt = 'gn,'(1);points = 'gn,'(2);']);
eval(['load 'gn,'.dat;x = 'gn,';']);
plot(dt*[0:length(x)-1],x(:,1),'r:');
hold off;
legend('Experimental','Simulation',0);
ylabel('X'' (m/s)');
subplot(2,1,2); plot(time,freq,'k');
ylabel('Freq (Hz)'); xlabel('Time (s)');

MATLAB® Code to Plot Figure 3.3

% Periodic Chirp, "Poly" Tuning
fn = 'chirp_poly_30s_exp'; gn = 'chirp_poly_30s_sim_newc';

% Create Chirp Frequency Plot
totaltime = 30; chirpstart = 4; chirpend = 26; % (s)
freqstart = 50; freqend = 65; % (Hz)
points = 1000;
time = linspace(0, totaltime, points);
for i = 1:length(time)
    if i < (chirpstart/totaltime*points)
        freq(i) = freqstart;
    elseif i > (chirpend/totaltime*points)
        freq(i) = freqend;
    else
        freq(i) = freqstart + (freqend - freqstart) * ((i-chirpstart/totaltime*points) /...
            (points-((chirpstart)*2/totaltime*points)));
    end
end
figure(1)
subplot(2,1,1);
eval(['load 'fn,'.cfg;dt = 'fn,'(1);points = 'fn,'(2);']);
eval(['load 'fn,'.dat;x = 'fn,';']);
plot(dt*[0:length(x)-1],x(:,1),'b');
hold on;
eval(['load 'gn,'.cfg;dt = 'gn,'(1);points = 'gn,'(2);']);
eval(['load 'gn,'.dat;x = 'gn,';']);
plot(dt*[0:length(x)-1],x(:,1),'r:');
hold off;
legend('Experimental','Simulation',0);
ylabel('X'' (m/s)');
subplot(2,1,2); plot(time,freq,'k');
ylabel('Freq (Hz)'); xlabel('Time (s)');
for i = 1:length(time)
    if i < (chirpstart/totaltime*points)
        freq(i) = freqstart;
    elseif i > (chirpend/totaltime*points)
        freq(i) = freqend;
    else
        freq(i) = freqstart + (freqend - freqstart) * ((i-chirpstart/totaltime*points) / ...
            (points-((chirpstart)*2/totaltime*points)));
    end
end

figure(2)
subplot(2,1,1);
eval(['load 'fn,'.cfg;dt = 'fn,'(1);points = 'fn,'(2);']);
eval(['load 'fn,'.dat;x = 'fn,';']);
plot(dt*[0:length(x)-1],x(:,1),b);
hold on;
eval(['load 'gn,'.cfg;dt = 'gn,'(1);points = 'gn,'(2);']);
eval(['load 'gn,'.dat;y = 'gn,';']);
plot(dt*[0:length(y)-1],y(:,1),r);
hold off;
legend('Experimental','Simulation',0);
ylabel('X'' (m/s)');

subplot(2,1,2);
plot(dt*[0:length(x)-1],x(:,4),b);
hold on;
plot(dt*[0:length(y)-1],y(:,4),r);
hold off;
ylabel('RMS Error'); xlabel('Time (s)');

MATLAB® Code to Plot Figure 3.4

% Periodic Chirp, “Halving” Tuning
fn = 'chirp_half_30s_sim_newc';

figure(1)
subplot(2,1,1);
eval(['load 'fn,'.cfg;dt = 'fn,'(1);points = 'fn,'(2);']);
eval(['load 'fn,'.dat;x = 'fn,';']);
plot(dt*[0:length(x)-1],x(:,1),r);
ylabel('X'' (m/s)');

subplot(2,1,2);
plot(dt*[0:length(x)-1],x(:,4),r);
ylabel('RMS Error'); xlabel('Time (s)');
Visual Basic® Code in Chapter 3

See Figure 3.1 for a screen-shot of the interface for this program. Some code provided by National Instruments™ is required for this program to function.

' Declare global variables and constants
Const Pi = 3.1415927

Private PointsInFile As Double ' Number of iterations in file so far
Private a As Variant ' Output File Object
Private fs As Variant ' FileSystem Object

'Controller Constants
Const BigV = 10 ' Top output voltage of card
Const SmallV = 4.9 ' Smallest output voltage that neutralizer (* PowerMultiplier) should see
Const PowerMultiplier = 41.2 ' Voltage amplifier multiplier

'Simulation Constants (which are assigned values when the form is loaded)
Private m1 As Double
Private c1 As Double
Private k1 As Double
Private m2 As Double
Private eta2 As Double

'Controller Global Variables
Private PrevDelV As Double ' Previous voltage step
Private PrevOffset As Double ' Previous DC offset value
Private GrabFirst As Boolean ' This tells if we need more data before tuning
Private Steps As Double ' Number of steps to converge on vibration minimum
Private Volts As Double ' Current voltage across neutralizer (* PowerMultiplier)
Private VelArray1() As Double
Private VelArray2() As Double

'Simulation Global Variables
Private x1p As Double ' Base displacement
Private x2p As Double ' Neutralizer displacement
Private x1dotp As Double ' Base velocity
Private x2dotp As Double ' Neutralizer velocity
Private ttot As Double ' Current simulation time from time = 0s
Private PrevPart As Integer ' What part of the input function we were just in (chirp)
Private PrevPhase As Double ' Phase to remove discontinuities in input functions
Private dt As Double ' 1 / sampling frequency

Private Sub Disable()
  If RealDataOption(0) Then
    Factor.Enabled = False
    FactorL.Enabled = False
  Else ' Simulation
    Factor.Enabled = True
    FactorL.Enabled = True
  End If ' RealData

  If InputFuncOption(3) Then
    "Code continues...
"
StartFreq.Enabled = True
StartFreqL.Enabled = True
EndFreq.Enabled = True
EndFreqL.Enabled = True
StartT.Enabled = True
StartTL.Enabled = True
EndT.Enabled = True
EndTL.Enabled = True
ElseIf InputFuncOption(1) Then
    StartFreq.Enabled = True
    StartFreqL.Enabled = True
    EndFreq.Enabled = False
    EndFreqL.Enabled = False
    StartT.Enabled = False
    StartTL.Enabled = False
    EndT.Enabled = False
    EndTL.Enabled = False
Else
    StartFreq.Enabled = False
    StartFreqL.Enabled = False
    EndFreq.Enabled = False
    EndFreqL.Enabled = False
    StartT.Enabled = False
    StartTL.Enabled = False
    EndT.Enabled = False
    EndTL.Enabled = False
If InputFuncOption(4) Then
    InputAmp.Enabled = False
    InputAmpL.Enabled = False
Else
    InputAmp.Enabled = True
    InputAmpL.Enabled = True
End If
End If
End Sub

Private Sub VoltageChange(NewVolts As Double)
    If RealDataOption(0) Then
        CWAOPoint0.SingleWrite NewVolts
    End If

    'Display the new voltage everytime it changes
    Dim TempStr As Variant
    Dim TempFreq As Variant
    NewVolts = Round(NewVolts * PowerMultiplier, 1)
    If NewVolts = Round(NewVolts, 0) Then
        TempStr = NewVolts & ".0"
    Else
        TempStr = NewVolts
    End If
    If CurrentVoltage.Text <> TempStr & " V" Then
        CurrentVoltage.Text = TempStr & " V"
        CurrentVoltage.Refresh
    End If
End Sub
TempFreq = Round(0.000000472 * (Volts * PowerMultiplier) ^ 3 - 0.000402 * (Volts * PowerMultiplier) ^ 2 + 0.13 * (Volts * PowerMultiplier) + 40.7, 1)
If TempFreq = Round(TempFreq, 0) Then
    TempFreq = TempFreq & ".0"
End If
TestText.Text = TestText.Text & Chr(13) & Chr(10) & TempFreq & " Hz @ " & TempStr & " V w/ " & Format(PrevOffset, "Scientific")
TestText.Refresh
End If
End Sub

'ScaledData contains TuningPoints.Text number of points for both CH0 and CH1
Private Sub CWAI1_AcquiredData(ScaledData As Variant, BinaryCodes As Variant)
    Dim iScan As Integer

    For iScan = LBound(ScaledData, 2) To UBound(ScaledData, 2)
        VelArray1(iScan) = ScaledData(0, iScan)
        VelArray2(iScan) = ScaledData(1, iScan)
    Next iScan

    Process
End Sub

Private Sub CWAI1_DAQError(ByVal StatusCode As Long, ByVal ContextID As Long, ByVal ContextDescription As String)
    If StopTimer.Caption = "Stop Timer" Then
        StopTimer_Click
    End If
End Sub

Private Sub CWAI1_DAQWarning(ByVal StatusCode As Long, ByVal ContextID As Long, ByVal ContextDescription As String)
    If StopTimer.Caption = "Stop Timer" Then
        StopTimer_Click
    End If
End Sub

'Initialization of variables and randomization
Private Sub Form_Load()
    m1 = 0.2 ' Base Mass (kg)
    c1 = 39 ' Base damping coefficient (kg/s) = 2 * eta1 * m1 * wn1
    k1 = 23000 ' Base spring constant (kg/s^2) = wn1^2 * m1
    m2 = 0.006 ' Neutralizer Mass + Accelerometer (kg)
    eta2 = 0.045 ' Neutralizer damping constant
    Randomize
    Set fs = CreateObject("Scripting.FileSystemObject")
    VoltageChange (0)
End Sub
TuningMethod(0).Text = "0. None"

filename1.Text = "c:\neutraliser\All_Tune_Data\newsim_temp1"

Disable
End Sub

'When the program stops, cut output to zero
Private Sub Form_Terminate()
    VoltageChange (0)
End Sub

'This function is the simulated forcing function into the shaker
Private Function InputFunction(t As Double) As Double
    Dim beta
    If InputFuncOption(0) Then ' Impulse
        If t > (9 + 0.1) * dt And t < (11 - 0.1) * dt Then
            InputFunction = InputAmp.Text
        Else
            InputFunction = 0
        End If
    ElseIf InputFuncOption(1) Then ' Sine
        InputFunction = InputAmp.Text * Sin(StartFreq.Text * 2 * Pi * t)
    ElseIf InputFuncOption(2) Then ' Random
        InputFunction = Rnd(InputAmp.Text * 2) - 1 / 2
    ElseIf InputFuncOption(3) Then ' Periodic Chirp
        ' Stuff for Periodic Chirp (from matlab file)
        If t < StartT.Text Then
            PrevPart = 1
            InputFunction = InputAmp.Text * Cos(2 * Pi * StartFreq.Text * t)
        ElseIf t < EndT.Text Then
            If PrevPart = 1 Then
                PrevPart = 2
                PrevPhase = 2 * Pi * StartFreq.Text * StartT.Text
            End If
            beta = (EndFreq.Text - StartFreq.Text) * ((EndT.Text - StartT.Text) ^ (-1))
            InputFunction = InputAmp.Text * Cos(2 * Pi * (beta / (2) * ((t - StartT.Text) ^ (2)) + StartFreq.Text * (t - StartT.Text)) + PrevPhase)
        Else ' No input function
            InputFunction = 0
        End If
    End If
End Function

'This function is the simulated stiffness according to the curve fit of experimental data
Private Function NeuStiff(V As Double) As Double
    Dim neufreq As Double
    neufreq = (0.000000472 * V ^ 3 - 0.000402 * V ^ 2 + 0.13 * V + 43.7)
    NeuStiff = m2 * (2 * Pi * neufreq) ^ 2 ' k2
End Function

Private Sub InputFuncOption_Click(Index As Integer)
    Disable
End Sub

' This function simply takes the gathered data (residing in VelArray1 and VelArray2)
' and calculates the DCOffset from it and spit the data to a file
Private Function CalcDCOffset() As Double
    Dim X As Integer
    Dim BigString As String
    Dim VelDotX As Double
    Dim VelMeanS1 As Double
    Dim VelMeanS2 As Double
    If PointsInFile = 0 Then
        Set a = fs.CreateTextFile(filename1 & ".cfg", True)
        a.WriteLine (Round(1 / SamplingFreq.Text, 6) & ";" & TotalPointsInFile.Text & ";;"
        a.Close
        Set a = fs.CreateTextFile(filename1 & ".dat", True)
    End If
    If StopTimer.Caption = "Stop Timer" Then
        VelDotX = 0
        VelMeanS1 = 0
        VelMeanS2 = 0
        For X = 1 To TuningPoints.Text
            BigString = Format(VelArray1(X), "Scientific") & " " & Format(VelArray2(X), "Scientific") & " "
            & Format(Volts * PowerMultiplier, "Scientific") & " "
            & Format(PrevOffset, "Scientific") & ";
            a.WriteLine (BigString)
            VelDotX = VelDotX + VelArray1(X) * VelArray2(X)
            VelMeanS1 = VelMeanS1 + (VelArray1(X)) ^ 2
            VelMeanS2 = VelMeanS2 + (VelArray2(X)) ^ 2
        Next X
        VelDotX = VelDotX / TuningPoints.Text
        VelMeanS1 = VelMeanS1 / TuningPoints.Text
        VelMeanS2 = VelMeanS2 / TuningPoints.Text
        PointsInFile = PointsInFile + TuningPoints.Text
        Else
            TestText.Text = TestText.Text & Chr(13) & Chr(10) & "Not Finished Executing!"
        End If
    CalcDCOffset = VelDotX / (Sqr(VelMeanS1) * Sqr(VelMeanS2))
End Function

' This is the control algorithm which decides on the new change voltage
Private Sub Process()
    Dim DCOffset As Double
    DCOffset = CalcDCOffset ' This compiles the velocity arrays and gives the new DC Offset
Const BiggestStep = 1.25 ' Largest voltage step that program can try
Const SmallestStep = 0.005 ' Smallest voltage step that card can output
Const ThresholdS = 0.5 * 10 ^ -2 ' Biggest DCOffset to stop tuning due to
Dim nDelV As Double

If Left(TuningMethod(0).Text, 1) = "1" Then
    ' Grab two data sets and using curve-fit to tune
    ' Curve fit equation obtained from Voltage-Slide and a Sine input
    If GrabFirst Then
        GrabFirst = False ' Next time around we'll tune
    Else
        If Abs(DCOffset) < ThresholdS Then
            nDelV = 0
            PrevDelV = -Sgn(DCOffset) * SmallestStep
        Else
            GrabFirst = True
            nDelV = -(78.2631 / PowerMultiplier * DCOffset)
            PrevDelV = -Sgn(DCOffset) * BiggestStep
            Volts = Volts + nDelV
        End If
        PrevOffset = DCOffset
    End If
ElseIf Left(TuningMethod(0).Text, 1) = "2" Then
    ' Grab two data sets then use halving and doubling to tune
    If GrabFirst Then
        GrabFirst = False ' Next time around we'll tune
    Else
        GrabFirst = True
        If PrevOffset = 0 Then
            nDelV = -Sgn(DCOffset) * BiggestStep
        ElseIf Sgn(PrevOffset) <> Sgn(DCOffset) Then ' Just jumped 0 error
            nDelV = -1 / 2 * PrevDelV
        Else ' PrevOffset and DCOffset Signs are same
            If Abs(PrevOffset) < Abs(DCOffset) Then
                nDelV = 2 * PrevDelV
            Else
                nDelV = PrevDelV
            End If
        End If
    End If
End If

' Now constrain nDelV to the min and max step sizes
If Abs(nDelV) > BiggestStep Then
    nDelV = Sgn(nDelV) * BiggestStep
ElseIf Abs(nDelV) < SmallestStep Then
    nDelV = Sgn(nDelV) * SmallestStep
End If
PrevDelV = nDelV
Volts = Volts + nDelV
PrevOffset = DCOffset
End If

ElseIf Left(TuningMethod(0).Text, 1) = "3" Then
Volts = Volts + (BigV - SmallV) * TuningPoints / TotalPointsInFile
PrevOffset = DCOffset

Else ' No Tuning
  Volts = SmallV
  PrevOffset = DCOffset

End If ' Tuning

'Keep the voltage in the range before actually outputting it
If Volts > BigV Then
  Volts = BigV
ElseIf Volts < SmallV Then
  Volts = SmallV
End If

'Update the real neutralizer with the new voltage and update the text's VoltageChange (Volts)

'Stop running if we've done all the points
If PointsInFile >= Int(TotalPointsInFile.Text) Then
  StopTimer_Click
  Beep
End If

End Sub

Private Sub RealDataOption_Click(Index As Integer)
  Disable
End Sub

Private Sub SamplingFreq_Change()
  If IsNumeric(TotalTime.Text) And IsNumeric(SamplingFreq.Text) Then
    TotalPointsInFile.Text = TotalTime.Text * SamplingFreq.Text
  End If
End Sub

'Button to start and stop the tuning timer
Private Sub StopTimer_Click()
  If StopTimer.Caption = "Stop Timer" Then

    If RealDataOption(0) Then
      'Turn off the continuous data-collection
      CWAI1.Stop

      'Turn off the buffer writer
      CWAO1.Reset
    End If

    TestText.Text = TestText.Text & Chr(13) & Chr(10) & " End Time : " & Time
    StopTimer.Caption = "Start Timer"

  End If

End Sub
Form1.MousePointer = 0

DataFrame.Enabled = True
SimulationFrame.Enabled = True

VoltageChange (0)

a.Close

Else
    'Reset General Globals
    GrabFirst = False ' Grab a second one before tuning (if applicable)
    PrevDelV = 0 ' Previous voltage change
    PrevOffset = 0 ' This might need to be changed!
    Volts = (BigV - SmallIV) / 2 + SmallV ' Actual voltage currently being sent to neutralizer
    Steps = 0 ' Number of steps to converge

    'Reset Simulation Globals
    x1p = 0 ' Base displacement
    x2p = 0 ' Neutralizer displacement
    x1dotp = 0 ' Base velocity
    x2dotp = 0 ' Neutralizer velocity
    tot = 0 ' Simulation time elapsed
    dt = 1 / SamplingFreq.Text / Int(Factor.Text) ' dt is 1 / the sampling frequency

    ReDim VelArray1(TuningPoints.Text) As Double
    ReDim VelArray2(TuningPoints.Text) As Double
    PrevPhase = 0 ' For multi-part input functions (ie. chirp)
    PrevPart = 1 ' For multi-part input functions (ie. chirp)

    PointsInFile = 0 ' Number of iterations so far
    TestText.Text = "Start Time: " & Time & ", "

    StopTimer.Caption = "Stop Timer"
    DataFrame.Enabled = False
    SimulationFrame.Enabled = False
    Form1.MousePointer = 13

    'Set the voltage to the middle value
    VoltageChange (Volts)

    If RealDataOption(0) Then
        Const PointsPerSecond = 1000 ' Set in CWAO1 properties page
        Dim BufferSize
        BufferSize = Round(TotalTime.Text * PointsPerSecond, 0)
        ReDim buffer(BufferSize) As Double
        CWAO1.NUpdates = BufferSize ' Total number of updates
        CWAO1.Configure

        'Fill the buffer
        Dim counter As Long
        For counter = 1 To BufferSize
            buffer(counter) = InputFunction(counter / BufferSize * TotalTime.Text)
        Next counter

        'NI property page is set up for Channel 0 and Channel 1 (continuous)
' already, so set the sampling frequency and the number
' of points per data-set to collect and start it
CWA11.ScanClock.Frequency = SamplingFreq.Text
CWA11.NScans = TuningPoints.Text
CWA11.NScansPerBuffer = TuningPoints.Text
CWA11.Configure
CWA11.Start

'Start the output function using the buffer
CWAO1.Write buffer
CWAO1.Start

Else ' Run simulation

'Start the simulation data loop
GrabData

End If

End If
End Sub

'The simulation, itself, which runs the processor after execution
' (not simultaneous processing as with real data and data_acquired function
Private Sub GrabData()
Dim X As Integer

Do While PointsInFile < Int(TotalPointsInFile.Text)

    Steps = Steps + 1

    For X = 1 To TuningPoints.Text * Int(Factor.Text)

        'Start preliminary calculations (simulation of neutralizer)
        Dim dx1 As Double ' Base displacement
        Dim dx2 As Double ' Neutralizer displacement
        Dim dx1dot As Double ' Base velocity
        Dim dx2dot As Double ' Neutralizer velocity
        Dim x1ddot As Double ' Base acceleration
        Dim x2ddot As Double ' Neutralizer acceleration
        Dim c2 As Double
        Dim k2 As Double
        Dim df1 As Double
        Dim dftil1 As Double
        Dim dftil2 As Double
        Dim ktili1 As Double
        Dim ktili2 As Double
        Dim ktili3 As Double
        Dim ktili4 As Double
        Dim ktilidenom As Double

        k2 = NeuStiff(Volts * PowerMultiplier)
        c2 = 2 * eta2 * Sqr(m2 * k2)

        x1ddot = 1 / m1 * (InputFunction(ttot) - (c1 + c2) * x1dotp + c2 * x2dotp - (k1 + k2) * x1p

    Next X

Next X

End Sub
\[ x_{2ddot} = \frac{1}{m_2} \left( c_2 \cdot x_{1dotp} - c_2 \cdot x_{2dotp} + k_2 \cdot x_{1p} - k_2 \cdot x_{2p} \right) \]

\[ df1 = \text{InputFunction}(ttot + dt) - \text{InputFunction}(ttot) \quad \text{Forcing function} \]

\[ df_{til1} = df1 + m_1 \left( \frac{6}{dt} \cdot x_{1dotp} + 3 \cdot x_{1ddot} \right) + 3 \left( c_1 + c_2 \right) \cdot x_{1dotp} - c_2 \cdot x_{2dotp} \]
\[ + \frac{dt}{2} \left( c_1 + c_2 \right) \cdot x_{1ddot} - c_2 \cdot x_{2ddot} \]

\[ df_{til2} = m_2 \left( \frac{6}{dt} \cdot x_{2dotp} + 3 \cdot x_{2ddot} \right) + 3 \left( -c_2 \cdot x_{1dotp} + c_2 \cdot x_{2dotp} \right) \]
\[ + \frac{dt}{2} \left( -c_2 \cdot x_{1ddot} + c_2 \cdot x_{2ddot} \right) \]

\[ k_{tilidenom} = \frac{\left( dt \left( 3 \cdot c_2 + dt \cdot k_2 \right) \cdot \left( 3 \cdot c_1 \cdot dt + \frac{dt^2}{2} \cdot k_1 + 6 \cdot m_1 \right) + 6 \left( \frac{dt}{3} \left( c_1 + c_2 \right) + \frac{dt}{2} \cdot \left( k_1 + k_2 \right) \right) + 6 \cdot m_1 \cdot m_2 \right)}{dt^2} \]

The common part to all the \( k_{tili}'s \), the denominator / \( dt \)

\[ k_{tili1} = \frac{3 \cdot c_2 \cdot dt + \frac{dt^2}{2} \cdot k_2 + 6 \cdot m_2}{k_{tilidenom}} \quad \text{First row, first column} \]

\[ k_{tili2} = \frac{dt \cdot \left( 3 \cdot c_2 + dt \cdot k_2 \right)}{k_{tilidenom}} \quad \text{First row, second column} \]

\[ k_{tili3} = k_{tili2} \quad \text{Second row, first column} \]

\[ k_{tili4} = \frac{dt \cdot \left( 3 \cdot \left( c_1 + c_2 \right) + dt \cdot \left( k_1 + k_2 \right) \right) + 6 \cdot m_1}{k_{tilidenom}} \quad \text{Second row, second column} \]

\[ dx_1 = k_{tili1} \cdot df_{til1} + k_{tili2} \cdot df_{til2} \]

\[ dx_2 = k_{tili3} \cdot df_{til1} + k_{tili4} \cdot df_{til2} \]

\[ dx_{1dot} = 3 / dt \cdot dx_1 - 3 \cdot x_{1dotp} - \frac{dt}{2} \cdot x_{1ddot} \]

\[ dx_{2dot} = 3 / dt \cdot dx_2 - 3 \cdot x_{2dotp} - \frac{dt}{2} \cdot x_{2ddot} \]

<table>
<thead>
<tr>
<th>ttot = ttot + dt</th>
<th>New time</th>
</tr>
</thead>
<tbody>
<tr>
<td>x_1p = x_1p + dx_1</td>
<td></td>
</tr>
<tr>
<td>x_2p = x_2p + dx_2</td>
<td></td>
</tr>
<tr>
<td>x_{1dotp} = x_{1dotp} + dx_{1dot}</td>
<td></td>
</tr>
<tr>
<td>x_{2dotp} = x_{2dotp} + dx_{2dot}</td>
<td></td>
</tr>
</tbody>
</table>

'TEnd preliminary calculations

If \( X / \text{Int(Factor.Text)} = \text{Round}(X / \text{Int(Factor.Text)}, 0) \) Then

\[ \text{VelArray1}(X / \text{Int(Factor.Text)}) = x_{1dotp} \]

\[ \text{VelArray2}(X / \text{Int(Factor.Text)}) = x_{2dotp} \]

End If

Next X

'Tune via the velocity arrays

Process

Loop

End Sub

Private Sub TotalTime_Change()
If IsNumeric(TotalTime.Text) Then
    TotalPointsInFile.Text = TotalTime.Text * SamplingFreq.Text
End If
End Sub
Appendix C

This appendix contains the MATLAB® code used to plot the figures in Chapter 4.

MATLAB® Code to Plot Figure 4.10

```matlab
figure(1)
load small_bub_1_35g_FRF
plot(freq,10*log10(autospec(1,:)*25/1000000),'b');
xlabel('Frequency (Hz)');
ylabel('Mobility, X''/V (dB)');
axis([1367 1826 -45.05 -41.98]);
```

MATLAB® Code to Plot Figure 4.13 and Figure 4.14

```matlab
% Mass 2 FRF
eta2 = 0.090020;  % damping ratio FROM RFP
m2 = 1.52e-4;  % kg FROM measurement
wn2 = 1570.96*2*pi; % rad/s
k2 = wn2^2*m2;  % N/m
c2 = eta2 * 2 * sqrt(k2*m2);

% Shaker 1 FRF
eta1 = 0.171722;  % damping ratio FROM RFP
m1 = 0.21;  % kg FROM Spec Sheet and Calculations
wn1 = 53.707*2*pi; % rad/s FROM RFP
k1 = wn1^2*m1;  % N/m
c1 = eta1 * 2 * sqrt(k1*m1);

% Build Matrices for Analytical Model
w = 2*pi.*linspace(0, 2000, 1000);
v = zeros(2,length(w));
F = [5 ; 0];
for n=1:length(w)
    A = [ i*w(n)*m1+(k1-k2)/(i*w(n))+(c1-c2), (c2+k2/(i*w(n)));
         -(c2+k2/(i*w(n))), i*w(n)*m2+k2/(i*w(n))+c2 ];
    v(:,n) = inv(A)*F;
end

% Experimental Data
load 0_152g_1
freq = freq(1:1365);
autospec = autospec(:,1:1365); % This is the Velocity / Input Voltage (to Shaker)
Exp = autospec(1,:).*exp(i.*autospec(2,:)); % 5 mm/(s V) from vibrometer

figure(1)
plot(freq, 20*log10(abs(Exp*5/1000)),'ko'); % 5 mm/(s V) from vibrometer
hold on;
plot(w./(2*pi), 20*log10(abs(v(2,:))),'r^');
hold off;
legend('Experimental Data','Analytical Neutralizer',0);
xlabel('Frequency (Hz)');
ylabel('Mobility, X", V (dB)');
axis([0 2000 -65 -10]);

figure(2);
factor = 180/2/pi; % 57.29578 degrees/radian
unwrap2 = 970; % Point in experimental phase to unwrap with 0.152g mass
plot(freq(1:unwrap2),factor*(autospec(2,1:unwrap2)-autospec(2,3)),'ko',';
    freq(unwrap2+8:1365),factor*(autospec(2,unwrap2+8:1365)-autospec(2,3))-180,'ko');
hold on;
plot(w./(2*pi),factor*(unwrap(angle(v(2,:)))-angle(v(2,2))),'r^');
hold off;
xlabel('Frequency (Hz)');
ylabel('Phase (Degrees)');
axis([0 2000 -190 0]);
Appendix D

This appendix contains the MATLAB® code used for Chapter 5.

MATLAB® Code to calculate Table 5.1

% Initial Values for Macroscale Line of Table
lx = 10e-3; % half length of membrane (m)
ly = 20e-3; % depth of composite beam (m)
lz = 20e-3; % height of composite plate (m)
hp = 1.5e-3; % thickness of PZT (m)
hb = 1.5e-3; % half-thickness of plate (m)
hm = 1.5e-4; % thickness of membrane (m)
To = 5.0; % pre-tension (N)
m = 5.1e-3; % mass of hanging mass and accelerometer (kg)
V = 400; % voltage across PZT elements (V)
Eb = 6.9e10; % Young's modulus of plate (Pa)
Ep = 6.6e10; % Young's modulus of PZT (Pa)
Em = 1e8; % Young's modulus of membrane (Pa)
d31 = 166e-12; % PZT dielectric constant (C/N)

for pass = 1:5

% Calculate Resonant Frequency
n = Eb / Ep;
Ab = 2 * hb * ly;
Ap = hp * ly;
xbar = (Ap / 2 * hp + n * Ab * (hp + hb)) / (Ap + n * Ab);
dp = xbar - 1/2 * hp;
db = hp + hb - xbar;
Ib = 2/3 * ly * hb^3 + Ab * db^2;
Ip = 1/12 * ly * hp^3 + Ap * dp^2;
Ic = Ip + n * Ib;
Ec = Ep;
Kc = 12*Eb*Ep*hp*hp*(2*hp+hp)/(16*Eb^2*hb^4+Ep^2*hp^4+...
8*Eb*Ep*hp*hp*(4*hb^2+3*hb*hp+hp^2));
dx = 1/2 * Kc * d31 * V / hp * lz^2;
dlx = (Em * hm * ly * lz^3 / (lx * 3 * Ec * Ic) + 1)^-1 * dx;
T = Em * hm * ly / lx * dlx;
f10 = 1/pi * abs(sqrt((To + T) / (2 * lx * m)));

% Display Table Values
pass
ly
hm
To
m
f10
V

% Prepare for next pass
if pass == 1
    num = 20;
elseif pass == 4
    num = 100;
end
else
    num = 10;
end
lx = lx/num; % half length of membrane (m)
ly = ly/num; % depth of composite beam (m)
lz = lz/num; % height of composite plate (m)
hp = hp/num; % thickness of PZT (m)
hb = hb/num; % half-thickness of plate (m)
hm = hm/num; % thickness of membrane (m)
To = To/num; % pre-tension (N)
m = m/num^3; % mass of hanging mass and accelerometer (kg)
V = V/num; % voltage across PZT elements (V)
  % Eb stays the same
  % Ep stays the same
  % Em stays the same
  % d31 stays the same
end
Appendix E

This appendix contains the MATLAB® code used to plot the figures of Chapter 6.

MATLAB® Code to Plot Figure 6.4, Figure 6.5, and Figure 6.6

```matlab
load weber_time_3;
[Bcyl,f,t]= specgram(data_in(:,1),2048,6000);
[Bhd,f,t]= specgram(data_in(:,2),2048,6000);
Bcyl = Bcyl(1:683,:);
Bhd = Bhd(1:683,:);
f = f(1:683,:);

figure(1)
surf(t,f,20*log10(abs(Bcyl))), shading interp
xlabel('Time (s)')
ylabel('Frequency (Hz)')
zlabel('Acceleration (dB)')
view(25,60)

figure(2)
surf(t,f,20*log10(abs(Bhd))), shading interp
xlabel('Time (s)')
ylabel('Frequency (Hz)')
zlabel('Acceleration (dB)')
view(25,60)

load weber_time_5;
[Bcyl,f,t]= specgram(data_in(:,1),2048,6000);
Bcyl = Bcyl(1:683,:);
f = f(1:683,:);

figure(3)
surf(t,f,20*log10(abs(Bcyl))), shading interp
xlabel('Time (s)')
ylabel('Frequency (Hz)')
zlabel('Acceleration (dB)')
view(25,60)
```
References


Vita

Michael A. Weber was born in Quantico, VA on April 12, 1977. He acquired his B.S. degree in Physics from Marshall University in Huntington, WV. Moving to Blacksburg, VA, in the fall of 1999, he began attending Virginia Tech. In January of 2002 he was commissioned as an officer in the United States Navy. The requirements for his M.S. in Mechanical Engineering at Virginia Tech were completed that spring. He will be continuing his education at the Naval nuclear-power training facility in Charleston, SC.