Composite Materials in Dynamic Shipboard Structural Mounts

by

Joanna Faulk

SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING
AT THE
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

JUNE 2011

Signature of Author: ______________________
Department of Mechanical Engineering
May 6, 2011

Certified by: ______________________
James H. Williams, Jr., PhD
Professor of Mechanical Engineering and Writing and Humanistic Studies
School of Engineering Professor of Teaching Excellence, Emeritus
Thesis Supervisor

Accepted by: ______________________
John H. Lienhard V
Samuel C. Collins Professor of Mechanical Engineering
Undergraduate Officer

Composite Materials in Dynamic Shipboard Structural Mounts

by

Joanna Faulk

Submitted to the Department of Mechanical Engineering
On May 6, 2011 in Partial Fulfillment of the
Requirements for the Degree of Bachelor of Science in
Mechanical Engineering

ABSTRACT

The purpose of this thesis is to investigate the viability of replacing traditional metal structural and machinery mounts with padding made of composite material. The two types of padding or isolation materials are represented by steel and CFRP (carbon fiber reinforced polymer). Machinery and instruments in ships are often mounted for two main reasons: they create unwanted vibrations and they need to be isolated from shock and external vibration. In order to analyze this problem, the machinery or instrument plus its padding are modeled as a mass-spring-damper system. The results show that CFRP generally works better for vibration isolation, while steel works better for shock isolation.

Thesis Supervisor: James H. Williams, Jr., PhD
Titles: Professor of Mechanical Engineering and Writing and Humanistic Studies
School of Engineering Professor of Teaching Excellence, Emeritus

Acknowledgements

The author would like to show appreciation to the following people for their contributions in the completion of this thesis:

Professor James H. Williams, Jr., thesis advisor, for his mentoring and guidance.

Anna Haas, David Hawes, and Rachel Lewis, for their assistance in research and computations.

Cedrick and Joanne Faulk, for their continued love and support.

The author is also grateful for the financial contribution of the DDG-1000 Program Manager (NAVSEA PMS 500) and the technical guidance of the DDG-1000 Ship Design Manager (NAVSEA 05D).

Table of Contents

List of Figures

List of Tables

1. Introduction

1.1 Background and Motivation

In electric ships, there are many rotating/oscillating components where vibration is generated and also many sensitive components that need to be isolated from these vibrations. Vibrations within the ship can cause other problems as well. Machinery, instruments, mounts, or even the ship itself can be damaged. These vibrations can also spread to the ocean, making the ship more perceivable by detection devices. Presently, different metals (i.e., steel) are utilized in the structural and machinery mounts. The assessment in this paper explores the possibility of replacing these mounts with composite materials. A composite material is made by physically combining two or more materials encompassing different properties; however, the materials are not combined chemically. Composites may be less stiff than most metals, but they generally are lighter and combine compressive with tensile strength, giving composites a greater appeal. They also tend to be less susceptible to corrosion, making them especially valuable for use in ships where salt water may cause damage in metals.

1.2 Vibration and Shock

This section details the different types of unwanted sound, also known as noise, that are desired to not be transmitted: vibration and shock. Mechanical vibration is defined as the oscillation of an object about a central, equilibrium position. A simple example of this type of oscillation is a mass on an extension spring. In the case that the mass displaced to a point above its rest position, the force of gravity causes the mass to accelerate downward. The restoring force of the spring then pulls the mass back up, and then the cycle repeats. If the mass is first displaced to a point below the equilibrium position, the same steps occur, with the restoring force acting first. This cycle would continue in an ideal situation, but due to the addition of friction and other damping forces, the magnitude of these vibrations decreases until finally the mass no longer moves at all.

Shock is different than vibration, in that it is caused by one single force. The duration of the force is often very short, and the magnitude is very large. Figure 1 below shows the difference between vibration and shock, both of which are explored as possible disturbances to the system.

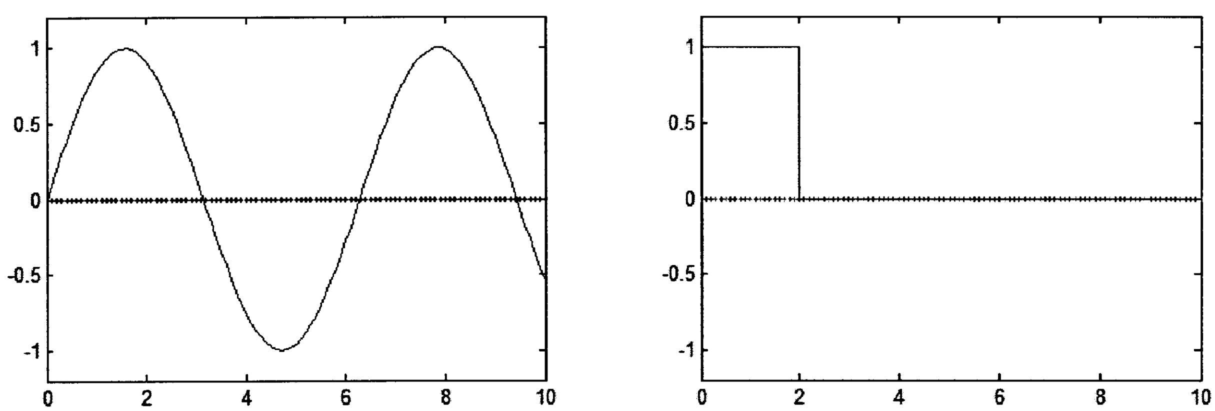

Figure 1: Graphs representing vibration (left) and shock (right).

1.3 Transmissibility and Isolation

When dealing with vibrations, it is important to know the difference between transmissibility and isolation. Transmissibility is associated with preventing the vibrations from a mass from spreading to the base, while isolation involves preventing external vibrations from affecting the mass. Figure 2 below shows this difference.

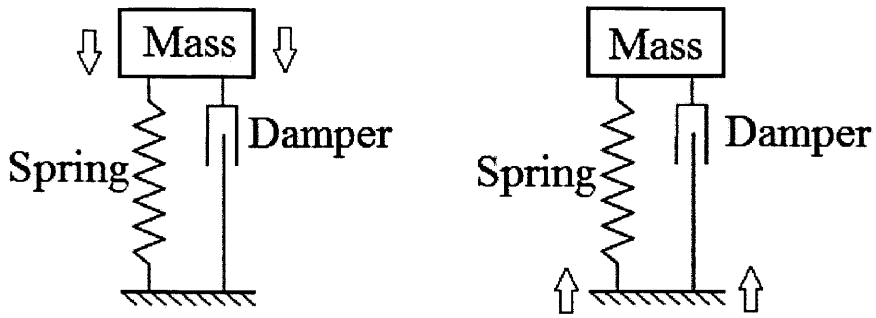

Figure 2: Picture depicting the difference between transmissibility (left) and isolation (right).

1.4 Overall Approach

For this thesis, different instruments and machinery that are relevant to this type of vibration on a ship are researched. A list of these different items is compiled, and the components most worthy of being noted are investigated. The system is modeled as a mass-spring-damper system, to which engineering mechanics equations are applied. Calculations are done to measure the degree of isolation that composite and traditional materials can provide. Using the data visualization properties of the computer program MATLAB, graphs are provided comparing the efficiency of isolation of composites to traditional materials. The main objective of an isolator system is to diminish the noise created as quickly as possible, which is the criterion of deciding the best material to use for the isolator.

2. Theoretical Analysis

2.1 Mass-Spring-Damper Model

In order to examine the behavior of the system, it is approximated with a mass-spring-damper model, depicted in figure 3 below. The mass represents whatever object would be mounted on the ship, either machinery or an instrument. The spring and damper represent either the composite material padding or the metal structural mount, also known as the isolation system. The base represents the floor, ceiling, or other wall of the ship.

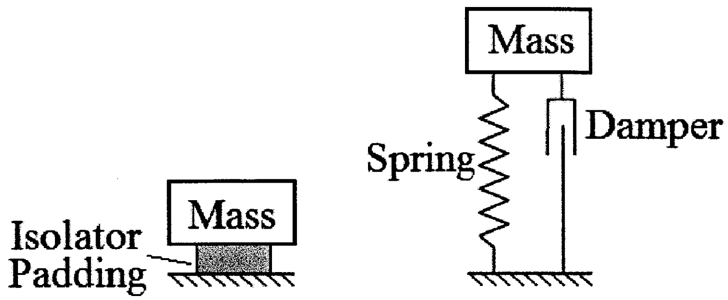

Figure 3: Picture showing the replacement of the isolator padding (left) with a theoretical spring and damper (right).

The causes of noise disturbances can be natural or artificial. Natural causes include waves, land, or even sea creatures. These natural disturbances are almost always a result of shock. Vibrations are usually caused by machinery within the ship itself. These types of disturbances are artificial.

2.2 Governing Equations

This section details the different equations input into MATLAB in order to produce the graphical results that will be discussed later in the paper. As previously stated, the isolation system will be modeled as mass-spring-damper system. The first equation comes from the study of Young's modulus, E, in engineering mechanics. Young's modulus is the comparison of tensile stress, σ, to tensile strain, ε. Tensile stress is represented as the force, F, applied to an object divided by the cross-sectional area, A_0, through which the force is applied. Tensile strain, on the other hand is represented as the change in length, ΔL, divided by the original length, L_0. Equation 1 below shows the aforementioned relationships.

Equation 1

$$E = \frac{stress}{strain} = \frac{\sigma}{\varepsilon} = \frac{F/A_0}{\Delta L/L_0}$$

Equation 1 can now be rearranged to define the force exerted on the object, shown in equation 2. This force is comparable to the restoring force of a spring, which is defined as the product of the spring constant, k, and its displacement, x. This relation is known as Hooke's law, shown in equation 3.By equating the two forces, and realizing that ΔL and x denote the same quantity, the spring constant equivalent for any material, given the Young's modulus, area, and length, can be found. Equation 4 defines this new spring constant equivalent, in which L is the thickness of the isolator padding.

Equation 2

$$F = \left(\frac{EA_0}{L_0}\right)\Delta L$$

Equation 3

$$F = -kx$$

Equation 4

$$k = \frac{EA}{L}$$

Another important aspect of this model is accurately modeling the effective mass, M_{eff}, which is defined for a spring as the mass of the object attached to the spring, M_{obj}, added to a fraction of the spring mass, which in this case is actually the mass of the isolator pad, M_{iso}. Although, there is some variability in the value of this fraction, a value of one-third is chosen in order to model the padding as a stiff spring. This relation is shown in equation 5 below.

Equation 5

$$M_{eff} = \frac{M_{iso}}{3} + M_{obj}$$

In order to properly model the motion of the system, there are still two more parameters needed, ω_n, the natural frequency, and c, the damping coefficient. The natural frequency is the same as the un-damped frequency, while the damping coefficient represents the energy dissipated. Both of these values depend on the spring constant and effective mass, as shown in equations 6 and 7. The damping coefficient also depends on ξ, the damping ratio, a scaling factor describing how the oscillations decay over time. The damping ratio was chosen arbitrarily at 0.3, a commonly used value.

Equation 6

$$\omega_n = \sqrt{\frac{k}{M_{eff}}}$$

Equation 7

$$c = 2\xi\sqrt{kM_{eff}}$$

With the aforementioned equations, the equation of motion, shown in two different versions in equation 8 below, is able to be completely defined for the system. For more information on how these equations were derived, please refer to the Williams and Hawes paper reported in the references section.

Equation 8

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n^2 x = m\ddot{x} + c\dot{x} + kx = 0$$

2.3 Computer Simulation

The computer application MATLAB is used to create a graphical interpretation of the data. Two different types of objects are used. The marine diesel engine, which creates vibrations itself, most commonly needs a system designed to increase transmissibility. The Electronic Chart Display and Information System (ECDIS) is a sensitive instrument that may require a system with increased isolation capability. These two objects and their parameters are listed below in table 1.

Table 1: Marine Diesel Engine and Electronic Chart Display Information System (ECDIS) parameters.

	Marine Diesel Engine	**Electronic Chart Display Information System (ECDIS)**
Type	Oscillating Machinery	Sensitive Instrument
Mass	2000 kg	200 kg
Area of bottom	3 m^2	2 m^2

Two different programs were made with MATLAB. Most objects need protection from shock, while not all of them will need protection from vibrations. The first program creates a graph of the position of an object subject to an impulse. The second program shows the transmissibility a measure of the efficiency of the isolator system.

For the time being, there are only two materials being used in the MATLAB programs to analyze the padding. These materials, steel and carbon fiber reinforced polymer (CFRP), are representative of composite and traditional materials. These materials and their properties are shown below in table 2.

Table 2: Properties of Steel and Carbon Fiber Reinforced Polymer (CFRP)

	Steel	**CFRP** (carbon fiber reinforced polymer)
Type	Metal	Composite
Density	7,800 kg/m^3	1,600 kg/m^3
Young's Modulus	210 GPa	150 GPa

3. Results and Discussion

3.1 Impulse

The impulse program represents how quickly motion is dissipated after a shock is applied. Figures 4 and 5 below show how the system reacts to the ECDIS and the marine diesel engine, respectively. Steel proves to be better suited for this scenario where shock is involved.

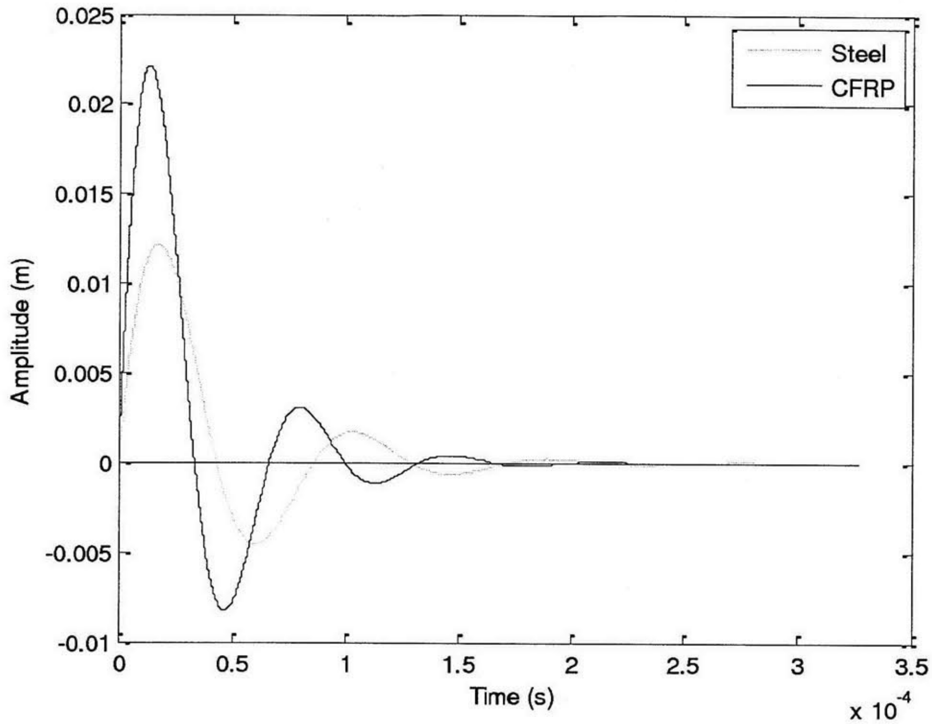

Figure 4: Graph of the amplitude of oscillations of the ECDIS (ship navigation system) as function of time for two possible isolator pads.

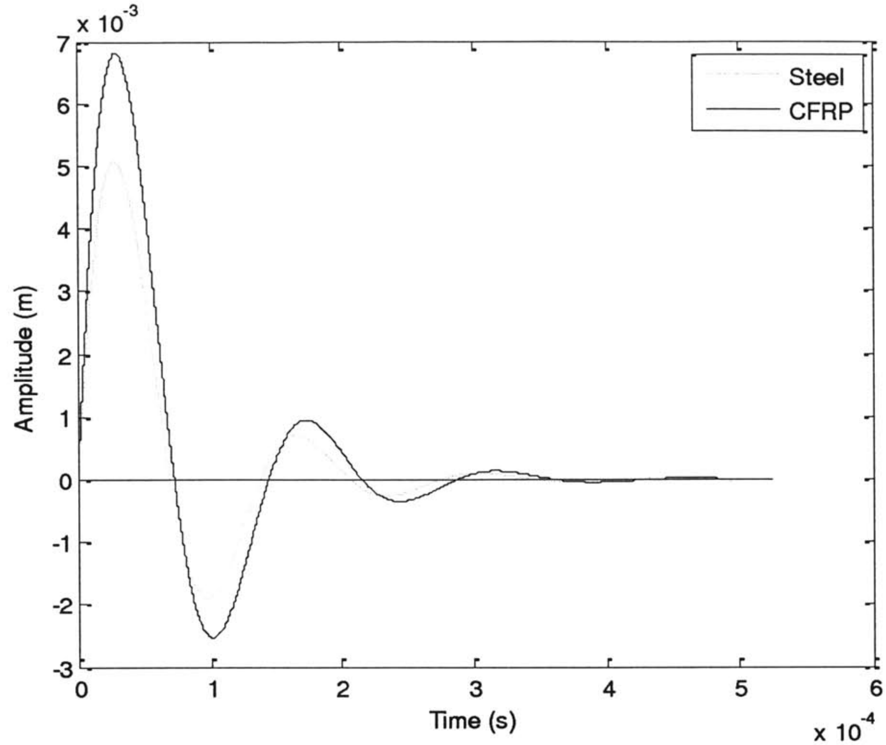

Figure 5: Graph of the amplitude of oscillations of the marine diesel engine as function of time for two possible isolator pads.

3.2 Transmissibility

The second program shows how well the isolator pad prevents the transmission of vibrations. Figures 6 and 7 below show how steel and CFRP would perform for both the ECDIS and marine diesel engine. Here the transmissibility is plotted as a function of a normalized frequency, which is the excitation frequency divided by the natural frequency of the system. The "better" material is the one whose peak comes first in the plot. Here, steel works better for the smaller mass of the ECDIS, while CFRP works better for larger masses like the marine diesel engine.

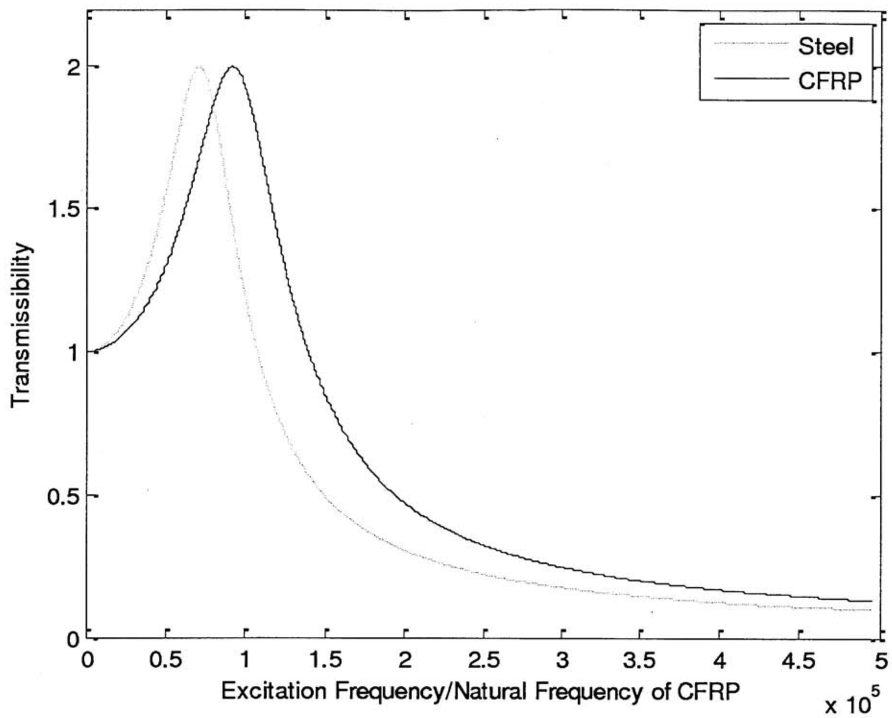

Figure 6: Graph of the transmissibility provided by each of the two possible isolator pads for the ECDIS (ship navigation system) as a function of the normalized frequency.

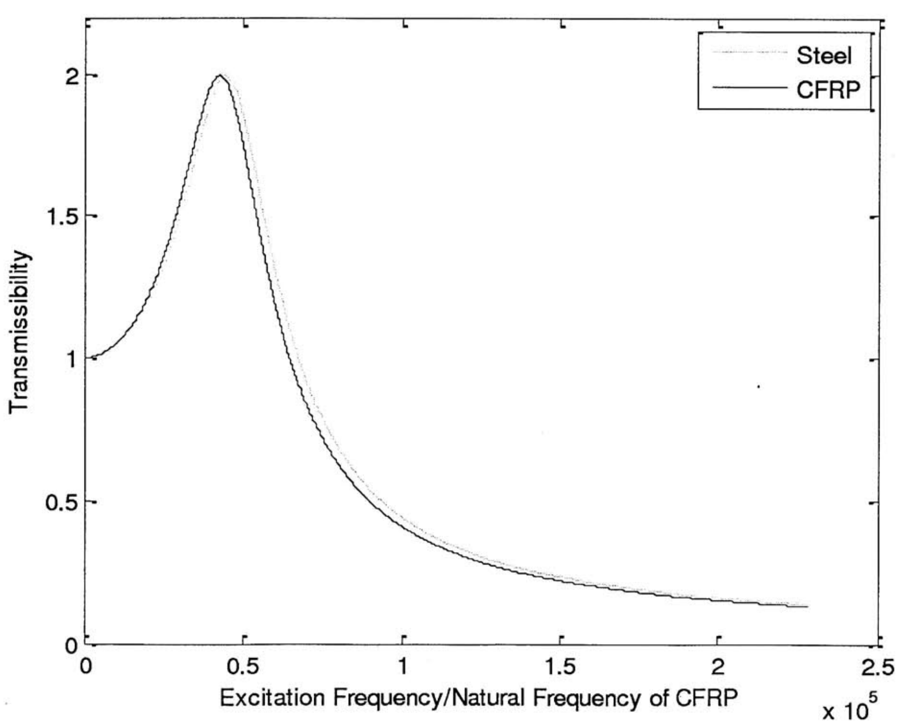

Figure 7: Graph of the transmissibility provided by each of the two possible isolator pads for the marine diesel engine as a function of the normalized frequency.

4. Conclusions

4.1 General Trends

As the parameters of the machinery or instrument change, the effectiveness of the vibration or shock isolation also changes. In the impulse program, increasing the mass and/or increasing the area of the object causes the gap between steel and CFRP to decrease. This means that CFRP becomes more efficient at reducing noise due to shock as mass and area increase. Still, steel is a better choice in the case of shock. In the transmissibility program, increasing the mass and/or decreasing the area causes CFRP to have a better efficiency, while decreasing the mass and/or increasing the area results in a better efficiency with steel. However, most vibrations are caused by heavy machinery, meaning a higher mass. This means that usually, CFRP works better for vibration isolation, while steel works better for shock isolation.

4.2 Future Recommendations

For future research into this topic, finding the effects of changing the area and mass of the object for the different types of disturbances would be very helpful in determining which material to use on a case-by-case basis. Also, creating an experimental test setup would be helpful in order to experimentally validate the model presented in this paper. Another possible area of interest is changing the model to account for a different fraction of the pad mass. Last but not least, one could add new materials as isolator padding.

References

"Composite Materials." *Nova - Science in the News*. Australian Academy of Science, Nov. 2000. Web. May 2011. <http://www.science.org.au/nova/059/059key.htm>.

"ECDIS Basics." *Fürstenberg*. 5 Feb. 2011. Web. Apr.-May 2011. <http://www.fuerstenberg-dhg.de/index.php?id=ecdis_startseite>.

Krodkiewski, J. M. "Mechanical Vibration." *Mechanical Engineering: Melbourne School of Engineering*. University of Melbourne, Australia, 2008. Web. Mar.-May 2011. <http://www.mech.unimelb.edu.au/dynamics/14lec.pdf>.

Rodríguez, Eduardo E., and Gabriel A. Gesnouin. "Effective Mass of an Oscillating Spring." *The Physics Teacher* 45 (2007). Print.

"Technical Section: Vibration and Shock Isolation." *Anti Vibration Mounts*. Advanced Antivibration Components. Web. Feb.-May 2011. <http://www.vibrationmounts.com/V100/V100Cat.htm>.

"Vibration Isolation for Precision Equipment." *Herzan - Vibration and Acoustic Isolation*. Web. Mar.-May 2011. <http://www.herzan.com/herz1.htm>.

Williams, James H., Jr. and David Harry Hawes, "Vibration and Shock Isolation: Performance of Different Isolator Pads with Focus on Polymeric Composites," Department of Mechanical Engineering, MIT, June 2011.

Williams, James H., Jr. *Fundamentals of Applied Dynamics*. New York: J. Wiley & Sons, 1996. Print.