

Carbon Fiber Substitution in Shipboard Instrument Structural Mounts for Vibration and Shock Isolation

by

Anna C. Haas

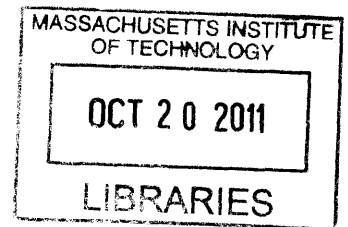
Submitted to the Department of Mechanical Engineering in Partial Fulfillment of the
Requirements for the Degree of Bachelor of Science in Mechanical Engineering at the

Massachusetts Institute of Technology

June 2011

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ABSTRACT

Insufficient support and isolation of rotating and non-rotating shipboard equipment and instruments may cause damage to the components or the ship itself. Reduction of noise emissions to the ship's hull and protection of equipment from outside disturbances may be attained through isolator mounting systems. The objective of this thesis is to investigate and analyze the effects of substituting carbon fiber reinforced polymer (CFRP) for use in the structural mounts on rotating marine diesel engines and on sensitive, non-rotating Electronic Chart Display and Information Systems (ECDIS). The high specific stiffness of CFRP make it an interesting candidate for mounting material substitution. The shipboard components were modeled with mounts made of steel, rubber and CFRP and their step response, impulse response and transmissibility behavior were analyzed. It was concluded that the choice in appropriate mounting materials depends on the application, but CFRP was found in many cases to display similar vibration isolation behavior to steel while being much less dense.

Thesis Advisor: James H. Williams, Jr.

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Acknowledgement

This thesis would not have been possible without the help of several individuals. First and foremost, Professor J.H. Williams, Jr. provided incomparable knowledge, guidance and patience as my thesis advisor. I am thankful to have had the opportunity to learn from him. Secondly, I am lucky to have wonderfully supportive parents, whose love and encouragement helped me greatly throughout my undergraduate experience. Thirdly, I am appreciative of the time and effort David Hawes put into developing related concepts in his project with Professor Williams. Finally, I am 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).

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Introduction

The structural mounting systems for shipboard equipment are an important part of a ship's design. A well-designed mounting system will function as a damper and reduce vibrations from the machinery and shock from external disturbances.

Many ships frequently undergo sudden, impulsive forces or shocks that result in a sudden change in velocity. These shocks could be caused by rough seas, a collision, or an explosion elsewhere aboard the ship. In order for the shipboard equipment to remain operational in the case of large shocks, a combination of robust mechanical equipment design and a damping mounting system may be incorporated to isolate the un-wanted vibrations and keep their effects under control. The concept in this situation is called isolation and the goal of the mounting system is to reduce the motion transmitted from the hull of the ship to the instrument and therefore keep the displacement at a minimum [1].

Another function of the mounting systems of shipboard machinery is to reduce the noise emitted from the ship into the surrounding water. Such vibrations are caused by machinery with rotating parts, such as pumps and engines and may be traced by enemy radar or missiles. The concept in this situation is called transmissibility and the goal of the mounting system is to reduce the force transmitted from the rotating component to the hull of the ship.

Commercial ships and military-associated ships have different needs with respect to the mounting structures of their machinery. Both types need protection via isolation of their sensitive instruments from external disturbances, but military ships typically need to be capable of withstanding larger impulses. Transmissibility can be a security threat for military ships, because it makes them more easily detected by enemy ships. In addition, on any type of ship vibrations

caused by insufficient damping of rotating components can cause discomfort for those on board and be undesirable for general ship maintenance. According to military specifications, most shipboard installations on naval ships must be tested on their ability to withstand shock loadings which may be incurred during wartime service [2].

This thesis will investigate the substitution of carbon fiber for materials used in current mounting systems including steel and rubber and how the material properties of each affect transmissibility and isolation of vibrations. Table 1, below, shows the fixed values of the Young's moduli and densities used in this thesis for comparative purposes.

Table 1: Material Properties of Mounting Materials

	Young's Modulus (GPa)	Density $\left(\frac{kg}{m^3}\right)$
Carbon Fiber Reinforced Polymer	150	1600
Steel	210	7800
Rubber	0.1	1100

Note that the Young's modulus of rubber assumes small strain and that the Young's modulus of CFRP may vary greatly with the orientation of the fibers. CFRP was chosen for this study because it is rarely used as an isolator pad and its high strength to density ratio make it an interesting candidate.

The two shipboard machinery components modeled in this thesis were chosen such that both rotating and non-rotating components were represented as well as large and small masses. Their masses, areas and frequencies are shown in Table 2, on the next page.

Table 2: Component Characteristics

	Marine Diesel Engine	Electronic Chart Display and Information System (ECDIS)
Mass (kg)	2000	200
Area in contact with mount (m^2)	3	2
Rotating Frequency (rpm)	2000	N/A

Marine diesel engines have almost completely dominated the propulsion market for merchant ships. They are an extremely critical component on any ship and each has an operating frequency determined by its stroke cycle. For this thesis, mass, area and frequency parameters fitting of a mid-size diesel engine were chosen [3].

ECDIS is an informational system which must be approved by the International Maritime Organization and is present on most modern ships that is capable of automatic functions such as route planning, route monitoring and route documentation. It may also include safety functions such as tracking tide and current information and collision avoidance [4]. Like any intricate instrument, the ECDIS contains fragile parts which benefit from an isolating mounting system. ECDIS's can be tabletop or floor mounted. For this thesis, mass and area parameters fitting of a fairly large, freestanding ECDIS were chosen.

Analytical Approach

The shipboard components and their mounting systems can be modeled as a simple mass, spring and dashpot system shown below in Figure 1, where m is the mass of the component, k is the stiffness of the mounting material and c is the damping coefficient of the system.

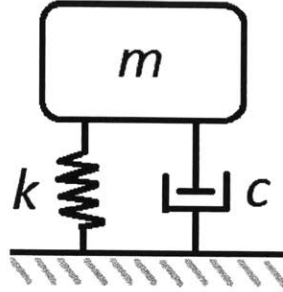


Figure 1. Mass-spring-dashpot model used to represent the component and mounting pad system

The mass of the spring, which in this case is the structural mount, can be accounted for using an energy approach to solve for the effective mass of the system, m_{eff} . To simplify the process, it can be assumed that the mount has an evenly distributed mass and a discrete stiffness [5]. After integrating over the length of the spring, the kinetic energy of the system, T , can be written as

$$T = \frac{1}{2} \left(m + \frac{\rho AL}{3} \right) \dot{x}^2 \quad (1)$$

where ρ is the density of the mounting material, A is the area of contact between the component and the mounting pad, L is the thickness of the mounting pad and \dot{x} is the velocity of the component. It can then be concluded that the effective mass is given by

$$m_{eff} = m + \frac{\rho AL}{3} \quad (2)$$

The equation of motion of this system is derived to be

$$y = 1000000 * \left(\frac{1}{\sqrt{km_{eff} - \frac{c^2}{4}}} \right) * e^{\frac{-ct}{2m_{eff}}} * \sin \left(\sqrt{\frac{k}{m_{eff}} - \frac{c^2}{4m_{eff}^2}} * t \right) \quad (3)$$

where y is the vertical displacement of the component. The derivation of this equation is shown in detail in “Vibration and Shock Isolation: Performance of Different Isolator Pads with Focus on Polymeric Composites,” a paper by James H. Williams, Jr. and David Harry Hawes [6].

Before moving forward with analysis of the system, several assumptions must be made. First, the damping ratio ζ , of which c is a function must be decided upon. The damping ratio characterizes the frequency response of the system. It is not a material property so a reasonable value of $\zeta=0.3$ may be chosen and kept constant for all cases. A thickness, L , for the isolator pad, which affects the mass of the system, must also be chosen. Again a reasonable value of $L = 0.1$ m will be used for each case.

An important intermediate value used to create the plots of interest is the system’s natural frequency, ω_0 which is defined as

$$\omega_0 = \sqrt{\frac{k}{m_{eff}}} \quad (4)$$

A system’s natural frequency is its frequency of unforced vibrations.

Settling time is one of the parameters of interest for the systems and is defined as the amount of time it takes for the response to reach, and stay within, 2% of its final value. It can be calculated as

$$T_s = \frac{\ln(0.02)}{\zeta\omega_0} \quad (5)$$

As mentioned previously, the two concepts of transmissibility and isolation are important in the analysis of the structural mounts. Transmissibility is applicable when the component has

vibratory motion that could be transmitted to the base and is defined as the ratio of the maximum transmitted force to the maximum excitation force. Isolation is applicable when the vibrations of the base are being transmitted to the component and is defined as the ratio of the maximum transmitted displacement to the maximum excitation displacement.

Results

Using the derived equations of motion and the component and material parameters given in Tables 1 and 2, the behavior of the systems under various conditions can be graphed and analyzed using MATLAB software.

Impulse Response

Figures 2 through 5 show the systems' impulse responses. An impulse in this case is defined as an infinitely large, but infinitesimally brief spike in the input function. A key parameter of interest is the maximum vertical displacement.

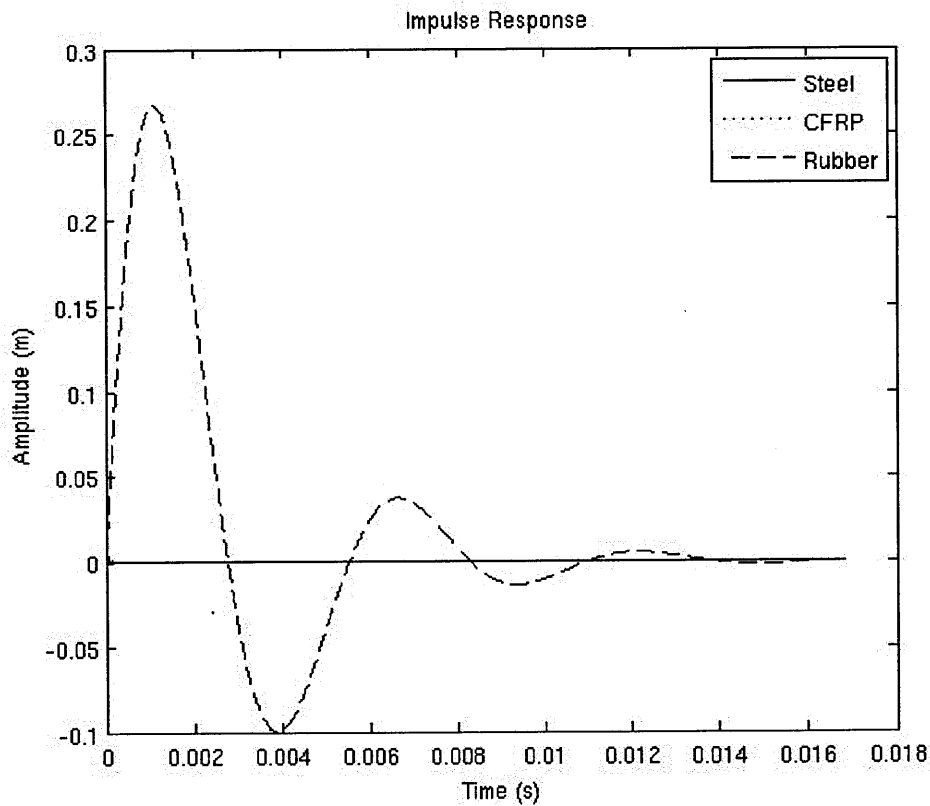


Figure 2. Vertical displacement of the engine as a function of time in response to an impulse. Only rubber is visible on this scale

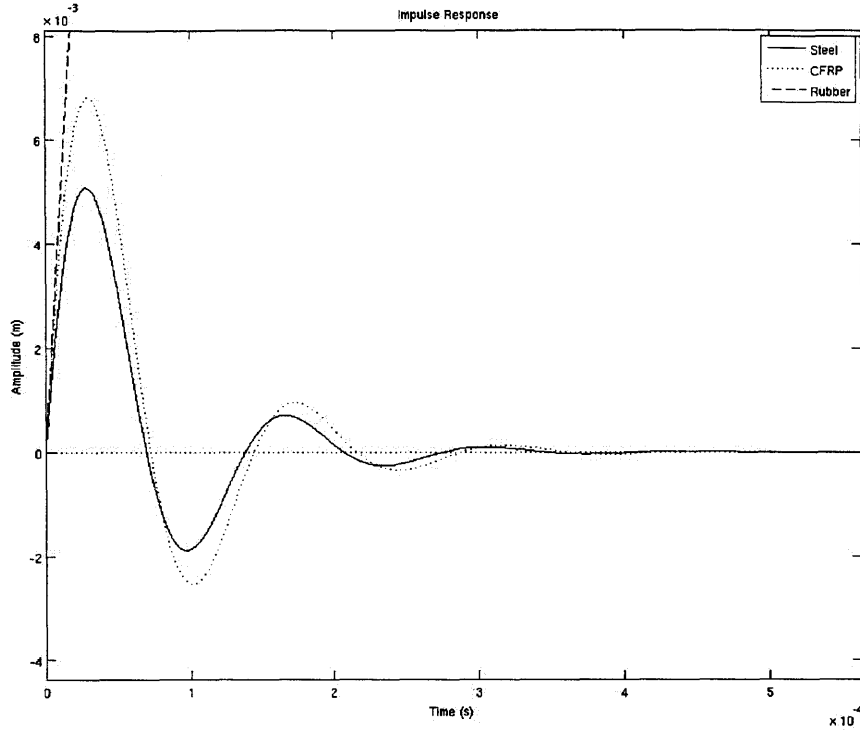


Figure 3. Close-up version of vertical displacement of engine as a function of time. CFRP and steel are now visible

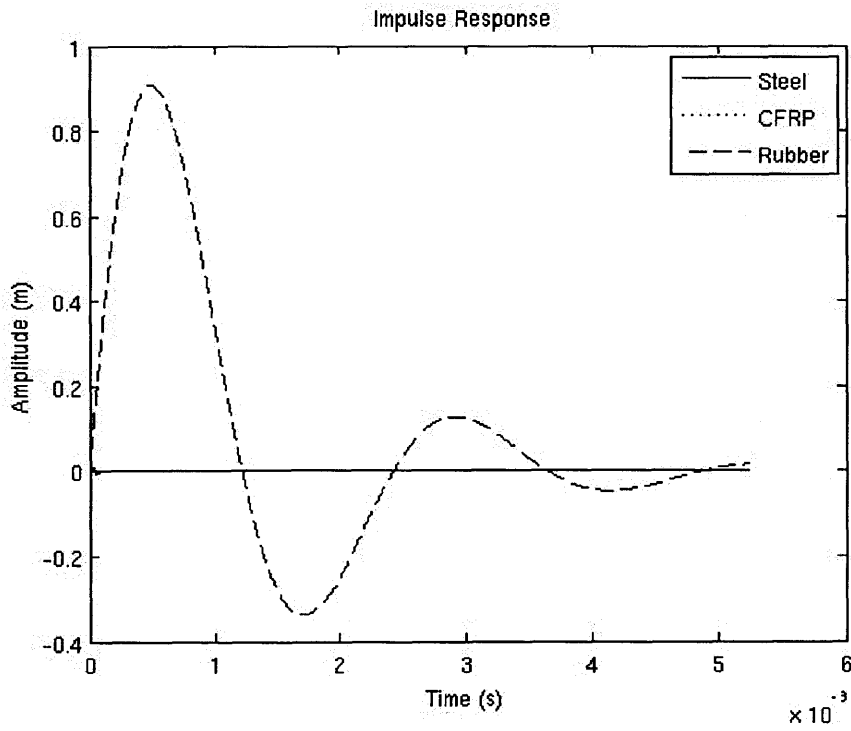


Figure 4. Vertical displacement of ECDIS as a function of time in response to an impulse. Only rubber is visible on this scale

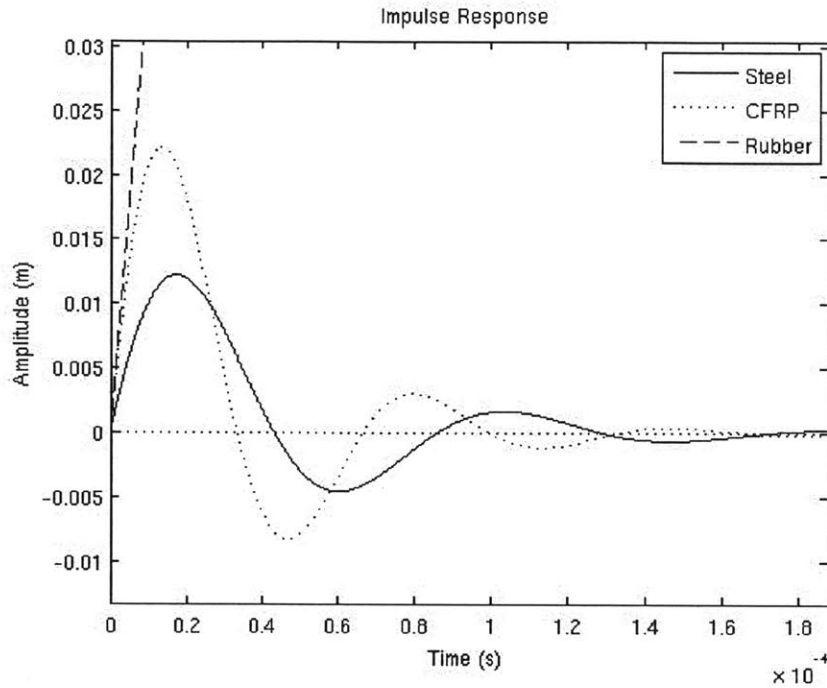


Figure 5. Close-up version of vertical displacement of ECDIS as a function of time in response to an impulse. Steel and CFRP now visible.

Table 3 shows the maximum vertical displacement extracted from the impulse response graphs.

Table 3: Maximum Displacement of Impulse Response

	Maximum Displacement (m)
Engine with Rubber Mount	0.267
Engine with CFRP Mount	6.81e-3
Engine with Steel Mount	5.07e-3
ECDIS with Rubber Mount	0.908
ECDIS with CFRP Mount	0.0221
ECDIS with Steel Mount	0.0122

Step Response

Figures 6 through 9 show the systems' response to a step input with a step size of 0.005 meters. The data for these graphs has been non-dimensionalized and normalized about the settling time for CFRP. A key parameter of interest is the settling time, as defined in Equation (5).

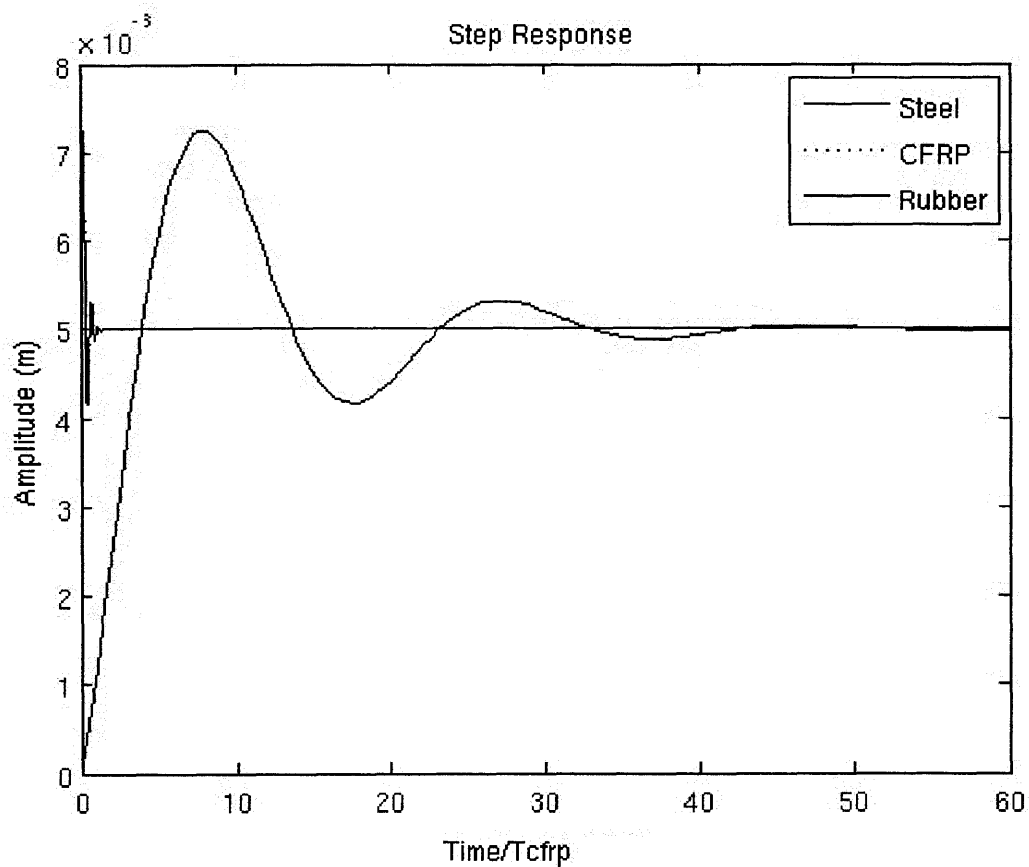


Figure 6. Step response of the engine. Rubber's data is the most visible.

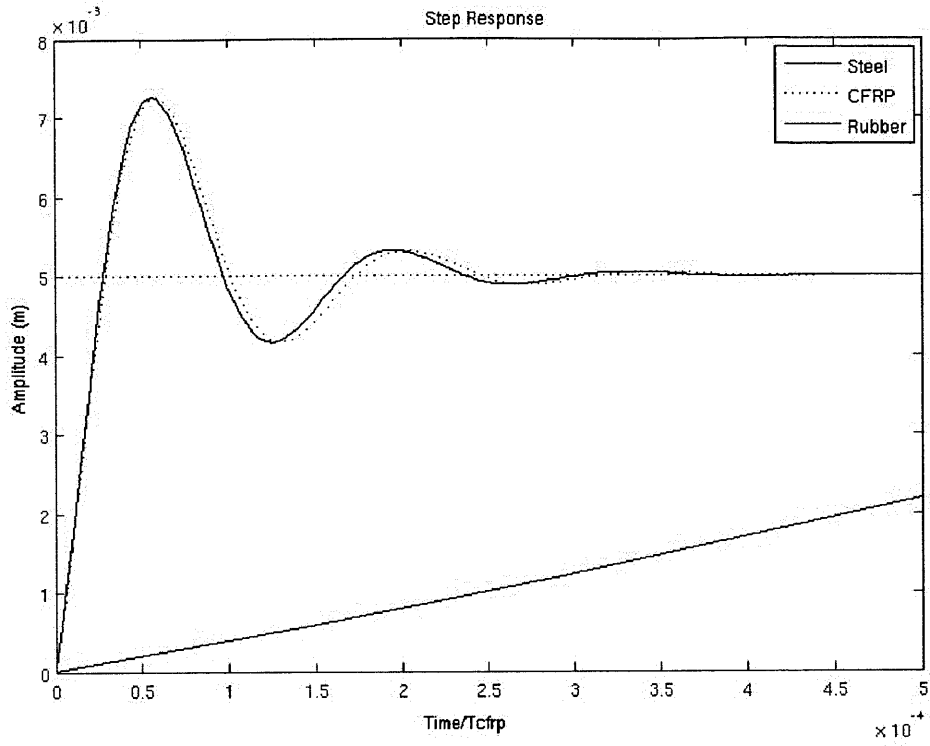


Figure 7. Close-up of step response of the engine with CFRP and steel mounts.

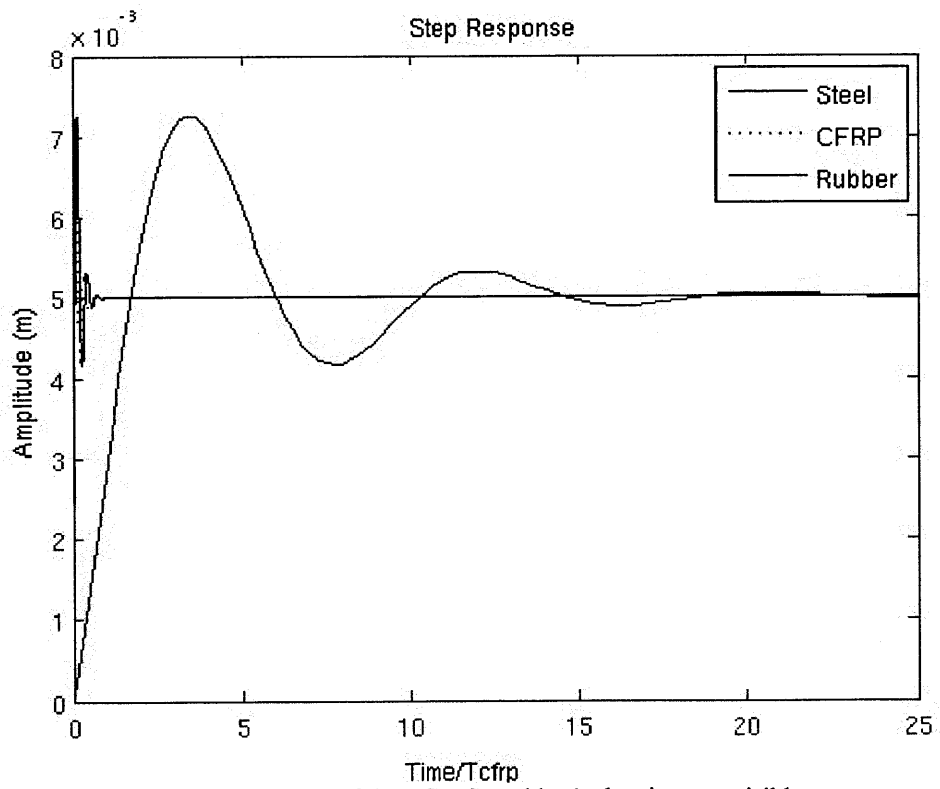


Figure 8. Step response of the ECDIS. Rubber's data is most visible.

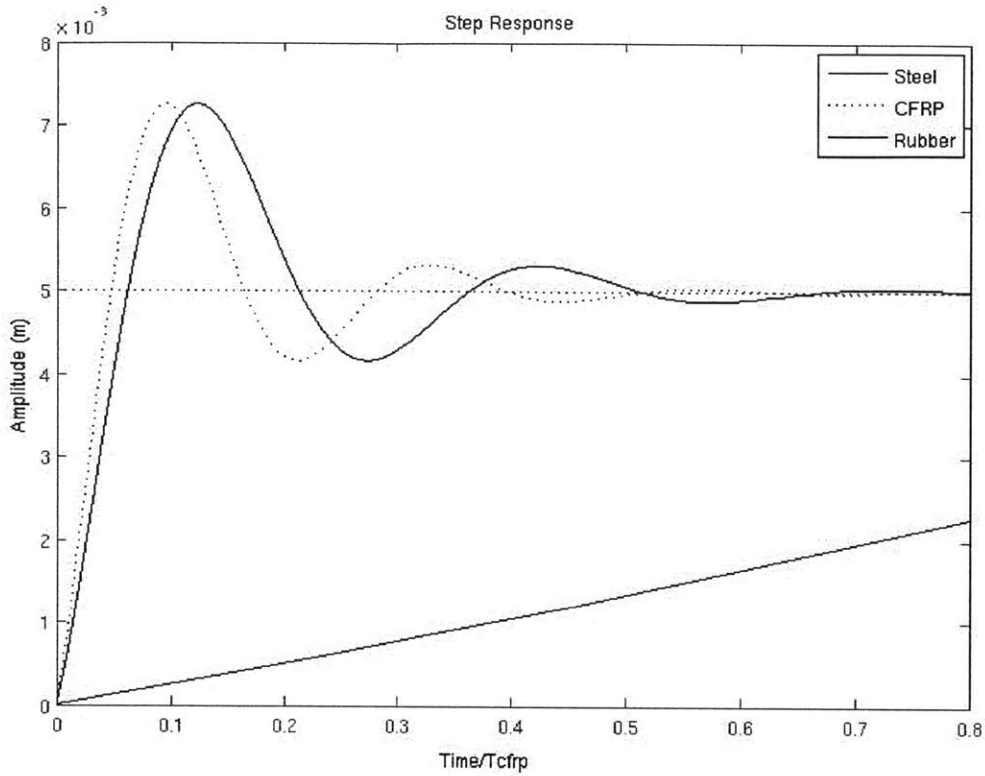


Figure 9. Close up of step response of ECDIS with CFRP and steel mounts

Table 4 shows the settling time of the components with different mounting materials, calculated using equation (5).

Table 4: Settling Time

	Settling Time (s)
Engine with Rubber Mount	0.011
Engine with CFRP Mount	2.85e-4
Engine with Steel Mount	2.73e-4
ECDIS with Rubber Mount	4.80e-3
ECDIS with CFRP Mount	1.31e-4
ECDIS with Steel Mount	1.70e-4

Transmissibility

Figures 10 and 11 show the plots of transmissibility as a function of the ratio of excitation frequency to natural frequency for the case of the diesel engine.

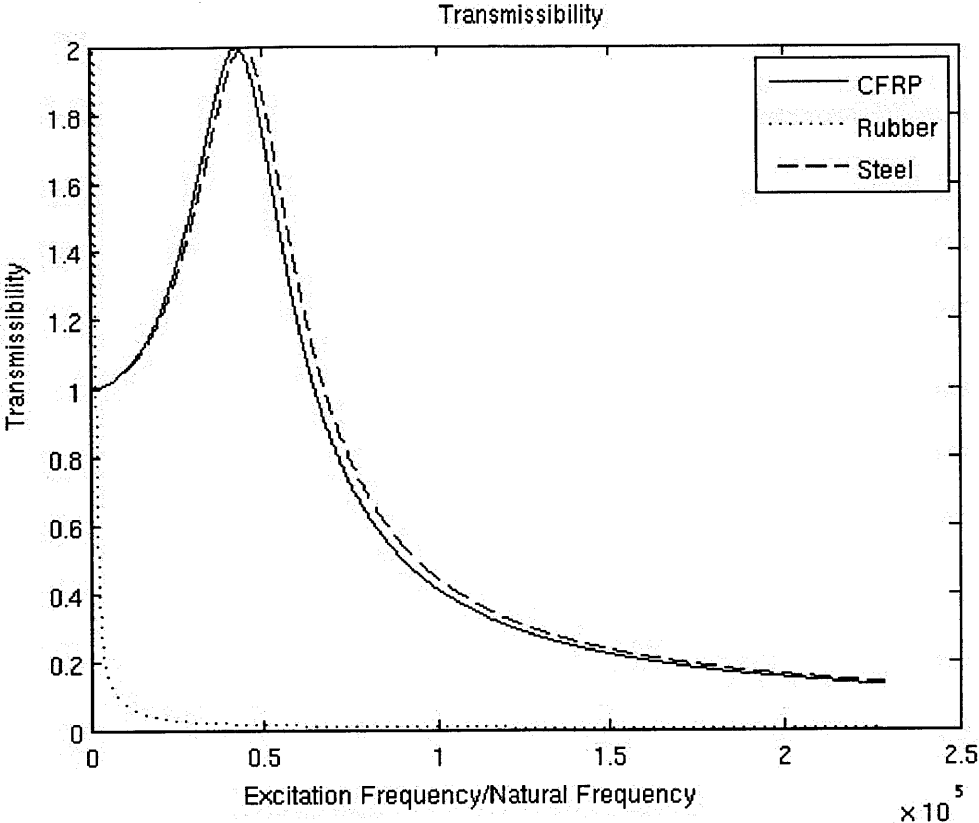


Figure 10. Transmissibility of engine vibrations through different mounting materials as a function of frequency.

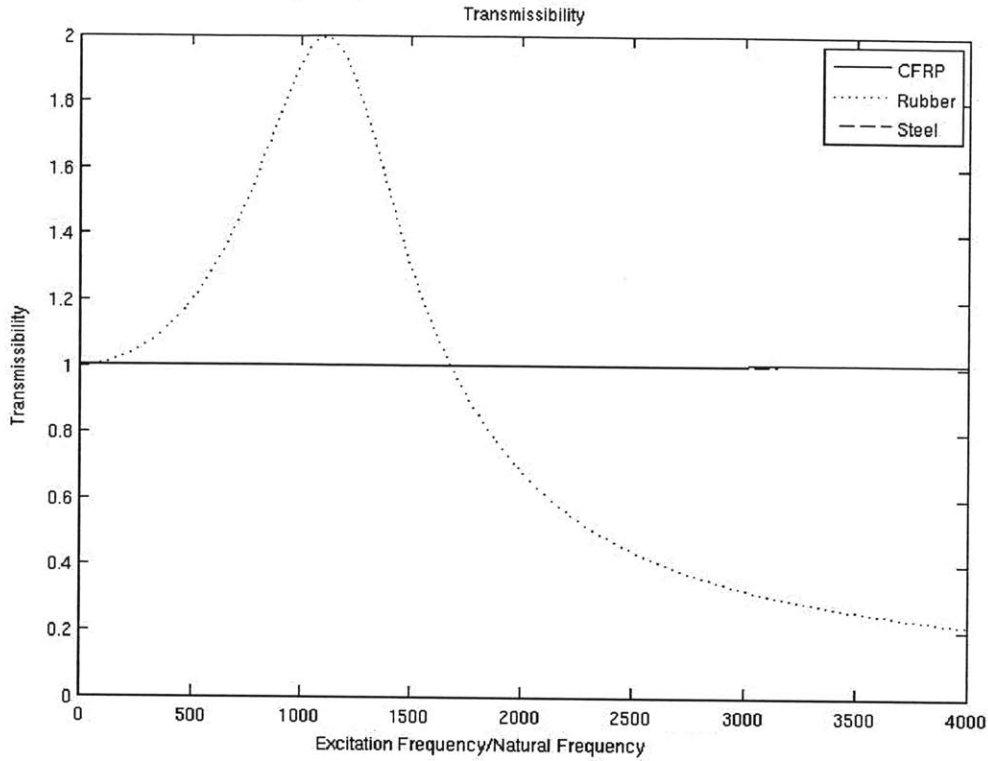


Figure 11. Close-up of transmissibility of engine vibrations through a rubber mounting pad as a function of frequency.

Table 5 shows the excitation frequencies that correspond to resonance in transmissibility. They are calculated using the peak ratio extracted from the data and the known natural frequencies of the engine and mounting pad systems.

Table 5: Resonant Excitation Frequencies

	[Hz]	[rpm]
Engine with Rubber Mount	4.4e6	2.6e8
Engine with CFRP Mount	1.9e9	1.2e11
Engine with Steel Mount	2.1e9	1.3e11

The peak amplitude of transmissibility is not relevant since it depends on the damping ratio of the system which was chosen arbitrarily and set equal for all mounts.

Discussion and Conclusions

The choice of mounting material can affect the behavior of a shipboard mounting system in different ways depending on the component being mounted and the types of vibrations to be encountered.

Maximum Amplitude of Displacement

When deciding on a mounting system, maximum amplitude of displacement is an important factor to be considered. A large displacement can interfere with a sensitive instrument's readings and/or transmissions. Displacements can also be very damaging to the shipboard components themselves. Even large, seemingly insensitive machinery can malfunction if parts become misaligned, which can happen when small displacements are propagated. Figures 2 through 5 show the impulse response of both the engine and the ECDIS with each material used for mounting. For both components considered in this thesis, the impulse response of an object with a steel mount results in a smaller maximum displacement than that of an object with a CFRP mount. When using rubber, however, the maximum displacement is several orders of magnitude larger than either steel or CFRP. These differences are due largely to the low Young's modulus of rubber. The light ECDIS system has maximum amplitudes of several times greater than the corresponding cases of the much heavier engine. The exact values recorded in Table 3 are only relevant if our assumed value of the damping ratio, ζ , is accurate. Therefore, their relative magnitudes are of more importance.

Settling Time

Another consideration in the design of mounting systems is the settling time of an object in response to a step input. Since it is not a realistic goal to achieve complete isolation from external forces, it is important for many instruments to recover from disturbances at a rapid rate. This can be accomplished by having a mounting system with a short settling time. In the case of the diesel engine, as shown in Figures 6 through 9 and Table 4, a steel mounting system results in a higher system natural frequency than CFRP and therefore a shorter settling time. This may be attributed to the large stiffness of steel.

In the case of the ECDIS, a CFRP mounting system results in a higher natural frequency and the shortest settling time. This can be attributed to the lower density of CFRP, which is more influential in the ECDIS case because of its smaller mass.

For both the engine and the ECDIS cases, a rubber mounting system resulted in a much longer settling time than the other materials.

Aside from the discussed material properties, other factors that can reduce the settling time of the component include decreasing the thickness of the mounting pad, decreasing the mass of the component, increasing the surface area between the component and the mounting pad and increasing the damping ratio.

Velocity and Acceleration

Depending on the shipboard component, it may be important to control the velocity and/or acceleration of the object in addition to its displacement. For the cases studied in this thesis, steel and CFRP mounting systems resulted in systems that oscillated with similar speeds and accelerations. Components with rubber mounts on the other hand oscillate much more slowly and with less acceleration. This can be a considerable advantage in components that contain extremely fragile parts which are less likely to break at low accelerations.

Transmissibility

For rotating or oscillating components, transmissibility must be taken into account in the system mounting design. As the figures in the results section show, rubber isolates well at lower excitation frequencies. Neither CFRP nor steel isolate well at low frequencies, with the resonant frequency of CFRP slightly lower than that of steel. For the marine diesel engine, the operation frequency of 2000 rpm is low enough that any of the three discussed materials would neither propagate nor isolate vibrations by a significant amount. Increasing the thickness of the mounting pad will shift the frequencies to the right, resulting in better isolation at lower frequencies. The operating frequency of the shipboard component in question should be considered when choosing a mounting system design, to ensure that it does not approach the resonance frequency of the mounting material and incur resonant disaster.

Other Considerations

In addition to the vibration isolation and transmissibility behavior, density of the material may be taken into account in the design of mounting systems. A low density material may be desirable for vessels on which weight is a concern. CFRP has the highest stiffness to density ratio of the discussed materials. Another possibly important material property is the electrical conductivity of the mounting pad. Steel is the most conductive, followed by CFRP and then rubber. A non-conductive material such as rubber may be appealing for safety reasons. Thirdly, the designer may look at the thermal conductivity of the materials. Again, rubber is the least conductive. Finally cost will realistically always factor in to the choice in material and CFRP is currently the most expensive followed by steel and rubber, but this gap is subject to change with time as well as the quality of the materials considered.

Recommendations

There is no one material that emerges as the clear best performer for the systems in this thesis. The choice in mounting material depends greatly on the application and constraints of the design. When CFRP is compared with the currently used mounting materials, it achieves very similar maximum displacements, settling times and resonant frequencies to those of steel. Since CFRP also has a density which is closer to that of rubber, it provides an interesting middle ground for mounting material design.

In further research, it may be interesting to analyze the effects of changing the shape and/or area of the mounting pad. Furthermore, since materials with low stiffness tend to isolate better at lower frequencies, it may be worth investigating changing the orientation of the fibers in CFRP, such that their Young's modulus decreases which in turn decreases the stiffness without changing the material density.

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