EFFECT OF VIBRATION DAMPING COATINGS ON VIBRATION PROPERTIES IN LOW FREQUENCY

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This paper is an attempt made to clearly identify and define the effect of vibration damping coatings and its importance aboard naval platforms towards stealth and deterioration of hull. This study is aimed to experimentally determine the vibration damping capacity of various coating materials and their effect on vibrations. As a preliminary investigation, Finite Element and analytical modelling has been carried out to determine the length of specimen, so that its corresponding resonant frequencies fall within the required frequency range. Abaqus/CAE 6.14-1 is used for Finite element modelling of the beam specimen required for the experiments. Mild Steel (MS) of cross-section 24mm x 03mm is used as base metal for test specimens, on which different combination of composite layers are applied. An epoxy based High Damping Material (HDM), Neoprene rubber, Random Glass Fibre Mat with Epoxy (Araldit CY 230-1), are used individually and in different combinations, to make 13 test specimens. The test specimens are tested in a cantilevered boundary condition. Response of each test specimen is obtained using a Laser vibrometer. Free decay method is used to conduct experiment and damping ratio is calculated at each mode using log decrement method. Results of the damping test for each of the HDM coated, composite coated and Glass Fibre mat coated test specimen exhibits a marked increase in damping capacity from that of uncoated test specimen. This study has enabled us to find out the effect of different coating materials and layer thicknesses on the overall damping capacity of a system.

Key words: -

HDM; DAQ; FFT; Damping; Coatings;
INTRODUCTION

Overview

A naval ship is specifically designed for military purposes. It is different from civilian or merchant ships in its role, design and construction. Naval ships are more stable, manoeuvrable, damage resilient and are fitted with various radar and weapon systems. Depending upon its role, design and construction, naval ships are further categorised as warships and support ships. Naval ships specifically designed and constructed for warfare are called warships including aircraft carrier, destroyers, frigates etc, whereas ships meant for auxiliary role are called support ships which includes tankers, hospital ships etc..

A ship whilst sailing generates various disturbances (by virtue of its structure, speed, running machinery etc) like wake, noise, vibration, heat, electro-magnetic flux etc and because of this, presence of a ship can be detected by using appropriate sensors. These disturbances generated by a ship are known as signature, as each of these is unique for every ship and can be used to identify ship or class of ship.

Figure 1: Naval ship and its various parts [1]

It is known that a database/library of abovementioned signatures of all naval ships is generally maintained by navies across the world, to identify and target enemy ships in case of conflict. Besides this, various weapons like missiles, torpedoes, sea mines etc. Using different sensors and homing-in techniques, use these signatures to detect, classify, localize and destruct target naval ships. To avoid these threats, it is very important to cease or minimise these signatures from naval ship. An attempt to minimise these signatures is a step towards stealth.

Motivation

Stealth can also be defined as the ability to conceal or avoid detection. Stealth is not something new to naval warfare. For centuries man has used the vast area of the ocean to hide from the enemy. Submariners have long relied on stealth to avoid detection, to hide from enemy attack and to reach the optimum location to conduct a surprise attack on enemy ships or targets. There are various signatures generated by ship, which hinders her stealth capability.

One of the most important signatures generated by a naval ship is underwater noise. This signature helps enemy naval ships, submarines and torpedoes to search, locate, identify and destroy the target ships. Underwater signature comprises of noise from running machinery and cavitation noise from ships’ propeller. The cavitation noise generated by propellers is only dominant during critical speed at which cavitation occurs, whereas noise generated by machinery running aboard, dominates the underwater signature spectra of the ship for all the regimes before cavitation occurs.

To achieve the optimal level of underwater stealth for a naval ship, it is crucial to cease or minimise the vibrations generated by running machinery, which in turn gets transmitted to hull structure and
causes underwater noise. Presently, various methods including vibration isolators/dampers, active tuned dampers etc are being used to reduce the vibration transmission through machinery foundation.

As mentioned above the underwater noise generated by the machinery is major concern, as same is dominant over a broad range of exploitation regime. Since this also gives away the information about machinery fitted aboard like propulsion engines, power generation plant etc. and leads to the identification of the ship or class of ship. Therefore, it is required to limit the underwater noise to a desired level by redesigning that particular source or by providing some active or passive damping system.

Objective
Since a very large surface area is available at the machinery foundation, the idea is to put a vibration dampening coating/layers to reduce the vibrations. As various paints and enamel coatings are generally applied to every structural part and equipment fitted on-board naval ship. These coatings provide a protection against rust, helps in colour coding of different systems and also provide an pleasing appearance. However, some materials have unique capacity to absorb vibration and noise, and dissipate energy over a wide range of frequencies. These materials with high damping ratio and loss factor can be attached or applied in form of coating (single layer or multilayer, thin or thick) to the concerned structure, to achieve vibration damping. A damping treatment could consists of any material (or combination of materials) applied to a component to increase its ability to dissipate mechanical energy. Although all materials exhibit a certain amount of damping, many (steel, aluminium, magnesium and glass) have so little internal damping that their resonant behaviour makes them effective sound radiators. By bringing structures of these materials into intimate contact with a highly damped, dynamically stiff material, it is possible to control these resonances.

STEALTH AND SIGNATURE SUPPRESSION
Stealth can be defined as the ability to conceal and avoid detection. In case of naval warship, stealth is about reducing the various signatures generated by the different aspects of the ship, like shape, dimensions, heat, wake, noise etc. The various signatures that are to be considered whilst dealing with the stealth of naval warship are

- a) Shape/dimensions of ship or radar cross section (RCS)
- b) Infrared (IR)
- c) Magnetic
- d) Electric
- e) Other signatures such as hydrodynamic wake, contaminants, bioluminescence etc.
- f) Underwater noise

Radar cross section (RCS)
The radar cross section of an object is defined as "a measure of the power reflected in a specific direction" [2] and is normally expressed in square meters or logarithmically in decibels per square meter (dbsm). While an entire ship will reflect radar energy as a whole, individual part of the superstructure and smaller objects such as gunmounts, radar
antennas, lifeline stanchions and deck lockers will also reflect energy separately due to each object's shape, size and orientation to the direction of the incoming radar energy. Most superstructures (and the hull form) have been constructed with large, flat, vertical surfaces and include many dihedrals and trihedrals intersecting at 90 degrees. Topside configurations include numerous cylindrical kingposts, stanchions and antennas. These shapes, vertical plates, planes joining at 90 degrees, and cylindrical objects all intensify an already large RCS. The two principal ways of reducing a warship's RCS are the application of radar-absorbent material (RAM) to the most reflective parts of the ship, and the use of computer-aided design (CAD) programs to optimize the shape of the hull and superstructure. With shaping, the goal is to eliminate sharp corners and vertical surfaces and to cause the radar energy to be scattered away from the enemy rather than be reflected back in the specific direction of the enemy radar receiver [3].

**Infrared (IR) signature**

A ship has broadly two signatures, one in the spectral band 3-5 µm and the other in the spectral band 8-14 µm. The former is used in missiles for homing on to the ship and the latter is used for identifying the ship. The 3-5 µm band signatures is provided by the prime movers of propulsion, generation plants and the exhaust gases of the diesel engines used in warship. These exhaust gases are at temperatures of the order of 200-300°c. The gas turbines that are being used in warships have higher exhaust temperatures, of the order of 500°c. In general, the dominant sources of radiation in the 3-5 µm bands are the exhaust gases, plume and the funnel which carries the exhaust gases. The low temperature parts of the ship which are nearly at the ambient temperature provide the signature in the 8-14 µm. The low temperature parts are primarily the hull and the superstructure. A ship is detected against sea background or sea and sky together. The contrast between the background and the ship enables the detector to detect the ship. The background signature consists of emission from sea and sky and reflected radiation from other sources e.g. Reflection of solar radiation from the sea surface or clouds, or scattering by atmosphere. The contrast that is provided between the background and the ship depends on the atmospheric effects. Anti-ship missiles equipped with IR detecting systems can search, track and identify warships. IR guidance systems fitted in missiles can home on to the target passively. The ship under attack does not receive any advance information [3].

**Magnetic signature**

Reduction of magnetic signature is accomplished either by magnetically treating the ship in special facilities - Deperming - or by fitting the ship with special active degaussing coils through which electric current is passed. The strength of the electric current is controlled from within the ship in such a way that at any time and in any geographic position, the ship's magnetic signature plus that caused by its movement through the earth's magnetic field is silenced. A combination of deperming and active degaussing processes is also employed. Deperming involves putting the ship inside an arrangement of coils or placing an arrangement of coils around the ship and then
passing a powerful electric current through the coils to create a magnetic field in a direction opposite to that of the magnetic field of the ship. This cancels the ship's magnetic signature. Alternatively, deperming is used to create a permanent magnetic field on the ship which is matched to the area in which it will operate. Degaussing coils are built into the ship during construction to provide magnetic field correction facilities. The coils are fed with electric current provided from special computer-controlled generators to create an opposing magnetic field which is continuously matched to the ship's changing magnetic field as it crosses the ocean.

**Electric signature**

Surface ships produce an extremely low frequency electric signature (ELFE) that can trigger mines or be detected by comparatively simple and inexpensive surveillance devices. The signature results when electric currents - which may result from the electrochemical reaction between the steel hull and the bronze propeller or from an active cathodic protection system - return to the hull through the shaft. The widely varying impedances of the shaft bearings and seals, as a function of shaft angle, cause a modulation of this return current which generates the ELFE wave.

**Other miscellaneous signatures**

There are other types of signatures associated with a ship by which it may be detected. One of the most common for which probably nothing can be done is the pressure signature. Another signature known as hydrodynamic wake can be detected up to 40 km behind the ship by remote sensing techniques. Radars on aircraft and satellites can provide sufficient evidence of presence of ships. The proper interpretation of such data can lead to actual ship identification. Proper hull design, decreased draft, contra-rotating propellers and non-conventional propeller systems like pump jet can reduce hydrodynamic wake. Chemical traces left in the discharges from cooling water or engine exhausts, bioluminescence caused by the disturbance of minute organisms, hydrodynamic pressure, and surface wave patterns can now be detected by satellites [3].

**Underwater noise signature**

The underwater radiated noise produces a source signal for detection to the enemy’s sonar system and torpedo, reducing the noise is very important in terms of the survivability of a naval vessel. Weapons like sea mines and torpedoes use different sensors and homing-in techniques, including underwater acoustic/noise signature to detect, classify and localize targets. To deal with this threat, ship underwater signatures are key elements of naval platform designs and operations. In addition, because the self-noise can affect the detection capability of the self-SONAR systems, it should be reduced in order to improve their detection performance. It is widely known that one of the main noise sources for surface ships is the cavitation noise from the propeller. However, before the cavitation occurs, the fluid-dynamic noise of a ship does not significantly affect the underwater radiated noise but the noises from machinery such as the propulsion engine, generator and pump are dominant. In fact, when the underwater radiated noise was measured for a surface ship at various speed conditions under CIS (cavitation inception speed), it can be found that the radiated noise
LITERATURE REVIEW

Hydroacoustics is a branch of technical sciences which incorporates many disciplines, such as physics (acoustics), signal analysis, electronics, sensors etc. These branches are widely used in planning and realization of not only numerous military systems, but also non-military applications. Weapons like sea mines and torpedoes use different sensors to detect, classify and localize targets. To deal with this threat, ship underwater signatures are key elements of naval platform designs and operations. Ship noise spectra are usually classified in two types: broadband noise having a continuous spectrum of cavitation and tonal noise which is full of discrete noise frequency or line components related to machinery, propellers, generators, gears, etc. The measurements were carried out in the Measurement And Control Acoustic ranges (MACARS), which is located in the southern part of the Baltic Sea. The MACARS are approximately 20 km long by 8 km wide and the depth of the water over the area varies between 5 to 20 m. This range contains a precise noise measurements system consisting of bottom-mounted hydrophones. Use of such system eliminated much of the low-frequency wave-induced noise. The hydrophones were calibrated to frequencies as low as 5 Hz. The sound intensity probe is designed to capture the sound pressure together with the unit direction of flow as a vector quantity. This is achieved by setting up more than one hydrophone in a probe to measure the sound energy flow. At the same time, on-board measurements of the vibration of the diesel and electric engines were carried out. Most of the information about the ship noise and

from machineries is more dominant rather than propeller noise. Since machinery noise can supply information about machines installed in a ship and leads to identification of ship or class of warship, it should be controlled to be under a preliminary defined level [3].

Ship noise spectra are usually classified in two types: broadband noise having a continuous spectrum of cavitation and tonal noise which is full of discrete noise frequency or line components related to machinery, propellers, generators, gears, etc. It is important to say that for speed lower than critical speed, the main source is the machinery running aboard ship which is higher than the propeller noises. On the other hand, when the ship is running above the critical speed the noise generated by the propeller is higher than the noise created by the machinery. This is due to the strong cavitation of the propeller [4].

The critical speed of the ship corresponds to the maximum achievable speed and is very specific, and exploitation at this regime can be avoided unless specifically required. As mentioned above the underwater noise generated by the machinery is major concern, as same is dominant over a broad range of exploitation regime. Since this also gives away the information about machinery fitted aboard like propulsion engines, power generation plant etc. and leads to the identification of the ship or class of ship. Therefore, it is required to limit the underwater noise to a desired level by redesigning that particular source or by providing some active or passive damping system.
its vibration was said to be located between 0 and 200 Hz [4].

Assuming that the underwater radiated noise only occurs from the bending vibration of a hull plate, the underwater radiated noise can be estimated with the hull vibration level if the sound radiation efficiency of the hull structure is well defined. The sound radiation efficiency is well defined by many researchers for a simple plate. However, when the underwater radiated noise is calculated with the sound radiation efficiency such as Maidanik’s and Uchida’s equations for an actual ship, it can be found that the estimated sound is not well coincident with measured one. Therefore, in this paper, the sound radiation efficiency of the actual ship based on Uchida’s sound radiation efficiency is suggested considering that it is varied in accordance with the size and shape of the hull plate [5].

Effective control of noise and vibration, whatever the application, usually requires several techniques, each of which contributes to a quieter environment. For most applications, noise and vibration can be controlled using four methods: absorption, use of barriers and enclosures, structural damping and vibration isolation [6]. Out of these four methods, this paper discussed about the structural damping and vibration isolation. Since our scope of work includes passive vibration damping, the same has been discussed further.

*Structural damping* dissipates vibrational energy in the structure before it can build up and radiate as sound. All materials exhibit a certain amount of damping, but many structural materials like steel, aluminium etc. have so little internal damping that their resonant behaviour makes them effective sound radiators. By putting a layer of material with high damping capacity
and depending upon the frequency range, these resonant frequencies can be controlled.

*Free-layer or extensional damping* (FLD) is one of the simplest forms of material application. The material is simply attached with a strong bonding agent to the surface of a structure. Alternatively, the material may be sprayed or painted onto the surface, or the structure may be dipped into a vat of heat-liquefied material that hardens upon cooling. Energy is dissipated as a result of extension and compression of the damping material under flexural stress from the base structure. Damping increases with damping layer thickness. Changing the composition of a damping material may also alter its effectiveness [6].

![Figure 4: Free layer damping (FLD) model](image4)

*Constrained-layer damping* (CLD) systems are usually used for damping materials with low stiffness. A “sandwich” is formed by laminating the base layer to the damping layer and adding a third constraining layer. When the system flexes during vibration, shear strains develop in the damping layer. Energy is lost through shear deformation, rather than extension, of the material. Further, varying layer thickness ratios permits optimizing system loss factors for various temperatures without changing the material’s composition. The constraint method is not critical as long as there is adequate surface-to-surface pressure. The layers may be bolted or riveted instead of glued into a sandwich and still provide optimum performance. Adhesives, if used, must have high shear stiffness. Shear strains in the adhesive will reduce the strains in the damping layer, reducing its effectiveness. Another advantage of CLD systems is that they can be used in harsh environments. The damping layer is totally covered by the top constraining layer, so it typically is not subject to abrasion or deterioration. Structural damping, whether extensional or constrained-layer, provides an at-the-source solution to vibration control problems [6].

![Figure 5: Constrained layer damping (CLD) model](image5)

Materials for vibration damping are mainly metals and polymers, due to their viscoelastic character. However, viscoelasticity is not the only mechanism for
damping. Defects such as dislocations, phase boundaries, grain boundaries and various interfaces greatly contributes towards the damping, because these defects /interfaces, mating surfaces may slightly move or slip over each other, thus dissipate energy. Therefore the microstructure of the material greatly affects its damping capacity. Besides this the damping capacity also depends in the loading frequency and temperature [7].

**Metals for vibration damping** – Some of the vibration damping metals are shape memory alloys (SMA’S), ferromagnetic alloys and other alloys. In SMA’S, phase (austenite and martensite) transformation can be induced by stress instead of temperature. Beyond certain stress, austenite transforms to martensite, and on removal of stress, the martensite transforms back to austenite. But there is large hysteresis between loading and unloading which indicates that a large part of strain energy applied on SMA’s is dissipated as heat, and this same mechanism causes vibration damping. In ferromagnetic alloys, due to movement of the magnetic domain in boundaries during vibrations, energy is dissipated and thus enhances vibration damping capacity. Whereas, other metal and alloys exhibit vibration damping capacity, due to movement and slippage at their interfaces and within their microstructure. Various metal alloys for vibration damping are shown in table 1. Also, a combination of these energy dissipation mechanisms can happen in same alloy [7].

<table>
<thead>
<tr>
<th>Ser no.</th>
<th>Base metals</th>
<th>Alloys</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Iron</td>
<td>Fe-Ni-Mn, Fe-Al-Si, Fe-Al, Fe-Cr, Fe-Cr-V, Fe-Mn, Fe-Mn-Co</td>
</tr>
<tr>
<td>2</td>
<td>Aluminim</td>
<td>Al-Ge, Al-Co, Al-Zn, Al-Cu, Al-Si, Alloys 6061, 2017, 7022 &amp; 6082</td>
</tr>
<tr>
<td>3</td>
<td>Tin</td>
<td>Sn-In</td>
</tr>
<tr>
<td>4</td>
<td>Zinc</td>
<td>Zn-Al</td>
</tr>
<tr>
<td>5</td>
<td>Titanium</td>
<td>Ti-Al-V, Ti-Al-Sn-Zr-Mo, Ti-Al-Nb-V-Mo</td>
</tr>
<tr>
<td>6</td>
<td>Other metals and alloys</td>
<td>Zirconium, Copper, Magnesium, Lead, Nickel</td>
</tr>
</tbody>
</table>
frequency and damping ratios were determined by conducting sine sweeps. The effect of hard coatings on the damping of the structural material can be compared in terms of quality factor ‘Q’.

\[ Q = \frac{1}{2\zeta} \]

3.1

Where, \( \zeta \) is the damping ratio. The quality factor was developed by electrical engineers as a measure of the clarity of a signal. The quality factor is inversely proportional to damping ratio. As damping increases, \( \zeta \) increases and Q decreases. Electrical engineers aim to improve signal quality, therefore high value of Q is desired whereas mechanical engineers trying to dissipate energy from a system desire low value of Q. Results of the damping test for each of the three cases (uncoated, thin coated, and thick coated thick plates) are compared in table 2 and exhibits a marked increase in damping from uncoated to the thin coated specimen, but no significant difference between the thin and thick coated specimens is observed [9].

**Table 2 Comparison of Quality factor (Damping capacity) [9]**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Uncoated</th>
<th>Thin coated (0.13mm)</th>
<th>Percent decrease</th>
<th>Thick coated (0.25mm)</th>
<th>Percent decrease</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>239</td>
<td>103</td>
<td>56.8</td>
<td>101</td>
<td>1.6</td>
</tr>
<tr>
<td>2</td>
<td>2610</td>
<td>110</td>
<td>57.8</td>
<td>115</td>
<td>-4.6</td>
</tr>
<tr>
<td>3</td>
<td>194</td>
<td>116</td>
<td>39.9</td>
<td>128</td>
<td>-10.2</td>
</tr>
</tbody>
</table>

Polymers for vibration damping – Polymers especially thermoplastics are good vibration damping materials. In general, elastomers and other amorphous thermoplastics with a glass transition temperature below room temperature are good for damping. However, polymers have low stiffness but can be used by sandwiching a layer of high damping polymer inbetween two layers of steel laminates i.e. CLD [7]. If viscoelastic material (VEM) gets strained due to harmonic stress, the strain is not in phase but lags behind by an angle \( \delta \), which is the measure of the damping in a material. The loss factor \( \eta \) of a material equals to \( \tan \delta \). Optimum design studies are also undertaken for a partially covered constrained damping treatment, in which it is observed that higher system loss factor \( \eta \) may be obtained by partially covering at suitable locations as compared to that of a fully covered case [10].

The storage shear modulus \( G \) and loss factor \( \eta \) of viscoelastic material are frequency and temperature dependent.

If all of the basic components of vibrating system like mass, damper and spring are working linearly, the resulting vibrations are known as linear, whereas, if any of one or more of the components work non linearly, then the vibration are said to be non-linear. Oil filled micro-channels structured within block made of Polydimethylsiloxane (PDMS) are also tested in constrained and free layer vibration damping experiments. An increase in fundamental frequency due to change in
stiffness was observed, whereas an increase in damping ratio and loss factor because of energy dissipation due to slip between oil film and walls of microchannel was observed. Nanostructured viscoelastic inclusion in polymers can significantly change the loss factor of a matrix over a wide range of frequency band. Embedment of nanoparticles of gold, silver and platinum in PDMS lead to increase in loss factor from 0.09 to 0.13 (about 1.4 times) in frequency range of 1-9Hz [13].

Ceramics for vibration damping – In general ceramics are not considered good for damping but are high temperature resistant and have high stiffness. However considering concrete as structural ceramic – a cement-matrix composite, addition of silica fume and latex as admixture results in large amount of interface and viscoelasticity respectively, and enhances its vibration damping capacity. The damping capacity of conventional ceramics and of high temperature ceramic-matrix composites (e.g MoSi2/Si3N4) can also be explored [7].

For study case – a stainless steel blisk with deposited Nicocraly+YSZ (yttria-stabilized zirconia) hard coating on both the sides of the blades is chosen. Energy equations of the blisk with hard-coated blades are derived and then substituted in Lagrange equations. Additionally, eigenvalue equations of blisk with hard-coated blades are acquired by taking advantage of Rayleigh-Ritz method, and its natural characteristics are obtained subsequently. The results of numerical calculations and experimental tests in terms of natural frequency and mode shapes are compared. The variation of natural frequency, modal loss factor and frequency response functions of the blisk generated by hard coating are studied and also influence of thickness of the coating is considered. In other cases, a hard coating with high hardness and better stability can be used as anti-friction coating, thermal barriers coating and anti-corrosive coating. In case of metal coatings, the coating thickness is generally thinner as compared to the base plate, thus enhanced damping can be achieved without significantly changing the structural mass and stiffness of the system [14].

LOGARITHMIC DECREMENT METHOD

The decrease in amplitude from one cycle to the next depends on the extent of damping in the system. The successive peak amplitudes bear a certain specific relationship involving the damping of the system, leading us to the concept of "logarithmic decrement". It is often necessary to estimate the extent of damping present in a given system. Essentially the experimental techniques to determine damping in a system fall into two categories - those based on free vibration tests and secondly those based on forced vibration tests. The latter require more sophisticated equipment/instruments, while the former is a relatively simple test.
In a free vibration test, based on the measured peak amplitudes over several cycles (and thus estimating the "logarithmic decrement"), one can readily find the damping factor for the given system. The amplitude at A and B are $Y_a$ and $Y_b$ at time $t_a$ and $t_b$ respectively. The periodic displacement from $Y_a$ to $Y_b$ represents a cycle. The ratio of any two successive amplitudes for an underdamped system, vibrating freely, is constant and is a function of the damping only.

$$\zeta = \frac{1}{2\pi} \log_e \frac{Y_a}{Y_b}$$

Sometime, in experiments, it is more convenient and accurate to measure the amplitudes after say "$n$" peaks rather than two successive peaks (because if the damping is very small, the difference between the successive peaks may not be significant). The logarithmic decrement can then be given by the equation

$$\zeta = \frac{1}{2\pi n} \log_e \frac{Y_a}{Y_b}$$

**STUDY PROBLEM STATEMENT**

This study explores the effect of free layer coatings on the vibration damping in the range of 0 Hz to 200 kHz. The influence of the thickness, multilayer or composite coatings on the damping behaviour is investigated and compared by means of the definition of some damping parameter i.e. damping ratio ($\zeta$) and quality factor (Q). To achieve this goal, following list of tasks was envisaged:

(a) To determine the effect of coating of damping material in low frequency range.
(b) Experimentally determination of effect of coating thickness on damping capacity.
(c) Experimental determination of effect of addition of Glass Fibre mat on damping capacity of composite coating.
(d) Experimental determination of damping capacity of Glass fibre and neoprene rubber, with different number of layer.
(e) To study the effect of fibre-polymer interface on damping.

**THEORY AND METHODS**

A passive method for vibration reduction of the ship’s structure and machinery foundation, by depositing coatings of material with high loss factor in the frequency range below 200 Hz has to be developed in conditions of high temperature and harsh conditions. An experiment set up has to be rigged to obtain the damping ratio $\zeta$ and other damping properties, and to experimentally determine the vibration damping capacity of sample materials. Prior proceeding with the experiments, to zero downs the various parameters of the beam and to enable us to work within the required frequency range of below 200 Hz, Finite element and analytical modelling has to be undertaken.

**FE modelling using Abaqus [16]**

Considering the cross section of various machinery foundations and structural
members used aboard ship, I-section and rectangular cross-section is considered for FE modelling and analysis. The resonant frequencies of the beam are dependent on its length and material, provided all other dimensions are kept constant. Since the material chosen for all specimens is mild steel, therefore the resonant frequencies now depend on the length only. FE models with different lengths are made and analysed. It is to be noted that the width of beam is not a factor in calculating resonant frequencies and mode shapes, but care must be taken during analysis to avoid any torsional vibration.

**Figure 6: Cantilever beam model with I cross section**

FE analysis of I-Section beams with different lengths is undertaken, whilst keeping other parameters constant. The various dimensions of I-section beam considered are,

- Length, $l = 0.2$ m to 1.6m
- Flange width, $d = 0.02$m
- Flange thickness, $t = 0.002$m

![Figure 7: I-beam in transverse vibrational mode](image)

**Figure 7: I-beam in transverse vibrational mode**

It is observed that I-section is prone to torsional vibrational at much lower resonant frequencies as shown in figure 9. Also, for a given length of specimen, it is found that resonant frequencies of I-section are much higher than that of rectangular cross-section and to work in desired lower frequency range, longer I-section beam specimens are required, which is not desirable [15].

**Figure 8: I-beam in torsional vibration mode**

Therefore, beam specimens with rectangular cross-section are considered. A cantilever beam with thickness t, width d and effective length L is considered for analysis, as shown in figure 10 and figure 11. Also $a$ and $b$ are the length and thickness of the clamping (root) portion of the beam, respectively [13]. The various dimensions of rectangular beam considered are, roots of $\omega$.

- Length, $L = 0.2$ m to 1.6m
- Width, $d = 0.01$m
- Thickness, $t = 0.002$m
- Root Length, $a = 0.04$m
- Root thickness, $b = 0.006$m
Experimental test approach

The material and the aspect ratio of the beams (test specimen) were chosen to obtain low resonant frequencies to enable us to work in the desired range of low frequency. The test specimens were tested in a cantilevered boundary condition using Free-decay method. Damping ratio is calculated at each mode, using log decrement method.

Total 13 in number samples were made and tested. Commercially available Mild Steel (MS) of cross-section 24mm x 03mm was used for making cantilever beams and various combination of coating were applied on it. The dimension of each cantilever beam was 850mm x 24mm x 3mm, wherein out of total length i.e. 850mm, 50mm is for clamping one end of beam to achieve fixed-free end condition i.e. the effective dimensions of the cantilever beam were 800mm x 24mm x 3mm.

Experiment setup

The response of the test article is measured using a Polytec PDV 100 Portable laser vibrometer, a non-contact measuring instrument, with built-in excitation signal generator, as shown in figure 13. It is designed to remotely, and without contact, measure a sample’s vibrational velocities in the frequency range up to 22 kHz. The test specimen is clamped vertically in a bench vice at one end, to achieve fixed-free boundary condition. The laser beam of the vibrometer is focused on the reflector tape pasted at the tip (free end) of the vertically positioned test specimen. The uncalibrated hammer is used to hit the free end of the test specimen, for conducting experiments.

Figure 12: A typical experimental setup in this study
The output of the vibrometer is connected to a Laptop through a NI USB-6211 Data Acquisition Card (DAQ). It is a multifunction DAQ device and offers analog I/O, digital I/O, and two 32-bit counters. The device provides an onboard amplifier designed for fast settling times at high scanning rates.

![Figure 13: Experimental instruments. (a) Test specimen vertically mounted in bench vice; (b) DAQ card, workstation with Laser vibrometer; (c) Laser vibrometer PDV 100 mounted on tripod.](image)

A GUI program, as shown in figure 14, is made in LabView 2017 [17] to receive data from DAQ card and to save it in time domain. It also performs the Fast Fourier transform (FFT) on the time-dependent data to produce a frequency spectrum as output. Peak amplitudes on the curve indicate resonance frequencies.

![Figure 14: A GUI program in LabVIEW](image)

Table 3 Technical data of Portable Digital Vibrometer (Polytech PDV 100)

<table>
<thead>
<tr>
<th>Metrological Specifications</th>
<th></th>
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<tbody>
<tr>
<td>Decoder type</td>
<td>Digital velocity decoder, 3 measurement ranges</td>
</tr>
<tr>
<td>Frequency range</td>
<td>0.5 Hz - 22 kHz</td>
</tr>
<tr>
<td>Measurement range (mm/s/V)</td>
<td>5 25 125</td>
</tr>
<tr>
<td>Full scale output (peak, mm/s)</td>
<td>20 100 500</td>
</tr>
<tr>
<td>Analog output</td>
<td>Velocity, ±4 V, 24-bit DAC</td>
</tr>
<tr>
<td>Calibration accuracy</td>
<td>±1 % (20 Hz ... 22 kHz)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Optical Specifications</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Laser type</td>
<td>Helium Neon (HeNe)</td>
</tr>
<tr>
<td>Laser class</td>
<td>Class 2, &lt; 1 mW output power, eye-safe</td>
</tr>
<tr>
<td>Laser wavelength</td>
<td>633 nm, visible red laser beam</td>
</tr>
<tr>
<td>Focus</td>
<td>Manual</td>
</tr>
<tr>
<td>Stand-off distance</td>
<td>90 mm ... ~ 30 m</td>
</tr>
</tbody>
</table>
MATERIALS USED
As mentioned earlier, commercially available Mild Steel (MS) of cross-section 24mm x 03mm was used as base metal for test specimens, on which different combination of composite layers were applied. An epoxy based High Damping Material (HDM), Neoprene rubber, Random Glass Fibre Mat with Epoxy (Araldit CY 230-1), were used individually and in different combinations, to make the composite coated samples. The coatings were applied manually and with the help of a mould. All HDM samples were brought to desired thickness by removing extra thickness using milling machine and emery paper.

Three different coating thicknesses of HDM were tested: 2.5mm, 4mm and 6mm. Two more test specimens of 06mm thickness were made using Glass Fibre mat and HDM. A coating with a layer of Glass Fibre mat strip (800mm x 24mm) sandwiched between HDM (total thickness of coating was 06mm, including Glass Fibre mat) was applied on MS beam. In another test specimen, Glass Fibre mat strip (800mm x 24mm) was disintegrated into loose fibres and mixed with HDM, and was then applied as coating of 06mm on MS beam. It is to be noted that the amount of Glass Fibre in both the test specimens are same, as the strip of same dimension was disintegrated to make the later test specimen. Four test specimen of coating thickness 1mm, 2mm, 3mm and 6mm were made using layers of 1mm thick neoprene sheet.

Three test specimens made using varying number of Glass Fibre mat impregnated with epoxy were also tested: one layer, two layers and three layers. These layers were applied using hand lay-up method. To prepare these three test specimen, total six strips of Glass Fibre mat (of dimension 800mm x 24mm) were taken. The total weight of these six strips was 56 grams. Epoxy of about twice the weight of the Glass fibre mat i.e. 112 grams was taken and mixed with the 11.2 grams of hardener. All the test specimens were cured at room temperature for about 48 hrs. The thickness of each
specimen is measured at nine different points, 100 mm apart, using a digital Vernier calliper. In case of Glass fibre mat coated test specimens, the thickness was not constant along the length, as this cannot be machined to maintained constant dimensions. However, the variation in thickness was not more than 0.2 mm at any point.

**Table 4: Test Specimen parameters**

<table>
<thead>
<tr>
<th>Test Specimen</th>
<th>Coating thickness (mm)</th>
<th>Coating Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sample 2</td>
<td>2.5</td>
<td>HDM</td>
</tr>
<tr>
<td>Sample 1</td>
<td>4</td>
<td>HDM</td>
</tr>
<tr>
<td>Sample Without mat</td>
<td>06</td>
<td>HDM</td>
</tr>
<tr>
<td>Sample Layer mat</td>
<td>06</td>
<td>HDM + Glass Fibre mat (E-Glass)</td>
</tr>
<tr>
<td>Sample Random mat</td>
<td>06</td>
<td>HDM + loose Glass Fibre (E-Glass)</td>
</tr>
<tr>
<td>GLayer 1</td>
<td>1.2 (approx.)</td>
<td>Glass Fibre mat (E-Glass) + epoxy (Araldit CY 230-1)</td>
</tr>
<tr>
<td>GLayer 2</td>
<td>2 (approx.)</td>
<td>Glass Fibre mat (E-Glass) + epoxy (Araldit CY 230-1)</td>
</tr>
<tr>
<td>GLayer 3</td>
<td>3 (approx.)</td>
<td>Glass Fibre mat (E-Glass) + epoxy (Araldit CY 230-1)</td>
</tr>
<tr>
<td>Neoprene 1</td>
<td>1</td>
<td>Neoprene sheet pasted using abond adhesive</td>
</tr>
<tr>
<td>Neoprene 2</td>
<td>2</td>
<td>Neoprene sheet pasted using abond adhesive</td>
</tr>
<tr>
<td>Neoprene 3</td>
<td>3</td>
<td>Neoprene sheet pasted using abond adhesive</td>
</tr>
<tr>
<td>Neoprene 6</td>
<td>6</td>
<td>Neoprene sheet pasted using abond adhesive</td>
</tr>
</tbody>
</table>

**COMPARISON OF DAMPING RATIO AND QUALITY FACTOR FOR DIFFERENT COATINGS**

Results of the experimental test for each of the five cases (uncoated, HDM coated, composite coated, Glass Fibre mat coated and Neoprene coated test specimen) are compared in table 5 and exhibits a marked increase in damping from that of uncoated test specimen. Significant increase in damping has been observed in case of HDM coated specimen; also damping in case of Glass fibre mat and neoprene coated specimen is also appreciable. However, reverse trend has been observed in case of composite (HDM + Glass fibre mat) coatings i.e. decrease in damping capacity is observed.

**Table 5: Damping Ratio and Quality Factor**

<table>
<thead>
<tr>
<th>Sample</th>
<th>Coating Thickness (mm)</th>
<th>1st resonant Frequency (Hz)</th>
<th>2nd resonant Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uncoated</td>
<td>0</td>
<td>0.001673</td>
<td>0.000502</td>
</tr>
<tr>
<td>Sample 2</td>
<td>2.5</td>
<td>0.015699</td>
<td>0.011498</td>
</tr>
<tr>
<td>Sample 1</td>
<td>4</td>
<td>0.037615</td>
<td>0.004181</td>
</tr>
<tr>
<td>Without Mat</td>
<td>6</td>
<td>0.076739</td>
<td>0.06192</td>
</tr>
<tr>
<td>Layer Mat</td>
<td>6</td>
<td>0.064244</td>
<td>0.037829</td>
</tr>
<tr>
<td>Random Mat</td>
<td>6</td>
<td>0.076412</td>
<td>0.056879</td>
</tr>
<tr>
<td>GLayer 1</td>
<td>1.2</td>
<td>0.002331</td>
<td>0.001634</td>
</tr>
<tr>
<td>GLayer 2</td>
<td>2</td>
<td>0.00334</td>
<td>0.002311</td>
</tr>
<tr>
<td>GLayer 3</td>
<td>3.2</td>
<td>0.004982</td>
<td>0.003683</td>
</tr>
<tr>
<td>Neoprene 1</td>
<td>1</td>
<td>0.00199</td>
<td>0.001089</td>
</tr>
<tr>
<td>Neoprene 2</td>
<td>2</td>
<td>0.002305</td>
<td>0.001574</td>
</tr>
</tbody>
</table>
Damping ratios of all the materials / test specimens at 1st resonant frequency and 2nd resonant frequency is shown separately in figure 18 and 19

Damping capacity in terms of quality factor is shown in tables below. Table 6 compares the damping capacity of various thicknesses of HDM coating and its effect is compared in terms of percent decrease with each coating thickness. Table 7 compares the effect of different combination of Glass fibre and HDM on damping capacity, as the coating thickness is same (i.e. 0.06mm) for all three test specimens, therefore percent decrease is w.r.t quality factor of Sample-Without mat. Table 8 and 9 compares the effect of number of layers of Glass fibre mat and neoprene, respectively on damping
capacity, and its effect is compared in terms of percent decrease w.r.t that of uncoated test specimen.

![Graph showing damping ratio vs coating thickness](image)

Figure 21: Damping ratios in 2nd resonant frequency

**Table 6: Comparison of Quality factor of HDM coating (Damping capacity)**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Uncoated</th>
<th>Sample 2 (2.5 mm)</th>
<th>Sample 1 (4 mm)</th>
<th>Sample Without mat (6mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Q</td>
<td>Q</td>
<td>Percent decrease</td>
<td>Q</td>
</tr>
<tr>
<td>1</td>
<td>298.77</td>
<td>31.84</td>
<td>89.34</td>
<td>13.29</td>
</tr>
<tr>
<td>2</td>
<td>994.62</td>
<td>65.75</td>
<td>93.38</td>
<td>58.39</td>
</tr>
</tbody>
</table>

**Table 7: Comparison of Quality factor of Composite coating (Damping capacity)**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Uncoted</th>
<th>Neoprene 1 (One layer) (1 mm)</th>
<th>Neoprene 2 (Two layers) (2 mm)</th>
<th>Neoprene 3 (Three layers) (3 mm)</th>
<th>Neoprene 6 (Six layers) (6 mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Q</td>
<td>Q</td>
<td>Percent decrease</td>
<td>Q</td>
<td>Percent decrease</td>
</tr>
<tr>
<td>1</td>
<td>298.77</td>
<td>25</td>
<td>12.5</td>
<td>15.9</td>
<td>21</td>
</tr>
<tr>
<td>2</td>
<td>994.62</td>
<td>45</td>
<td>9.1</td>
<td>53.8</td>
<td>31</td>
</tr>
</tbody>
</table>

**Table 8: Comparison of Quality factor of Glass Fibre mat coating (Damping capacity)**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Uncoated</th>
<th>GLayer 1 (One layer) (1.2 mm)</th>
<th>GLayer 2 (Two layers) (2 mm)</th>
<th>GLayer 3 (Three layers) (3.2 mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Q</td>
<td>Q</td>
<td>Percent decrease</td>
<td>Q</td>
</tr>
<tr>
<td>1</td>
<td>298.77</td>
<td>214.51</td>
<td>28.20</td>
<td>149.68</td>
</tr>
<tr>
<td>2</td>
<td>994.62</td>
<td>306.05</td>
<td>69.22</td>
<td>216.39</td>
</tr>
</tbody>
</table>

**Table 9: Comparison of Quality factor of Neoprene coating (Damping capacity)**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Uncoted</th>
<th>Neoprene 1 (One layer) (1 mm)</th>
<th>Neoprene 2 (Two layers) (2 mm)</th>
<th>Neoprene 3 (Three layers) (3 mm)</th>
<th>Neoprene 6 (Six layers) (6 mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Q</td>
<td>Q</td>
<td>Percent decrease</td>
<td>Q</td>
<td>Percent decrease</td>
</tr>
<tr>
<td>1</td>
<td>298.77</td>
<td>25</td>
<td>12.5</td>
<td>15.9</td>
<td>21</td>
</tr>
<tr>
<td>2</td>
<td>994.62</td>
<td>45</td>
<td>9.1</td>
<td>53.8</td>
<td>31</td>
</tr>
</tbody>
</table>
CONCLUSION AND RECOMMENDATIONS

Conclusion
Considering the cross section of various machinery foundations and structural members used aboard ship, I-section and rectangular cross-sections were considered for FE modelling and experiments. The resonant frequencies of the beam are dependent on its length and material, provided all other dimensions are kept constant. However, the width must be chosen carefully during analysis to avoid any torsional vibration.

It is observed that, I-section is prone to torsional vibrational at much lower resonant frequencies. Also, for a given length of specimen, it is found that resonant frequencies of I-section are much higher than that of rectangular cross-section and to work in desired lower frequency range, longer I-section beam specimens are required, which is not desirable. Thus, test specimens with rectangular cross-section are considered. During FE analysis it was observed that the accuracy of output resonant frequencies depends upon the meshing size. To optimise meshing size, analysis was undertaken by varying the meshing size from 0.01m to 0.002m, in a model with constant length. It is found that the output resonant frequencies are constant beyond the meshing size of 0.02m, and accurate results for all test specimens were obtained with mesh size of 0.002m. Further, it was observed that higher the frequency, finer the mesh size is required to obtained the accurate results. The comparison of natural frequencies of the cantilever beam with rectangular cross-section for various values of its length, whilst keeping other dimensions and material properties constant, shows that the test specimen with effective length of 0.8m is suitable for further experiments, as its first five resonant frequencies are well below 200 Hz. As, most of the information about the ship noise and its vibration is located between the frequency range of 0 and 200 Hz, further experiments and analysis was done using the mild steel specimens with length 0.8m.

This study was undertaken to compare damping capacity of the uncoated, HDM, composite layer and epoxy impregnated Glass Fibre mat coated test specimens.

Results of the damping test for each of the HDM coated, composite coated and Glass Fibre mat coated test specimen exhibits a marked increase in damping capacity from that of uncoated test specimen. The increase in damping capacity of the HDM and Glass fibre mat coat material observed is also appreciable. However, appreciable drop in damping capacity has been observed in case of composite coatings with a layer of Glass Fibre mat sandwiched between HDM (Sample – Layer mat). A minor drop in damping capacity has also been observed in case of composite coating with random loose Glass Fibres mixed with HDM (Sample - Random mat). The damping in the abovementioned viscoelastic coatings can be attributed to the following,

(a) viscoelastic damping in the polymer matrix,
(b) hysteretic damping in the fibres, and
(c) friction damping at the fibre-polymer interface.

With the increase in HDM coating thickness, the damping capacity has also been increasing. This can be explained considering, viscoelastic damping in the polymer matrix, as with the increase in amount of damping material, more energy is consumed or dissipated. The reason for a sharp drop in damping capacity of Sample-Layer mat is that, when the composite materials were suffered from vibration perpendicular to the composite sample, it is subjected to bending stress, while the middle layer is subjected to shear stress. When the Glass Fibre mat with poor viscoelastic (material) damping properties was deposed in the middle layer, it may produce a larger deformation but consume less energy, thereby decreasing the damping performance of the system as a whole. Also, hysteretic damping in the fibres, and friction damping at the fibre-HDM interface is negligible, which otherwise would have compensated for the loss in damping capacity.

In case of Sample-Random mat, in which coating of loose glass fibres mixed with HDM is applied, slight drop in damping capacity can be attributed to poor hysteretic damping in the fibres, and poor friction damping at the fibre-HDM interface.

In case of coatings of Glass Fibre mat impregnated with epoxy resin, there is increase in damping capacity with increase in number of layers (or thickness) of Glass Fibre mat. If the epoxy resin with better damping properties is used, the overall damping properties of composite materials will improve. Glass Fibre with better viscoelastic properties and hysteretic damping can also enhance the overall damping capacity of the system. Also, when the composite material is subjected to stress which is passed to the fibres by the resin matrix, due to a larger deformation produced from the damping resin on the fibre surface, materials with better friction damping at the fibre-polymer interface, can increase energy dissipation at this interface, which can increase the overall damping capacity of the system. The length of test specimens can be reduced further to conduct experiments to obtain damping ratio at higher frequencies. Also, using better signal filtering techniques, higher resonant frequencies can be filtered out to obtain damping ratio at higher modes. The results obtained by free decay method can be compared with other experimental and analytical methods to establish the efficacy of the present approach.

**Recommendations**

Structures subjected to dynamic loads, generally show structural damping values which are just slightly capable of reducing oscillations amplitude. Onboard naval platforms, low structural damping, or high oscillations amplitudes, may impact negatively on structural vibrations and underwater emitted noise. By increasing structural damping it is possible to obtain a considerable underwater noise and vibration reduction. Viscoelastic materials coating is among the most achievable damping treatments that can be applied onboard naval platform.

The research activity can be split up in two parts: the first one related to experimental tests; the second related to the numerical simulations. About the
experimental part in present study, the objectives had been primarily the identification and validation of a procedure to extract the damping ratio and, subsequently, to characterize the performance of different test materials in low frequency range. About the numerical part, the objective has to be the identification of a numerical procedure, able to give output as same as that obtained experimentally. Thus, further work can be undertaken in the direction to validate the FE model with the obtained experimental results.

In the present work, damping ratio at first two modes is calculated, because of the limitation of signal filtering at higher resonant frequencies. However, further work can be undertaken to establish a methodology to obtain damping ratio at higher modes. Some experiments at different temperature range can be undertaken to establish its dependency on vibration damping capacity of the materials. Also using present experimental setup, test specimen of smaller length can be tested to obtain damping ratio at higher frequencies. Conducting experiments for test specimens of different lengths at different temperature can be envisaged to establish behaviour of vibration damping capacity of a material at different frequencies and temperatures.

Vibration damping using free layer coating of hard materials can also be explored. Hard coatings for vibration damping can be applied at machinery components and structures that are effective at the temperature and vibratory stress levels seen by those components. This hard damping coating material will absorb enough of the energy generated by the vibrations to prevent, or significantly postpone, the failure of the component to which it has been applied. Damping capacity of thermal barrier coatings like Magnesium aluminate spinel, for blades in gas turbines can also be explored and further enhanced to mitigate vibrations at root level, which in turn will enhance the life of the equipment.

REFERENCES


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