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CONSIDERATIONS FOR UNDERWATER DECOUPLING TREATMENTS

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1. INTRODUCTION

The reduction of noise radiation from a structure immersed within a waterborne environment has been conventionally achieved by application of a selected compliant layer to the structure. The compliant layer, usually a closed cell foam, acts like the spring or mounting used to isolate vibrating machinery, acoustically isolating or decoupling the noisy structure from the surrounding water. In common with isolation mountings, the effectiveness is critically dependant on the stiffness of the spring and mass of the structure. For any fixed stiffness and mass the isolation performance (transmissibility) can be divided into three frequency regimes; a low frequency range where nose vibration levels are actually enhanced, a higher frequency range beyond the so called "cut off" frequency, where noise radiation is reduced and finally a high frequency area where thicknesswise resonances reduce the effectiveness of the coating at discrete frequencies. Typically machinery or engine masses are easily defined and will usually equate to their rigid body mass (ie total mass), however, for large hollow structure such as ships or pipelines the effective mass is not as easily defined. This paper demonstrates the importance of quantifying the dynamic mass of the structure and its influence on the performance of selected decoupling coatings.

2. BACKGROUND

2.1 Application of decoupling coatings to structure immersed in water is a successful and well established technique. The material basically consists of a layer of a low modulus polymer rubber enclosing a series of individual air filled cells in such a way as to form a closed cell foam. A solid layer of rubber has a density, wave velocity product (acoustic characteristic impedance) similar to that of water. The inclusion of air in the rubber serves to increase its compliance and to reduce its density, thereby reducing its characteristic impedance. The resulting mismatch in impedance at the rubber layer/water interface reduces the transfer of acoustic pressure into the water thus reducing noise radiation from the structure.

2.2 Generally the prediction of decoupling performance can be achieved using normal incidence plane wave transmission line theory by comparing the noise transmitted from the unclad structure or ship hull, to that radiated from the same structure when coated with the decoupling treatment. Thus referring to Figure 1 the transmission loss from air through the hull plate, decoupling layer and into the water is given by:

$$TL_4 = 20 \log \frac{1}{2}\left[\left(1 + \frac{Z_1}{Z_4}\right)\cos k_2 l_2 \cos k_3 l_3 - \sin k_2 l_2 \sin k_3 l_3 \left(\frac{Z_2}{Z_3} + \frac{Z_1 Z_3}{Z_2 Z_4}\right) + j\cos k_2 l_2 \sin k_3 l_3 \left(\frac{Z_1}{Z_3} + \frac{Z_3}{Z_4}\right) + j\sin k_2 l_2 \cos k_3 l_3 \left(\frac{Z_1}{Z_2} + \frac{Z_2}{Z_4}\right)\right]$$

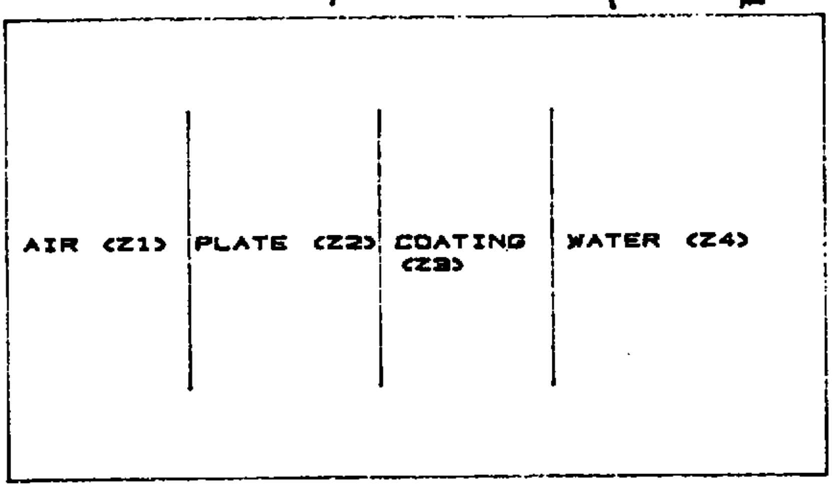

FIG 1

where Z_1 - The acoustic impedance of air
Z_2 - The acoustic impedance of the structure plating
Z_3 - The acoustic impedance of the layer
Z_4 - The acoustic impedance of water
k_2 - The wave number for hull plating
l_2 - The thickness of hull plating
k_3 - The wave number for the layer
l_3 - The thickness of the layer
k = $2\pi f/c$
f - frequency
c - wave velocity

To obtain the decoupling performance (DP) it is now necessary to subtract the transmission loss TL_3 for the 3 media air, hull plate and water (ie the unclad structure) thus:

$$DP = (TL_4 - TL_3)\text{dB}$$

2.3 Computation of this equation for an air backed structure when coated with a compliant layer, immersed in water, results in a typical decoupling performance curve as shown in Figure 2 illustrating the three features discussed earlier that:

2.3.1 Peak performance occurs at odd multiples of quarter wavelengths in the decoupling layer where the frequency of the peak is determined by the wave velocity and thickness of the layer.

2.3.2 Performance minima occur at multiples of half wavelengths, the frequency of which is again determined by the wave velocity and thickness of the layer, and

2.3.3 There is a low frequency "cut off" limit where decoupling ceases and noise radiation is actually increased.

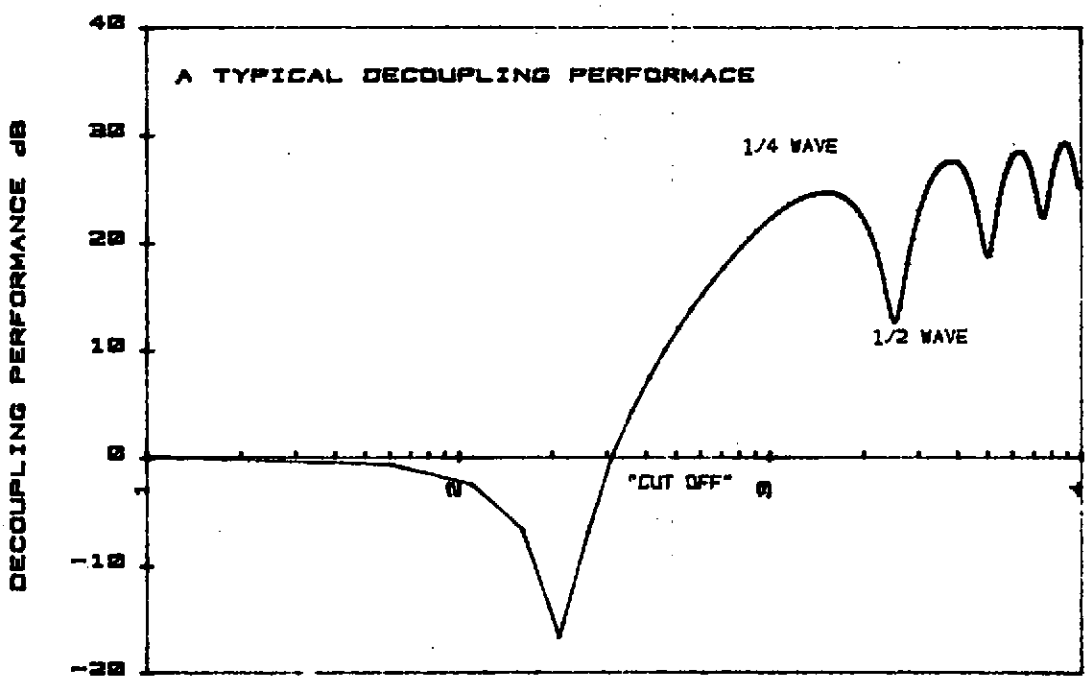

FIG 2

3. Obviously half wave resonances and noise enhancement must be restricted to frequencies considered to be unimportant. In practice the half wave resonance does not result in complete loss of decoupling performance as the rubber materials in use are never perfectly elastic and the noise radiation reduction is assisted by the viscoelastic loss within the rubber. Unfortunately, the low frequency noise enhancement cannot be avoided, but the "cut off" frequency can be shifted to lower frequencies by modification

to the stiffness of the decoupling layer. Thus it can be shown that the cut off frequency F_r is given by:

$$F_r = \frac{1.414}{2\pi}\sqrt{\frac{S}{M}}$$

where S = the layer stiffness

M = the apparent mass of substrate acting on the layer

F_r = the cut off frequency

4. It is apparent that accurate prediction of the "cut off" frequency for any particular coating of known stiffness can only be achieved with knowledge of M. For an engine mounting it may be sufficient to assume M to be the total rigid body mass, but for a large structure such as a ship this does not represent the effective mass acting on the decoupling layer. On the other hand it may be thought sufficient just to consider the hull plate whose thickness and density are known and feed this into the transmission line equation. However, experience has shown that "cut off" frequencies are considerably lower than that predicted assuming hull plate mass alone. In fact a ships hull consists of more than hull plating and the addition of frames and stiffeners as shown in Figure 3 for a typical ship structure must be taken into account.

SHIP HULL SECTION

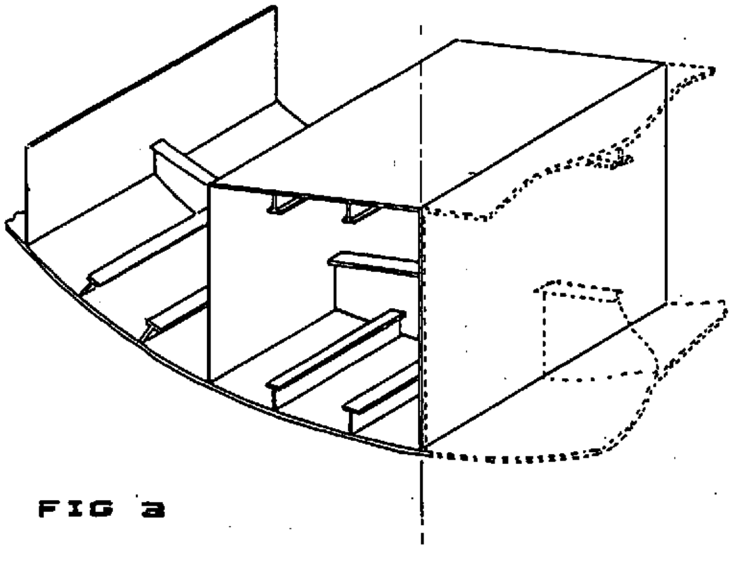

FIG 3

The hull is highly modal in nature and the mass to be considered therefore has to be the dynamic mass of the hull. This can be shown to vary considerably with frequency such that at high frequencies the dynamic mass does approach that of the hull plate and can under conditions of resonance by much lower. However, at frequencies lower than the fundamental frequency of the individual hull plates as bounded by the frames and stiffeners, then the mass equates to the local mass of hull plate plus frames and stiffeners. At lower frequencies, as the wavelengths in the hull increase, so will the dynamic mass increase as the additional loading effect of bulkheads and machinery become noticeable until at 0 Hz when the effective mass will be that of the vessel.

5. An opportunity was taken to demonstrate this effect by attempting to measure the dynamic mass of a ships hull whilst in dry dock. the technique employed was to use an impedance hammer consisting of a 20 kN force gauge attached to a 1 kg hammer head. The ship's hull was tapped with the hammer at various positions on the hull plate including keel, frames and longitudinals while at the same time monitoring the local acceleration levels. The force and acceleration were then fed into a dual channel spectrum analyser where an FFT was performed on the two impulses, the dynamic mass then established by dividing the force by the acceleration. Figure 4 illustrates some typical examples of dynamic mass of a ships hull where it can be seen just how rapidly it increases as the frequency decreases. As might be expected at high frequencies the dynamic mass in the vicinity of the frames, longitudinals and keel are generally higher than on the hull plate but at low frequencies they tend to coalesce.

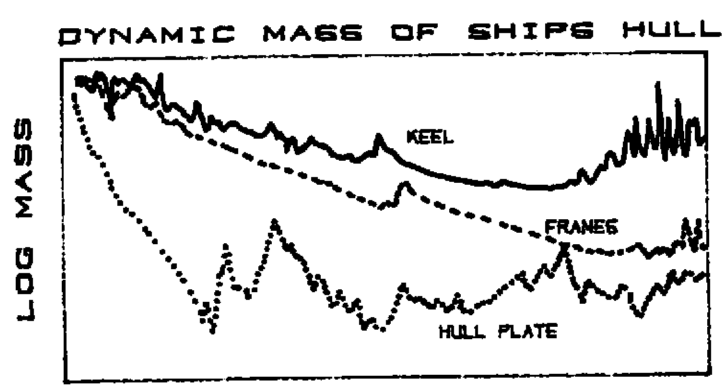

FIG 4

6. In order to confirm the low frequency mass effect on the decoupling performance, a simulated scale section of ship's hull was constructed as shown in Figure 5. The plate, frames and longitudinals approximated to 1/10th scale and although not exact were thought to be close enough to

demonstrate the particular effect under investigation. The 1/10th scale section represented an area of hull 5.6 m x 4.2 m, measuring in reality, 560 mm x 420 mm using 1.6 mm steel sheet for the hull plate, longitudinals and frames all joints being continuously welded.

1/10th SCALE FRAMED HULL SECTION (STEEL)

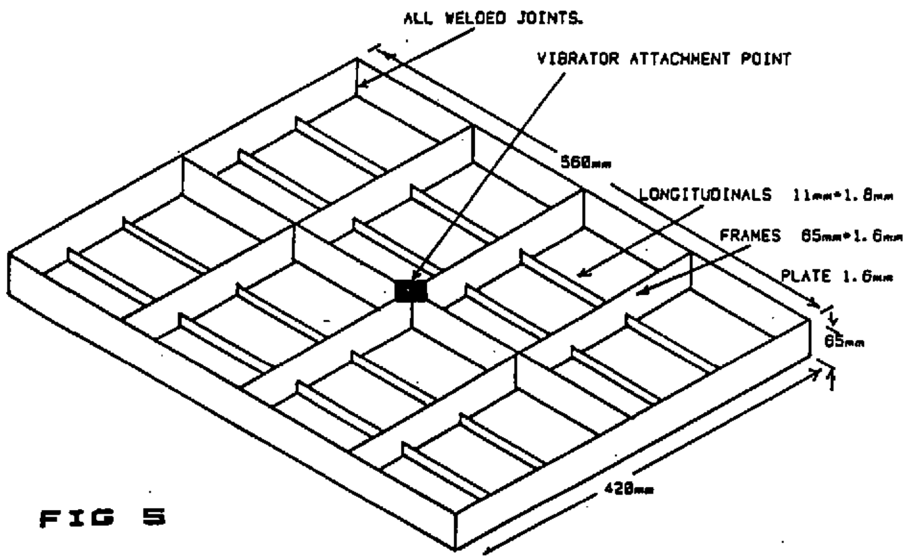

FIG 5

7. The scaled section was then floated on the surface of a 5 m deep anechoic tank facility, a lightweight mechanical vibration generator attached to the centre of the section and a hydrophone mounted 2.5 m below. The section was then excited with a haversine impulse of centre frequency 6 kHz and the noise radiated into the water monitored by the hydrophone. An FFT was performed on the radiated impulse as received by the hydrophone, and the frequency response recorded. The same experiment was repeated for the scaled section when coated with a 6 mm layer of Evazote (ethyl vinyl acetate) a low density closed cell foam of density 80 kgm^{-3} and wave velocity 140 ms^{-1}.

8. The decoupling performance of the Evazote coating when applied to the scaled section is shown by the solid line in Figure 6, also illustrating the typical features expected from predictions (as indicated by the dashed line). In this case the total mass of the 1/10th scale section is 7.4 kg which for simplicities sake has been equated to a layer of steel 4 mm thick. Applying this data to the model a "cut off" frequency of 700 Hz is predicted comparing well with the 710 Hz measured. At frequencies above 2 kHz however, the decoupling performance is lower than predicted and tends towards that predicted assuming the dynamic mass equates to a layer of steel 1.6 mm thick (ie the thickness of the plate used to cover the frames).

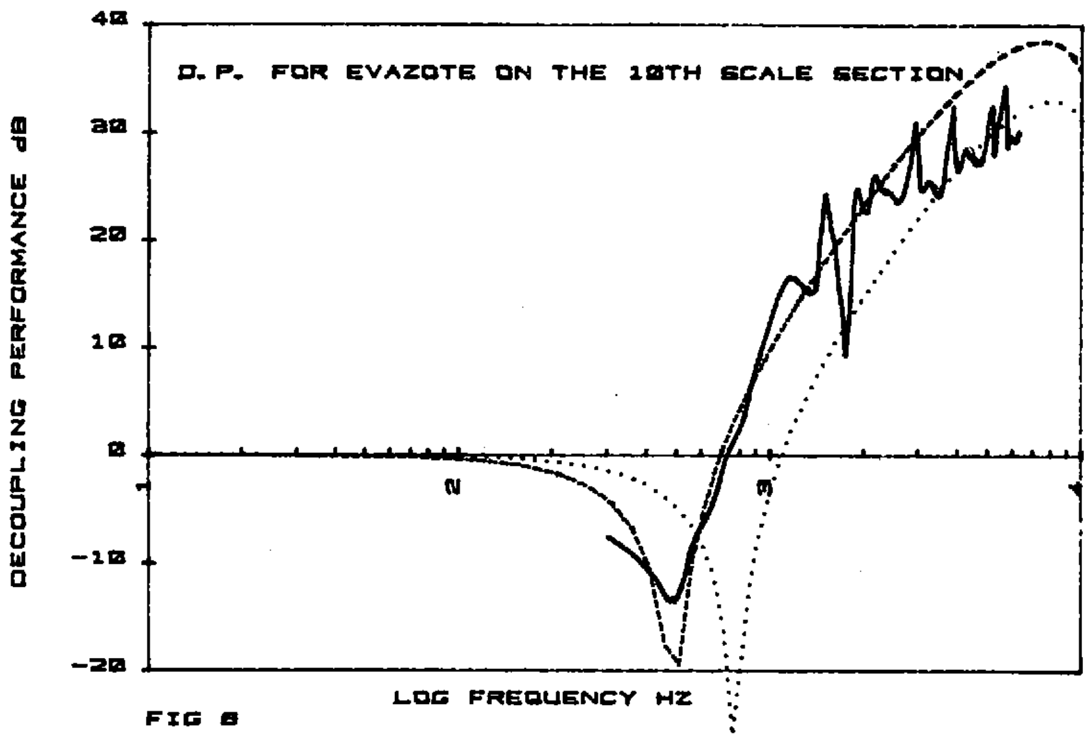

FIG 6

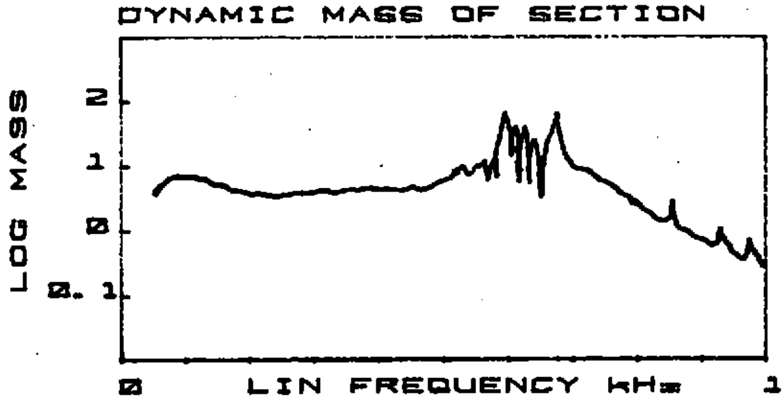

FIG 7

9. Observation of Figure 7 shows how a typical dynamic mass measurement on the 1/10th scale section varies with frequency over the range 0 to 1 kHz. Over the frequency range 100 Hz to 600 Hz the dynamic mass is approximately that of the static mass, between 600 Hz and 750 Hz there is some variation in dynamic mass due to the modal nature of the structure, then at

frequencies above 750 Hz the dynamic mass falls considerably below that of the static mass. It is suggested that the reason for the reduction in measured decoupling performance at frequencies above 2 kHz is due to the effective reduction in dynamic mass of the 1/10th scale section.

10. CONCLUSIONS

The use of attached acoustically compliant layers to ship hulls to reduce waterborne radiated noise is a well known technique, however, accurate prediction of decoupling performance is possible only if the dynamic mass of the hull structure is known. Experience has shown that at high frequencies the dynamic mass varies considerably depending on where on the hull the measurements are taken and what form of stiffening is used behind the hull plate. Accurate prediction of decoupling performance of a clad hull at these frequencies may be difficult and this has not been attempted. At lower frequencies however, the dynamic masses tend towards a common value and accurate prediction at these frequencies should present no problem. This is particularly useful in predicting the decoupling "cut off" frequency for any particular coating of known stiffness as demonstrated by the 1/10th scale model test. At higher frequencies the modal nature of the hull will mean a large variation in dynamic mass particularly if the structure has a high Q, this can have an adverse effect on the decoupling performance particularly at resonant frequencies where the dynamic mass falls. In such circumstances it is suggested that the additional use of applied damping to the structure would considerably improve the resulting decoupling performance.