APPLICATION OF STATISTICAL ENERGY ANALYSIS TO HIGH FREQUENCY SHIP VIBRATION

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Introduction

In the field of ship design, vibration considerations are increasingly important. The complexity of the ship structure and the excitations which arise from the main engine, propeller and sea result in vibration response over a frequency range from ½Hz to 8KHz.

Vibration at the lower frequencies, say up to 20Hz has been associated with perhaps the more severe problems such as crew discomfort and fatigue damage arising from dynamic stresses. As a result of concentrated research effort over the past few years, tools are now available to minimise these potential problems early in the design stage.

In the higher frequency range, the main consequence of the vibration is noise radiated from panels into accommodation and watch keeping spaces. Contractual and/or statutory considerations require that noise in these spaces must be below stated levels. Consequently it is essential that estimating methods should be available. Current methods are basically empirical and rely on a bank of experimental data for noise source strengths and transmission coefficients. These methods have been used successfully by BSRA, and other organisations, but tend to be applicable only to the ship types from which the data was obtained. What is required is a calculation procedure with an improved theoretical basis which will be applicable to a wider variety of general ship types and to novel designs. Published literature on the application of SEA methods to ship structures suggests that SEA may be usefully employed as a basis for noise prediction but that there are areas of difficulty.

The formulation of an SEA model requires definition of a set of sub-systems, each of which is a discrete part of the structure, usually permitted to vibrate with either flexural or longitudinal waveforms. Prediction of the vibrational energies of the sub-systems in a specified frequency band depends on the following parameters which constitute the energy balance equations:

Modal density	- the number of modes in the frequency band/frequency bandwidth
Mechanical Loss Factor (MLF)	- related to energy dissipated within a sub-system
Coupling Loss Factor (CLF)	- governs the energy flow between sub-systems

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Appropriate values for these parameters may be obtained from the literature.

Investigations Carried Out

For the lighter, repetitive structure of a ship's accommodation for example, the use of the basic SEA formulation might be adequate. This possibility was investigated at BSRA using a full-scale "mock-up" of two cabins situated between three deck levels as corresponding to a typical merchant ship. The fabricated stiffened panel structure is of overall dimensions 6.5m x 4.5m x 5m.

The distribution of vibration velocity levels over the eleven surfaces of the model, assuming a power input of 1W at the lowest deck, was calculated on a very simple basis. The model was assumed to consist of eleven flat, homogenous, isotropic plates vibrating flexurally. The MLF chosen were 4×10^{-3} for the lowest deck and 2×10^{-3} for all other boundaries.

The experimental work was carried out by exciting the lowest deck with an electrodynamic vibration exciter and measuring the resulting vibration on the panels, using small accelerometers. The input signal was obtained from a random noise generator, the output of which was passed through a set of third octave filters. The frequency range covered was 100Hz to 5KHz.

Examination of the distribution of vibration levels was carried out in some detail. The comparison between measured and calculated values is typified by the results which are shown in Fig. 1. Fig. 1(a) shows both the measured and calculated difference between the average vibration levels at the lowest deck and the mid height deck. With the exception of the 200Hz and 250Hz third octaves, the measured and calculated values are in good agreement. Less satisfactory results were obtained for comparison of the lowest and the uppermost decks (Fig. 1(b)).

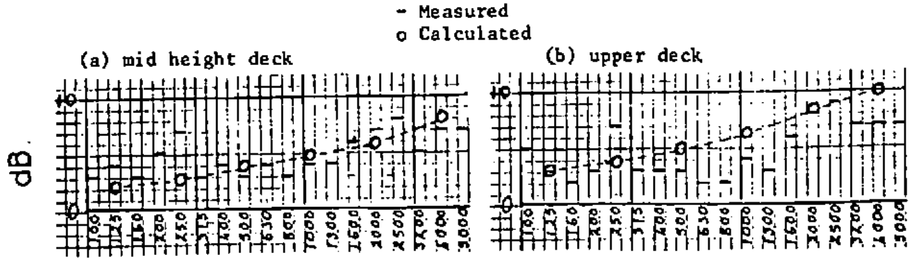

Fig. 1 Attenuation in Average Vibration Levels from Lowest Deck

The comparison between calculated and measured attenuations is considered to be encouraging and to suggest that the basic SEA approach is directly applicable to more extended structures of relatively light, uniform scantlings.

However, much of a ship is composed of relatively heavy structural components. For example, the grillage shown in Fig. 2 represents one of the decks in a typical engine room and consists of steel plate 8m x 12m x 12.5mm supported by a large girder and a matrix of heavy stiffeners. Clearly this cannot be considered as one sub-system and to represent each sub-panel and stiffener as a SEA sub-system is not practicable if many other structural components are to be considered in the overall model of the ship. Consequently, an equivalent modelling scheme has been evolved by the authors based on a modification to the conventional definitions of the SEA parameters. Identical stiffeners are combined into a single sub-system with larger unstiffened plates spanning several frames and for energy transmission by flexural waves only as follows:-

For n stiffeners combined:-

Thickness of equivalent stiffener = $\sqrt{n}$ x Stiffener thickness

Area of equivalent stiffener = $\sqrt{n}$ x Stiffener area

MLF of equivalent stiffener = $n^{0.7}$ x Stiffener MLF

MLF of stiffened deep beam = MLF for beam only x

$$\frac{2 \times \text{sectional length of beam} + \text{length of flat}}{2 \times \text{sectional length of beam}}$$

Length of junction with plate = $(0.7)^{n-2}$ x physical length. This latter modification thus reduces the CLF at each end of an extended plate sub-system.

Fig. 2(a) shows calculated velocity levels for the engine room deck using an SEA model comprising 77 sub-systems. A unit power input of frequency 1KHz is applied at sub-system (1). Using the above suggested modelling scheme, a smaller model comprising 28 sub-systems was also formed. Fig. 2(b) shows that the results agree fairly closely with those for the full model, this being the case for each frequency considered.

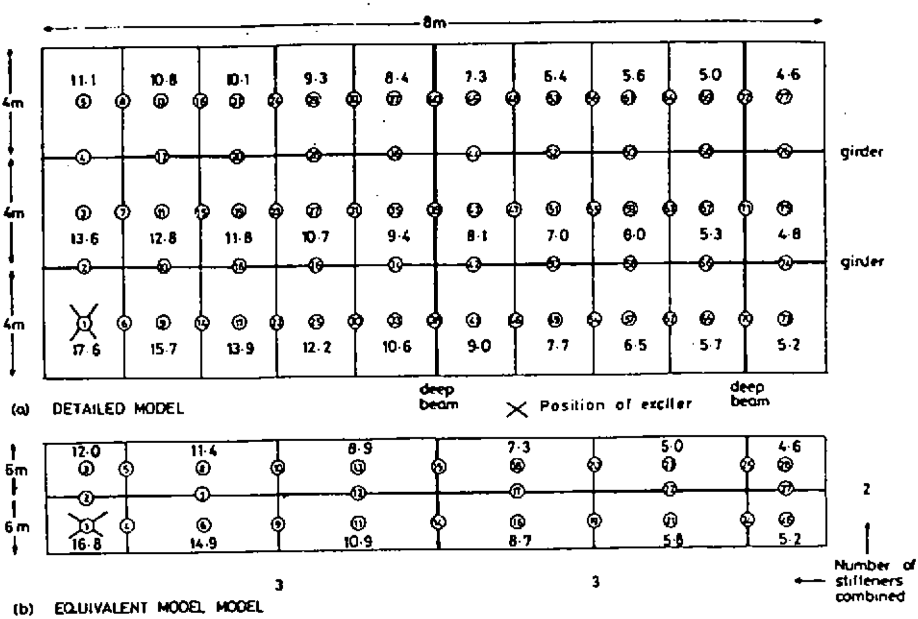

Fig. 2 Vibration Velocity at 1000Hz for Engine Room Deck Flat of Bulk Carrier

Concluding Remarks

Both the experimental work and the calculations indicate that the SEA methods may have a useful role to play in modelling high frequency vibration of a ship's structure. However, there are some areas which need further attention. The work described above and in much of the published literature considers only flexural vibration. It is suggested that realistic modelling of the structure in three dimensions requires the inclusion of longitudinal waves since they tend to be propagated in the many frames and pillars, acting as "short circuits" in the energy flow. To include longitudinal waves requires the formulation of appropriate expressions for the major parameters and introduces a complication in the choice of subsystems. What may be a suitable sub-system for flexural waves may not be a satisfactory choice for longitudinal waves. This difficulty together with other related problems, e.g. the acquisition of reliable data on mechanical loss factors and the estimation of the effective input power from main engine or propeller, need to be overcome by further theoretical and experimental work.