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Review of Methods for Structure Born Noise Prediction on Ships

Review paper

The aim of the paper is to review practical methods for structure born noise propagation assessment in complex ship structures. Three methods are reviewed: the Finite Element Method (FEM), the Statistical Energy Analysis (SEA) and the Energy Finite Element Method (EFEM). A brief theoretical background of the methods is provided together with inherent assumptions and limitations. Methods and tools are compared from different aspects in order to sort out the approach best suited for practical application on large cruise ships.

Keywords: *energy finite element method, finite element method, statistical energy analysis, structure born noise*

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Pregled metoda za analizu širenja buke kroz brodsku konstrukciju

Pregledni rad

Cilj rada je pregled praktičnih metoda za anlizu strukturne buke u složenim brodskim konstrukcijama. Obrađene su tri metode: metoda konačnih elementata, metoda statističke analize energije i metoda energetskih konačnih elemenata. Ukratko je dan pregled teorijskih osnova ovih metoda zajedno s "ugrađenim" pretpostavkama i ograničenjima. Metode su uspoređene s različitih aspekata s ciljem iznalaženja metode koja bi bila najpogodnija za praktičnu primjenu analize strukturne buke velikih putničkih brodova za kružna putovanja.

Ključne riječi: *metoda energetskih konačnih elemenata, metoda konačnih elemenata, metoda statističke analize energije, strukturna buka*

1 Introduction

The acoustic discomfort has many adverse effects on people as to produce sleep disturbance and irritation. Therefore, the prediction of noise levels on ships is a growing concern of ship owners. This is particularly true for passenger ships, as the acoustic criterion is the most important comfort criterion for cruise ship customers [1]. Other merchant ships are also frequently encountering noise and vibration problems as structural acoustic effects and noise are often omitted as design criteria [2].

There are a number of sources of vibration and noise present in ships [3]. Some typical sources are engines, shaft-line dynamics, propeller radiated pressures and bearing forces, manoeuvring devices such as transverse propulsion unit, air conditioning systems etc. When generated, the sound in a ship propagates in various ways. Airborne sound radiated by a source may be transmitted through walls, bulkheads and decks. At low frequency this transmission occurs as a result of membrane vibration of the structure, but at high frequency it has wave character. Furthermore, sonic vibration may be transmitted through foundation and hull structures, with subsequent radiation of airborne sound in the neighbouring and in remote compartments. In the case of machinery in which the vibration energy is produced in the form of sonic vibration (pumps, compressors, diesel engine), noise in neighbouring and remote compartments occurs mainly due to latter type of sound transmission. This is especially pronounced in the case when machinery is mounted on relatively light foundations in compartments with good airborne noise isolation. Appearance of noise in ship compartments remote from the source of vibration almost always may be explained by transmission of sonic vibration through the structure of the hull. The phenomenon is called structure born noise and is the main concern of the present paper.

Common ship vibration analyses are concerned with main engine and propeller induced vibration with the excitation frequencies that do not reach high values, being around 5 to 10 Hz in the average. Structure born noise actually represents high frequency vibratory structural response at frequencies above 1000 Hz. Intuitively, engineers are prone to consider noise as an extension of the low-frequency vibration and try to analyse noise propagation using the same methods. In that respect, the finite element method (FEM) is the traditional choice. However, prediction of structure born noise propagation in large ship structures by extending traditional low-frequency FEM vibration analysis to high frequencies is not straightforward. The reason is that the size of the finite elements for noise propagation assessment should be much smaller, which provokes many unwanted effects and complicates the analysis. A brief description of the standard procedure for vibration assessment of ships is presented and differences compared to noise propagation analysis are pointed out in the next section of the paper.

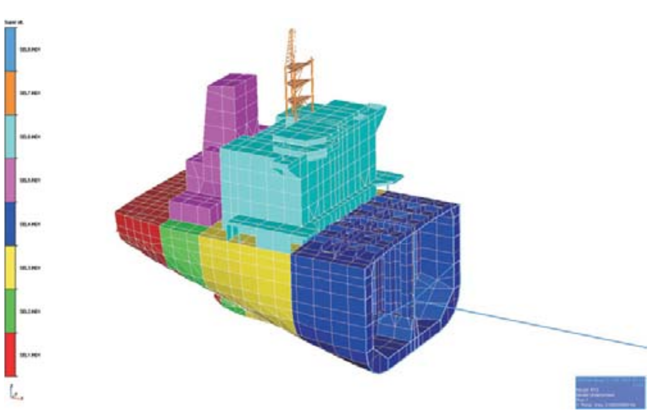
Other methods for structure born noise predictions, which are based on the vibration energy propagation, i.e. statistical energy method (SEA) and energy finite element method (EFEM), are reviewed in the following sections of the paper. These methods have been developed for the needs of aeronautical, aerospace and marine industry in past decades and some commercial software tools for their implementation are available on the market. The aim of this review is to find out which one of these methods is the most perspective for future implementation on large ships, especially cruisers, considering simultaneously several criteria such as the cost and complexity of the implementation of the method, its accuracy, feedback from experience in application to ship structures etc. Therefore, the intention of this review is to briefly summarize basic features of methods and to provide recommendations for further research. Although findings from this report are applicable to merchant ships, the principal aim of the authors was to focus on large cruise ships, being the most complex type of ship structures and also the most important regarding acoustic comfort.

2 Finite element method

Ordinary (low frequency) vibration analysis is generally solved by the finite element method (FEM). Such analysis is well established, referenced in the literature and has long tradition [4]. Vibration analysis of ship structures may be divided in the analysis of global and the analysis of local vibration.

The purpose of the global vibration assessment is to analyze behaviour of the ship structure as a whole and global vibration response of ship sub-structures (e.g. deckhouse). “Coarse” finite element mesh is applicable for global vibration with the typical size of finite elements from 1 to 4 m. Such FE mesh is such to model accurately only primary structural members. In other words, finite elements extend from one web frame to another in longitudinal direction, or from one longitudinal girder to another one in the transverse direction. Secondary longitudinal or transversal stiffeners are either included in the “coarse” finite elements (stiffened panel elements) or simply grouped along lines between plate element boundaries. In case of ships with well defined strength deck (oil tankers and bulk carriers), ship hull in cargo hold area may be considered as a beam, and only aft part of the ship is modelled with the 3D finite elements. However, for ships with long superstructures, such as passenger ships and RO-RO ships, such simplification is not allowed. Analysis of global vibration is important since in the case of excessive vibration levels, significant and expensive reinforcements would be required. It should be emphasized that a very similar FE model is applicable for strength assessment of passenger and RO-RO ships. That is a very important fact to notice, as the modelling of the whole ship with FE is a quite expensive and time consuming task, and it is therefore of great interest to be able to solve two tasks with the same FE model. For the purpose of illustration, a FE model of the aft part of an oil product tanker is presented in Figure 1.

Figure 1 **Finite element model of the aft part of an oil product tanker**
Slika 1 **Model konačnih elemenata krmenog dijela broda za prijevoz naftnih prerađevina**



Hull stiffness parameters, i.e. cross-section area, shear area and moment of inertia of the cross-section are determined by in-house program STIFF, based on strip theory [5]. Ship hull stiffness parameters are then applied to 1D beam model. The added mass and propeller excitation are calculated separately by dedicated methods [5], [6]. The 1D beam model is located at the level of the neutral line. The 3D model plating is represented by the shell elements. The web frames and deck girders are modelled mostly by the eccentric beam elements. Aft peak, stern, engine room, slop tank with part of the aft cargo tank, chimney, superstructure and radar mast are modelled.

Local vibration analysis is intended for detailed analysis of ship sub-structures, as decks or part of decks, parts of ship superstructure, mast or similar structures. Usually, the purpose of the local vibration analysis is to identify resonances of the substructures. In that case, only concerned substructure may be analyzed. Much finer finite element mesh, compared to the global vibration analysis, is to be used. Generally, wave length of the highest frequency should be divided in about 10 finite elements in order to accurately describe natural modes. To illustrate the problem, a very fine mesh model of the stiffened panel is created using only shell elements and is presented in Figure 2. The mesh size was about 50x50 mm, so the model consists of 11520 shell finite elements.

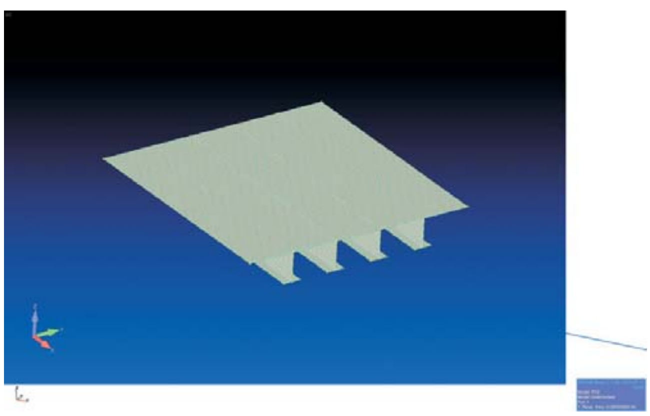


Figure 2 **Stiffened panel model**
Slika 2 **Model ukrepljenog panela**

Typical high frequency mode shape from free vibration analysis is presented in Figure 3.

It should be noted that even 1000 modes are detected in the range of frequencies 1000 to 1544 Hz. Furthermore, 300 modes are detected in the new analysis performed for the frequency range from 2000 to 2100 Hz.

The principal difference between noise propagation and local vibration analysis is that in the former case whole path from

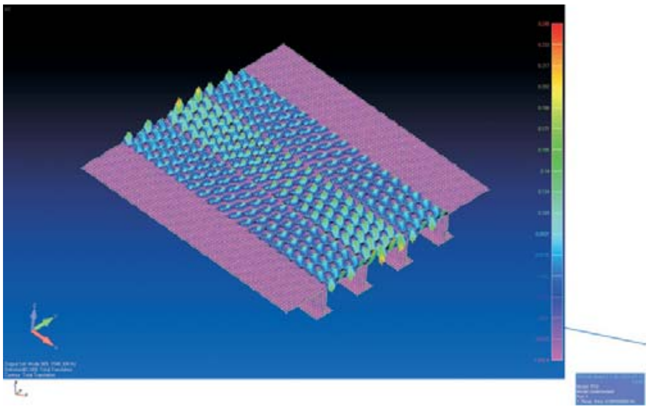


Figure 3 **Mode shape corresponding to the frequency of 1544 Hz**
Slika 3 **Oblik vibriranja za frekvenciju 1544 Hz**

the noise source to the point where noise is assessed needs to be modelled by the very fine mesh. This is necessary in order to model flow of the vibration energy through the structure. As there are usually different paths through which noise could propagate, it may eventually happen that the whole ship has to be modelled by such very fine mesh. Obviously, such analysis would cause difficulties that may not be handled easily.

To roughly illustrate the problem, let us consider equation for bending wave propagation velocity in the plate (in cm/s) [7]:

$$c_{tplate} = 950\sqrt{fh} \tag{1}$$

while associated wave length (in cm) then reads

$$\lambda_{tplate} = \frac{c_{tplate}}{f}. \tag{2}$$

Typical frequency for noise assessment is 1000 Hz, and typical plate thickness built in steel ship is about 1 cm. Then from the Eqs. (1) and (2) above, one may calculate typical length of bending waves to be 30 cm. Since about 10 finite elements are required per length of the wave, we find out that the size of the finite elements to model propagation of such a wave would need to be 30x30 mm. If that value is compared to 1 to 4 m element size for "coarse" mesh strength and vibration analysis of most ships, we may notice that the significant refinement would be necessary. Such large refinement would cause a lot of difficulties related to the creating, testing and solving FE model as number of DOFs would increase significantly. These difficulties are not acceptable and justified as the quality of the noise prediction results depends anyway on the uncertain input parameters, e.g. excitation and damping.

The application of the refined FE model to predict high-frequency vibration levels in a structure might be described as the "brute force" approach. The approach has the following obvious and serious limitations [8]:

- the appearance of matrix equations with more than million DOFs (even several millions of DOFs might be expected for large ship structures);
- modal superposition is the method best suited when the lowest eigenmodes of the structure are excited. However, in noise propagation problems, excitation frequencies are not among the lowest eigenvalues of a large complex structure making modal superposition an impractical numerical tool;
- due to the very fine mesh, modal density at high frequency of a complex thin-walled structure will be extremely high making further difficulties in the application of the modal superposition;
- governing high modes of the vibration will be very sensitive to the modelling details, such as modelling of small brackets, secondary stiffeners, type of modelling of beam elements (beams or shells) etc.
- in the modelling phase, such a fine mesh will be needed only for noise propagation assessment. Therefore, it would probably be very impractical for design engineers to make such complex modelling, with all mentioned consequences, having in mind large uncertainties of input parameters and uncertain outcomes.
- "Brute force" approach provides much more information than normally required for noise assessment. What we are normally interested in is only the distribution of the vibration energy through the structure. What one gets from the conventional vibration analysis is detailed distribution of the amplitudes in the whole structure with huge number of DOFs.

3 Statistical energy analysis

The basic idea of statistical energy analysis (SEA) is to divide a complex structure into a number of coupled subsystems and model the energy flow between them in the spirit of the transport theory. Energy balance equations are then set up for these subsystems in terms of their spatially averaged vibration levels, the rate of energy dissipation, the rate of energy exchange and the rate of energy input due to external forces.

SEA has long tradition, as the earliest works in the development of SEA done by Lyon et al. date back to the 1960s [9], [10], [11]. Following them, a number of references appeared in order to contribute to the improvement of the theory, like [12], [13], [14], [15]. Commercial software tools based on SEA are available, including those specialized for ship acoustics, e.g. [16].

Most of the basic ideas of SEA are derived from the study of two coupled sub-systems. Conclusions made for such systems are then generalized for cases that are more complex.

Consider coupled subsystems and their energy flow, as indicated in Figure 4. Each subsystem may be driven, both dissipate energy, and there exist a conservative interchange of energy between them. Considered in a long term time, the averages of these energies are related as:

$$\langle \Pi_{1diss} \rangle = \langle \Pi_{1IN} \rangle - \langle \Pi_{12} \rangle \tag{3}$$

and

$$\langle \Pi_{2diss} \rangle = \langle \Pi_{2IN} \rangle - \langle \Pi_{21} \rangle. \tag{4}$$

where angle brackets denote long-term time averages of dissipated, $\Pi_{i\,dises}$, input, $\Pi_{i\,IN}$, and exchanged, Π_{12} and Π_{21}, energy flow of the i-th subsystem. E_{itot} represents total amount of energy (kinetic and potential) contained within one system. The exchanged energy flow is related as $\langle \Pi_{12} \rangle = -\langle \Pi_{21} \rangle$, as long as the assumption of conservative coupling between subsystems is valid.

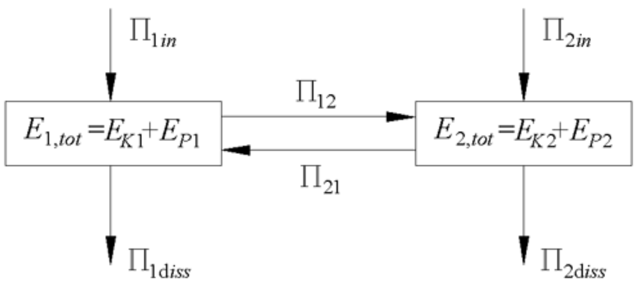


Figure 4 **Energy sharing by two coupled systems**
Slika 4 **Prijenos energije između dva spregnuta sustava**

According to the analogy with heat flow, one can conclude that the amount of energy flowing from one system to another is proportional to the relevant potential differences.

In the case of conservative energy exchange between two subsystem, Eqs. (3) and (4) express the relationship between input, shared and dissipated energy. For the case of more than two coupled subsystems, these equations can be generalized as:

$$\Pi_{i,in} = \Pi_{i,diss} + \sum_{j=1}^{N} \Pi_{ij}. \tag{5}$$

Dissipated and shared energy, can be expressed as:

$$\begin{aligned} \Pi_{i,diss} &= \omega\eta_i E_{i,tot} \\ \Pi_{ij} &= \omega\eta_{ij} E_{i,tot} - \omega\eta_{ji} E_{j,tot}, \end{aligned} \tag{6}$$

where ω is frequency and η_i and η_{ij} are dissipation and coupling loss factors respectively. Inserting these equations into Eq. (5) yields:

$$\begin{bmatrix} \eta_{1,tot} & -\eta_{21} & -\eta_{31} & \dots & -\eta_{N1} \\ -\eta_{12} & \eta_{2,tot} & -\eta_{32} & \dots & -\eta_{N2} \\ -\eta_{13} & -\eta_{23} & \eta_{3,tot} & \dots & -\eta_{N3} \\ \vdots & \vdots & \vdots & & \vdots \\ -\eta_{1N} & -\eta_{2N} & -\eta_{3N} & \dots & \eta_{N,tot} \end{bmatrix} \begin{Bmatrix} E_{1,tot} \\ E_{2,tot} \\ E_{3,tot} \\ \vdots \\ E_{N,tot} \end{Bmatrix} = \begin{Bmatrix} \Pi_{1,in}/\omega \\ \Pi_{2,in}/\omega \\ \Pi_{3,in}/\omega \\ \vdots \\ \Pi_{N,in}/\omega \end{Bmatrix}, \tag{7}$$

where

$$\eta_{i,tot} = \eta_i + \sum_{j=1}^{N} \eta_{ij}. \tag{8}$$

To solve Eq.(7), coupling loss factors η_{ij} and power input have to be estimated, while dissipation factors, η_i, are often taken as deterministic and not varying with frequency. When solved, Eq. (7) yields the total energy level of each subsystem, based on what one can determine root-mean-square velocity, acceleration, displacement, strain or stress. Basically, determination of coupling loss factors η_{ij} is the major challenge in the application of SEA.

It is worth to summarize the main assumptions and consequent limitations of SEA.

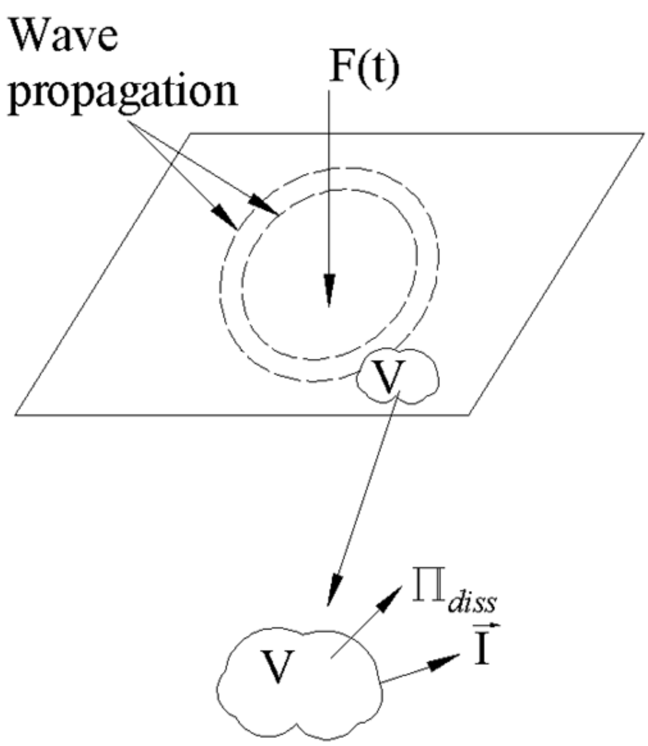
1. Subsystems and coupling mechanisms are assumed linear.
2. Only statistically independent, stationary forcing is applied (ergodic forcing).
3. The coupling between subsystems is conservative and weak.
4. Modes are statistically independent within subsystem so that all modes are equally excited by forcing.
5. Mode shapes near the coupling do not affect average energy flows.
6. Natural frequencies are uniformly distributed within given frequency band.
7. The frequency band containing natural frequencies is narrow with respect to the frequencies themselves.
8. The damping is light.
9. Each subsystem is characterized by single energy value.

4 Energy finite element method

The Energy Finite Element Method (EFEM) is a new approach for simulating high frequency vibration of large-scale structures. It is based on deriving governing differential equations with respect to energy density variables, and utilizing a FEM for solving them numerically.

The main advantage of the EFEM is the potential of modelling the ship structure by relatively coarse mesh of finite elements. Ideally, the geometrical mesh would be the same as for other structural analyses performed during ship structural design phase, i.e. strength assessment using FEM and low-frequency vibration FE analysis. Such an approach would make EFEM cost-effective

Figure 5 **Vibrating plate example illustrating control volume *V***
Slika 5 **Predodžba kontrolnog volumena za slučaj vibriranja ploče**



solution for noise propagation in complex ship structures. All other benefits of standard FEM may also be used, as easy implementation of different excitation sources, simple accounting for variation of vibration parameters within the structure (e.g. damping) as well as using standard FE post-processing capabilities for graphical representation of results.

EFEM was developed and first applied to coupled beams by Nefske and Sung [17]. Instead of characterizing each subsystem by a single energy value (as in SEA), EFEM is capable of analysing dissipation and conduction of vibration energy within each subsystem. Energy flow analysis is formulated in a form of differential equation of the heat conduction type which can be solved by applying a FEM.

In EFEM, an elemental control volume *V* of a physical system, as vibrating plate, is considered and shown in Figure 5.

In Figure 5, *F(t)* is the time varying force inputting power to the dynamic system, Π_{diss} is the internal power dissipated per unit volume, while $\vec{I}$ represent energy flux density crossing the boundary. From general theorem of mechanics, the averaged kinetic energy is equal to the averaged potential energy for conservative or slightly unconservative system, as considered here. The total vibration energy density *e* is thus twice the kinetic energy density, i.e.

$$e = \rho v^2 \tag{9}$$

where *v* is the average particle velocity, while ρ is the mass per unit volume.

If conservation of energy is applied to control volume *V*, the following expression is obtained:

$$\left\langle \underline{\Pi}_{in} \right\rangle = \left\langle \underline{\Pi}_{diss} \right\rangle + \nabla \left\langle \underline{\vec{I}} \right\rangle \tag{10}$$

where $\left\langle \underline{\Pi}_{in} \right\rangle$ corresponds to the input power density, $\left\langle \underline{\Pi}_{diss} \right\rangle$ dissipated power density and $\underline{\vec{I}}$ is the energy flux density, i.e. intensity, while "_" and "⟨⟩" indicates space averaging over a wavelength and time averaging over a wave period respectively.

Dissipated power density $\left\langle \underline{\Pi}_{diss} \right\rangle$ is assumed to be in the following form:

$$\left\langle \underline{\Pi}_{diss} \right\rangle = \eta\omega \left\langle \underline{e} \right\rangle \tag{11}$$

where η is a constant loss factor (=2ξ, where ξ is the critical damping ratio), while ω is the vibration frequency in rad/s.

Nefske and Sung [17] derived the following relationship between $\left\langle \underline{\vec{I}} \right\rangle$ and $\left\langle \underline{e} \right\rangle$:

$$\left\langle \underline{\vec{I}} \right\rangle = -\frac{c_g^2}{\eta\omega} \nabla \left\langle \underline{e} \right\rangle \tag{12}$$

where c_g is a group velocity of propagating waves. By inserting Eqs. (11) and (12) into (10), the governing differential equation for the space and time average energy density reads:

$$-\frac{c_g^2}{\eta\omega} \nabla^2 \left\langle \underline{e} \right\rangle + \eta\omega \left\langle \underline{e} \right\rangle = \left\langle \underline{\Pi}_{in} \right\rangle \tag{13}$$

The weak formulation may be employed to Eq. (13) to solve it numerically by a FEM. Formulation and matrices are derived and the following system of linear equations is produced [18]:

$$\left[K^e\right]\left\{e^e\right\} = \left\{F^e\right\} + \left\{Q^e\right\} \tag{14}$$

where $\{e^e\}$ is the vector of nodal values for the time and space average energy density for a finite element, $[K^e]$ is the system matrix for each finite element, $\{F^e\}$ is the excitation vector representing the energy input at each node of the finite element and $\{Q^e\}$ is the power flow across the element boundary. The term $\{Q^e\}$ provides the mechanism for connecting elements across discontinuities. Formulation presented by Vlahopoulos et al. [18] enables representation of $\{Q^e\}$ in term of matrices representing the power transfer mechanism across the joint and nodal values of energy densities. Thus, $\{Q^e\}$ in Eq. (14) may be included in the left hand side of the equation as part of $[K^e]$. Such procedure is not the part of the conventional FEM and consequently specialized module needs to be programmed to add effects of joints in assembly of FE matrices. Once assembly is performed, solution to the global FE system of equations results in distribution of the energy density over the entire system. After that, vibration velocities may be calculated by Eq. (9).

Practical application of EFEM to ship structures is described by several authors. Ship classification society *Germanischer Lloyd* (GL) reported development of the Noise-FEM package capable to forecast the propagation of structure-borne noise in complex ship structures [19]. Numerical implementation, validation and application of the EFEM to fishing boat structure is described by Vlahopoulos et al. [18]. Commercial software based on EFEM suitable for the analysis of high frequency vibroacoustic problems in aerospace, automotive, naval and other industries is available [20].

The principal reference source about research in progress related to ship structures is provided by the International Ship and Offshore Structures Congress (ISSC). In the last report of ISSC 2009 [21], EFEM is emphasised as a recently developed method for the prediction of the vibration behaviour of structures in the mid- and high-frequency range. The ISSC 2009 considers the spatial distribution of the energy density as a significant advantage of EFEM comparing to SEA methods, which gives only an average value of energy density for a subsystem. The ISSC 2009 also mentioned another known advantage of EFEM: that FE models of the structure can be used for EFEM analysis, for instance the same model as used for the strength analysis of the vessel. As the most important drawback of the EFEM, the ISSC 2009 mentioned the effects of junctions between structural components, which is difficult to model because the power transmitted by the bending, the longitudinal and the shear waves has to be considered and redistributed to the adjacent structural components in a realistic way. It is also emphasised that calculation of coupling loss factors (CLF) is more challenging comparing to CLF in SEA analysis.

5 Conclusions

For reasons elaborated in Section 2 of the paper, energy methods are preferred for the analysis of the noise propagation problems compared to the conventional finite element method. Two energy methods have been used by the industry and they are briefly described in this paper: Statistical Energy Analysis (SEA) and Energy Finite Element Method (EFEM).

SEA has long tradition and has been developing since the 1960s. Therefore, there is a long engineering experience with the application of the method. That is an important aspect regarding appropriate definition of input parameters, such as coupling loss factors. The main limitations of SEA are that vibration energy levels are assumed constant within subsystems and that it is not possible to establish a direct link between SEA model and FE models used for ships strength and vibration analysis.

EFEM is a new method developed within past two decades for the needs of aerospace and marine industry. The method improves two mentioned drawbacks of the SEA, i.e. EFEM is capable of providing energy distribution within subsystems and existing FE model may be used with some additional inputs. Drawback of the EFEM is that it represents a relatively new method with not so much feedback from experience with application to the ship structures. Therefore, additional researches and comparisons with experiments are necessary to get confidence in the EFEM results. However, this method represents emerging technology for the analysis of the noise propagation in complex structures.

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Disclaimer

The opinions presented herein are those of the authors and should not be construed as reflecting the views of any company or institution.

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