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Study of Noise Propagation for Small Vessels

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The paper presents the results of the noise propagation analysis in ship structures tested in a number of AHTS (Anchor Handling Tug Supply) vessels. Statistical Energy Analysis (SEA) based on numerical model developed specially for the purpose of this numerical investigation were conducted. This numerical model enabled the analysis of both the structural elements and the acoustic spaces. For the detailed studies 47 points fixed at various ship locations were selected. Prediction results with use of the numerical model were compared with the experimental results carried out in six identical AHTS vessels. Experimental studies were performed in accordance with the requirements of the International Maritime Organization (IMO) Resolution A.468 (XII). As a result one presented a comparison of the model analysis and experimental tests results.

Keywords: ship noise, noise analysis, statistical energy analysis.

1. Introduction

Noise propagation inside a relatively small vessel is still an issue both for the shipyards and the naval architects. Therefore a calculation and measurements comparison analysis of the AHTS vessels was performed. Statistical Energy Analysis (SEA), numerical analyses based on numerical model developed specially for the purpose of this analysis, were conducted. This numerical model enabled the analysis of both the structural elements and the acoustic spaces. For the detailed studies 47 points fixed at various ship locations were selected. Prediction results with use of the numerical model were compared with the experimental results carried out in six identical AHTS vessels (Weryk, 2013). Numerical investigations presented in this paper don't cover the underwater noise propagation phenomena which can be found, i.e., in (Kozaczka *et al.*, 2007; 2011; Grelowska *et al.*, 2013).

2. Numerical model

In order to predict noise in a such complex environment as the ship structure, one has to choose a proper analysis method. For this purpose SEA was chosen – Fig. 1.

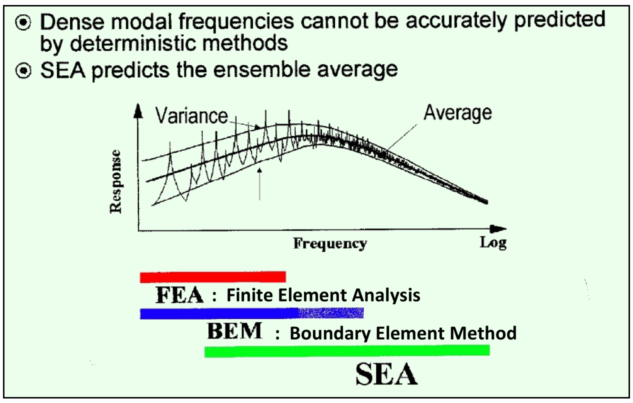

Fig. 1. Comparison of noise propagation methods (AutoSEA2 User's guide, 1999).

Knowing SEA limitation one had to develop own method for modelling an AHTS vessel. SEA method is based on an assumption that at a given frequency there is a minimum modal density. In every frequency band there should be at least 5 vibration modes and all modes are energetically equal. Energy can be transferred from subsystem with higher modal energy to the subsystem with lower modal energy. In order to calculate transferred energy one has to build a model composed of acoustic cavities, plates etc. SEA is a method where one can distinguish certain steps:

- structure division for subsystem determination,
- applying excitation (input power sources),
- SEA parameters estimation (modal density, coupling loss factor, damping loss factor),
- energy transferred calculation,
- resulting response levels calculation (i.e. sound pressures, vibration velocity levels).

Taking above into consideration one has built a model as presented in Fig. 2.

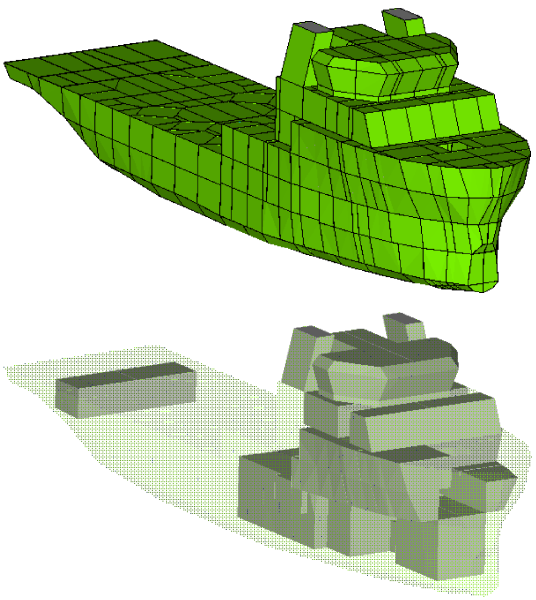

Fig. 2. SEA model of AHTS Vessel – structural (top) and acoustical (bottom).

3. Calculation method

Created model is ready for entry material properties of every subsystem and applying excitations. Below the SEA based on two elements is presented – Figs. 3, 4. In the case of AHTS this example of a typical connection between a wall and a deck.

Dissipated power $\varPi_{1,\text{diss}}$ in subsystem 1 is given by:

$$\varPi_{1,\text{diss}} = 2 \cdot \pi \cdot f \cdot \eta_1 \cdot E_1, \qquad (1)$$

where η_1 is the damping loss factor in subsystem 1 and E_1 is the subsystem 1 total vibration energy at frequency f.

Dissipated power $\varPi_{2,\text{diss}}$ in subsystem 2 is given by:

$$\varPi_{2,\text{diss}} = 2 \cdot \pi \cdot f \cdot \eta_2 \cdot E_2, \qquad (2)$$

where η_2 is the damping loss factor in subsystem 2 and E_2 is the subsystem 2 total vibration energy at frequency f.

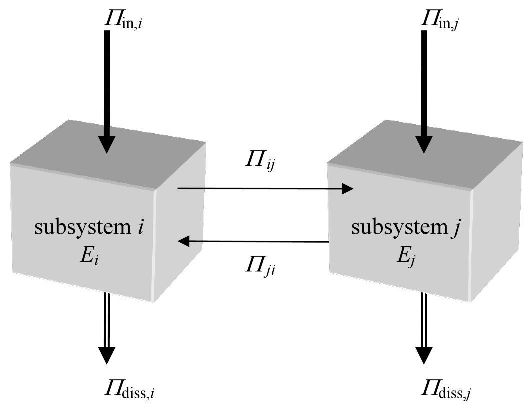

Fig. 3. Two subsystem SEA model.

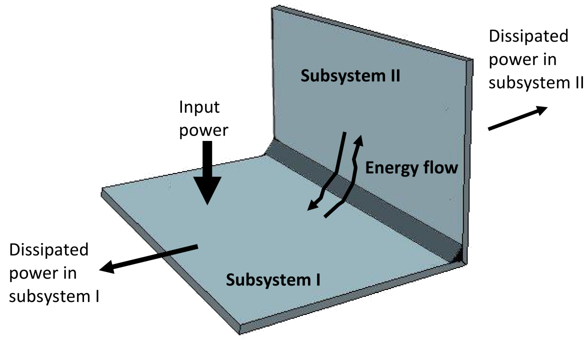

Fig. 4. Energy transfer between a deck and a wall.

Transferred power between subsystem 1 and 2 $\varPi_{1,2}$ is:

$$\varPi_{1,2} = 2 \cdot \pi \cdot f \cdot \xi_{12} \cdot E_1 - 2 \cdot \pi \cdot f \cdot \xi_{21} \cdot E_2, \qquad (3)$$

where ζ_{12} is the coupling loss factor between subsystem 1 and 2, ζ_{21} is the coupling loss factor between subsystem 2 and 1.

Connection between two subsystems is described with a use of two coupling loss factors.

Below there are energy balance equations for two subsystems:

$$\begin{aligned} \varPi_{1,\text{in}} &= \varPi_{1,\text{diss}} + \varPi_{12} = 2 \cdot \pi \cdot f\,(\eta_1 + \xi_{12}) \cdot E_1 \\ &\quad - 2 \cdot \pi \cdot f \cdot \xi_{21} \cdot E_2, \\ \varPi_{2,\text{in}} &= \varPi_{2,\text{diss}} + \varPi_{21} = -2 \cdot \pi \cdot f \cdot \xi_{12} \cdot E_1 \\ &\quad + 2 \cdot \pi \cdot f\,(\eta_2 + \xi_{21}) \cdot E_2. \end{aligned} \qquad (4)$$

After combining above equations into one the following equation is obtained:

$$\begin{Bmatrix} \varPi_{1,\text{in}} \\ \varPi_{2,\text{in}} \end{Bmatrix} = 2 \cdot \pi \cdot f \begin{bmatrix} (\eta_1 + \xi_{12}) & -\xi_{21} \\ -\xi_{12} & (\eta_2 + \xi_{21}) \end{bmatrix} \begin{Bmatrix} E_1 \\ E_2 \end{Bmatrix}. \qquad (5)$$

One of the fundamentals of the SEA method is so-called reciprocity relationship given below:

$$\Delta N_1 \xi_{12} = \Delta N_2 \xi_{21}, \qquad (6)$$

here ΔN_1 is the mode number in frequency band Δf in subsystem 1, ΔN_2 is the mode number in frequency band Δf in subsystem 2.

Having division of mode number and frequency band the modal density is given:

$$\Delta N = n \cdot \Delta f, \tag{7}$$

where n is the modal density.

Taking above into consideration, Eq. (7) can be presented in a given form:

$$n_1 \xi_{12} = n_2 \xi_{21}. \tag{8}$$

Substituting Eq. (8) to Eq. (5) one gets (Lyon, 1975):

$$\begin{Bmatrix} \Pi_{1,\mathrm{in}} \\ \Pi_{2,\mathrm{in}} \end{Bmatrix} = 2 \cdot \pi \cdot f \begin{bmatrix} (\eta_1 + \xi_{12}) \cdot n_1 & -\xi_{21} \cdot n_2 \\ -\xi_{12} \cdot n_1 & (\eta_2 + \xi_{21}) \cdot n_2 \end{bmatrix} \begin{Bmatrix} \frac{E_1}{n_1} \\ \frac{E_2}{n_2} \end{Bmatrix}. \tag{9}$$

The solution of the energy balance matrix is not an issue, provided that all coupling loss factors and input powers are known.

Energy stored in a plate one can derived from:

$$E_i = m_i \cdot S_i \cdot \langle v_i \rangle^2, \tag{10}$$

where m_i is the element unit mass, S_i is the element surface, $\langle v \rangle^2$ is the square of the mean value of the vibration velocity amplitude.

With the above formula, one can determine the vibration velocity of individual subsystems.

For a system with a given number of subsystems, the energy balance matrix will be as follows:

$$\omega \cdot [\eta] \cdot [E] = [\Pi], \tag{11}$$

where $[\eta]$ is the loss factors square matrix, $[E]$ is the stored energy in subsystems column matrix, and $[\Pi]$ is the power column matrix.

4. Noise measurements

4.1. Measurements points

Measurements points were chosen according to IMO (A.468(XII) Resolution, 1981). According to this document locations which have to be protected against excessive noise can be divided into five localization types:

- work places,
- navigation post,
- crew cabins,
- kitchens and pantry rooms,
- normally unoccupied locations.

Taking above recommendations into consideration we have chosen 47 measurements points. Those points were chosen in order to represent the general noise climate onboard. Measurements performed on all tested ships were executed at similar weather – wind (2–3°B) and sea state (1–2) conditions. Noise measurements were executed during 90% main engines loading, vessel speed was around 14 knots.

4.2. Measurements results

In Fig. 5 there are results of summary noise for all measurement points in dB(A) for all six tested ships.

For every measurement result there was octave analysis performed with minimum and maximum level marked – example in Fig. 6.

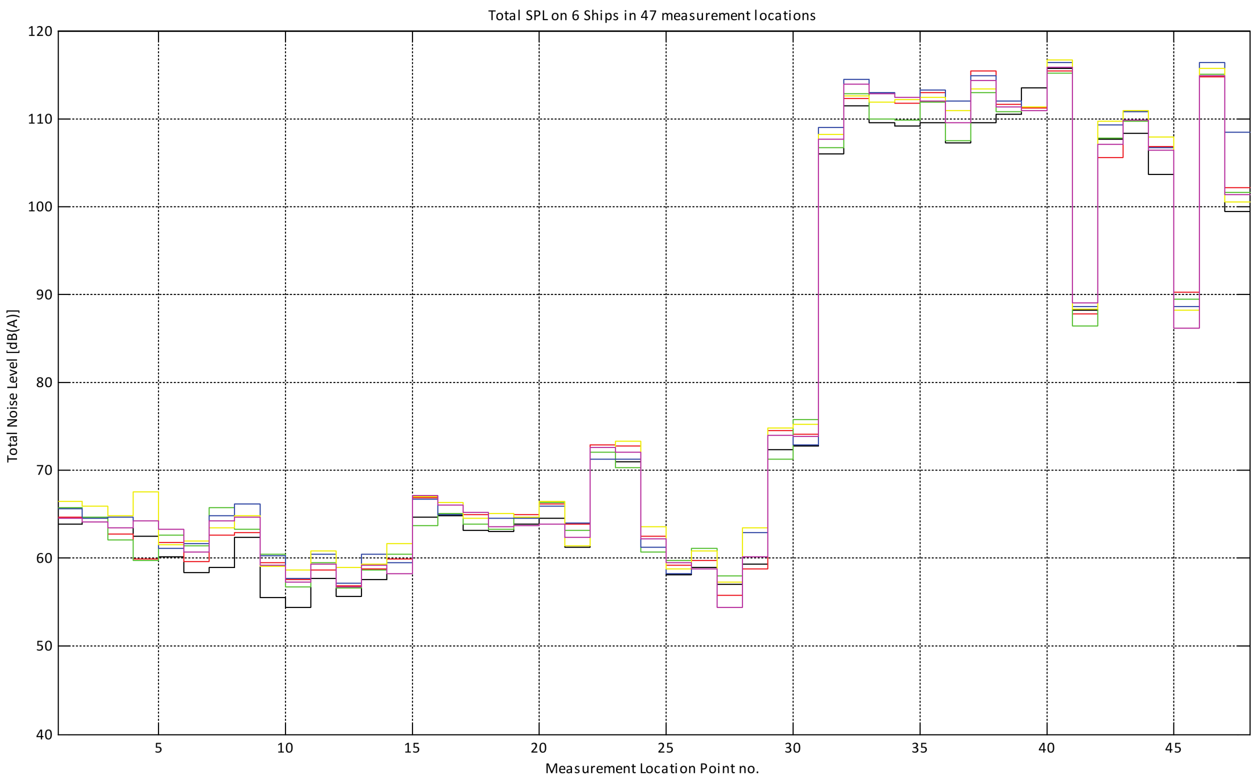

Fig. 5. Noise level in 47 measurements points on six ships.

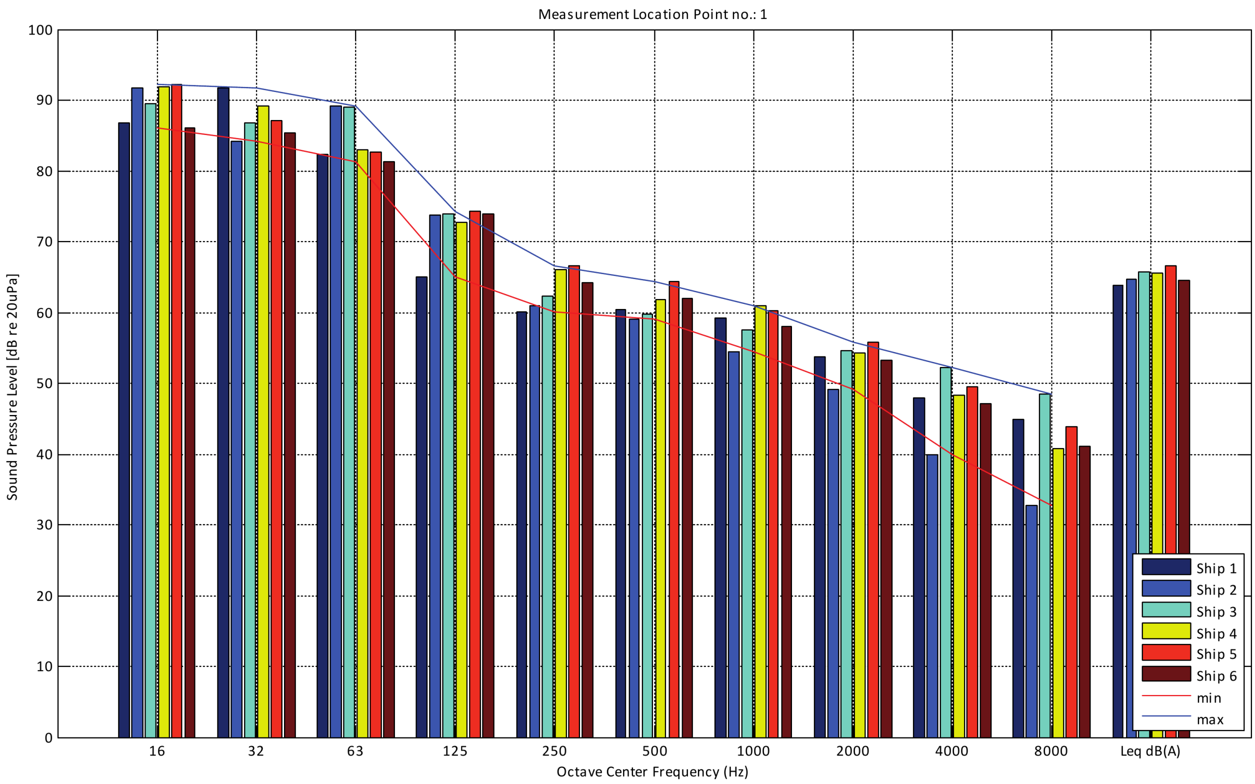

Fig. 6. Noise octave spectrum for measurement point 1.

5. Noise calculation

In order to perform noise calculation we have to possess data on characteristics of main noise sources (such as main engines, propeller (Kozaczka, 1978), exhaust, etc.) and damping loss factors for structural and cavity elements.

The noise distribution prediction model allows to determine the noise level in the selected frequency band for a given area of the ship. Example result of model calculations is shown in Figs. 7, 8. This is the final result of an analysis in the form of distribution of the noise level for the selected octave band frequency. In this case, the prediction is the result of noise in the

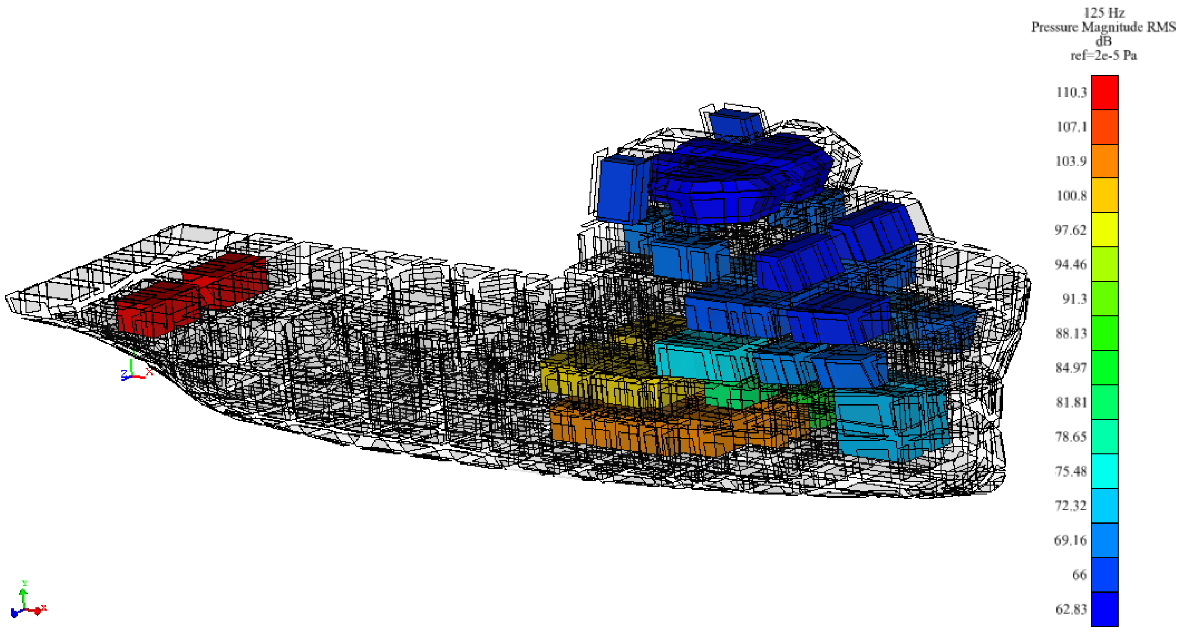

Fig. 7. Noise level in acoustic cavities for octave band with center frequency of 125 Hz.

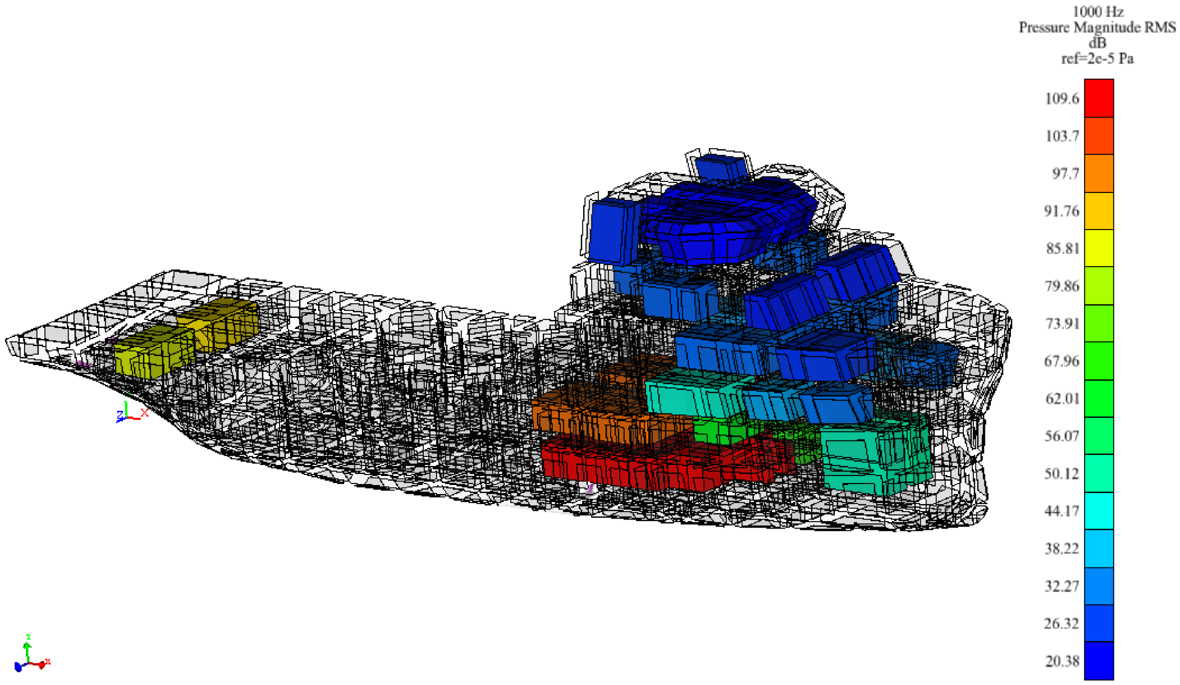

Fig. 8. Noise level in acoustic cavities for octave band with center frequency of 1000 Hz.

acoustic cavities for the selected frequency bands of 125 Hz and 1000 Hz.

On a series of 6 AHTS vessels driven with identical power propulsion and build according to the same initial technical design noise measurements have been executed. Measurements has been performed in 47 locations (crew cabins, engine room, etc) (Weryk, 2013). In Fig. 9 we have presented a comparison between calculation results and on-board noise measurements results.

SEA method assessment was performed with a use of proposed parameter:

$$\Delta L_{mS}\ \ [\text{dB(A)}]\ = |L_{\text{SEA}} - L_{eq}|\ \ (\text{dB(A)}). \tag{12}$$

ΔL_{mS} is a parameter which describes difference between a noise level calculated with SEA method and mean value of the on-board measurement noise level.

The percentage of ΔL_{mS} for all correctly modelled measurement points in range of **0–1 dB(A)** is

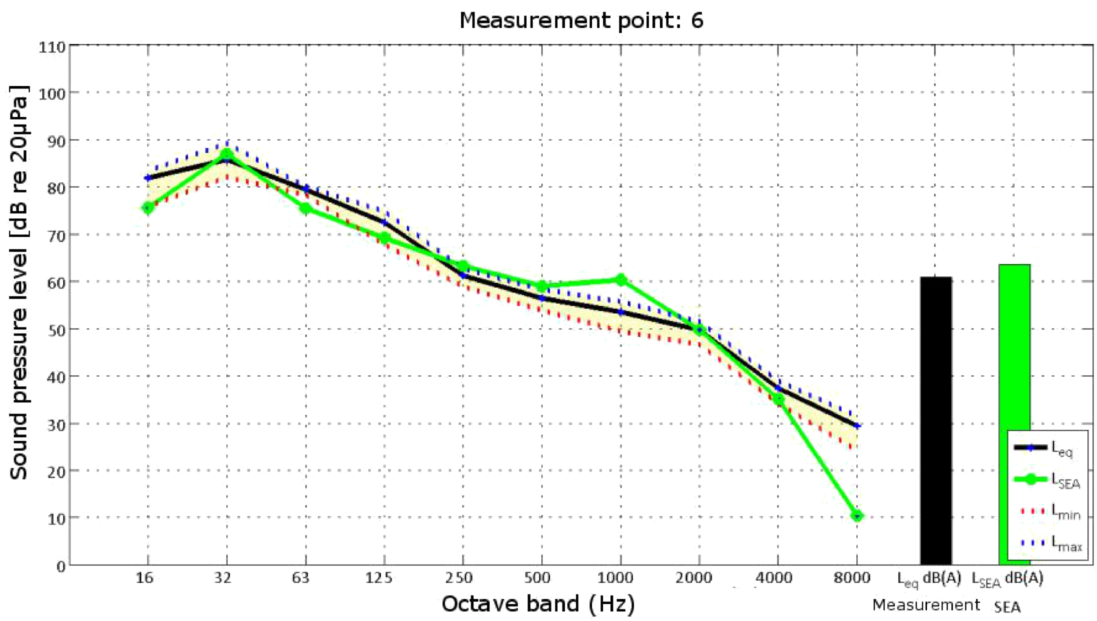

Fig. 9. Comparison between calculation results and on-board noise measurement results for measurement point no 6.

25.8%, in range of **0–3 dB(A)** is **61.3%**, in range of **0–4 dB(A)** is **74.2%**, in range of **0–5 dB(A)** is **87.1%**, in range of **0–6 dB(A)** is **93.5%**. This means that acceptable error of the method based on adopted excitation and damping parameters of vibroacoustic energy propagation in the hull of the ship is 4 dB(A) for a confidence of appr. 75%. To reduce this error we should further work to develop a model of reverse internal loss factor (i.e., to obtain the internal loss factors to achieve calculation results compliant with the results of measurements) and apply it to similar vessels.

6. Summary

The paper presents the results of the noise propagation in ship structure tested in a number of AHTS (*Anchor Handling Tug Supply*) vessels. Statistical Energy Analysis (SEA), numerical analyses based on numerical model developed were performed. This numerical model enabled the analysis of both the structural elements and the acoustic spaces. For the detailed studies 47 points fixed at various ship locations were selected. Prediction results with use of the numerical model were compared with the experimental results carried out in six identical AHTS vessels. Experimental studies were performed in accordance with the requirements of the International Maritime Organization (IMO) Resolution A.468 (XII).

The main aim of presented paper was to verify that Statistical Energy Analysis method is adequate to perform vibroacoustic analysis in ship's structure.

Based on literature analysis, Statistical Energy Analysis method was selected and used for the analysis of vibroacoustic energy in the ship structure. Based on own research and on existing studies, an AHTS numerical model (based on the Statistical Energy Analysis method) was build. Another major issue was to make a series of experimental studies on six AHTS ships equipped with the same devices such as propulsion (main noise sources). Result example of the comparative analysis and conclusions were presented.

The most important achievements among those presented in the work are listed below:

1. Development of a numerical model of vibroacoustic energy propagation in the structure of the AHTS vessel taking into account the current state of knowledge.
2. Verification of the numerical model through performance of a series of experimental studies in a number of AHTS vessels.
3. Uncertainty analysis of measurement results for six AHTS vessels and evaluation of the overall uncertainty of the measurements results.
4. The analysis of the applicability of the SEA method for prediction of vibroacoustic energy propagation in the structure of the AHTS vessel and definition of the method's clear confidence threshold level knowing the method's limitation.

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