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Application of the Energy based Finite-Element-Method to determine the sound emission of vibrating ship structures in the High Frequency Domain

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ABSTRACT

For large scale structures the applicability of classical numerical methods such as Finite-Element-Method (FEM) or Boundary Element Method (BEM) is restricted to the low and mid frequency range. The fine mesh resolution for higher frequencies rapidly leads to problem sizes which exceed given computational resources. Here, the energy based Finite Element Method (EFEM) provides an efficient solution.
The EFEM is based on balancing input, output and dissipated powers over a specific control element and is applicable to fluid and structure problems. Exchange of energy between elements is described by transmission coefficients. The unknowns of the EFEM are energy densities which aren't oscillatory in nature. Thus, the required discretization is independent of the frequency and results in a smaller system of equations compared to classical methods.
In this contribution the application of the EFEM to a ship is outlined. After a brief introduction to theoretical aspects the focus will be on the coupling of fluid and structure problems based on the radiation efficiency. In addition to determining the radiation efficiency the influence of the coupling parameters on the resulting system of equations is shown and utilized to define a lower frequency bound for the application of the EFEM to vibrating ship structures.

1. INTRODUCTION

Especially in the design and development phase of ships the prediction of underwater sound characteristics is of great interest. In the lower frequency range classical numerical methods like the Finite Element Method (FEM) are a suitable tool to predict the required physical quantities. In the higher frequency range the computational costs for classical numerical methods exceed the commonly available computer capacities. Energy based methods provide an efficient possibility to calculate acoustic quantities in the higher frequency range. A widely used energy based method is the Statistical Energy Analysis (SEA). To apply this method the whole model is divided into larger substructures before the calculation. An energy density is than assigned to each substructure as unknown value. For the application of the energy based Finite Element Method (EFEM) the substructures are further divided into finite elements, which yields the possibility to determine the local distribution of energy density on substructures. The power flow in coupling regions is therefore described more accurately due to this additional discretization.

The appropriate frequency range of applicability of the EFEM is often desired. Several approaches have been made regarding this topic (1, 2). In this contribution the influence of fluid-structure coupling on the system of equations is investigated in detail to give a lowest bounding frequency for EFEM calculations considering the solvability of the system matrix.

The correct determination of the input power is a crucial point for numerical calculations with energy based methods. Besides the vibrating systems on board of the ship, propeller noise has a large influence on the resulting sound pressure in the fluid since there is a direct transfer path into the water. The rotating propeller leads to a sound excitation with different excitation mechanisms that are explained in context of energy based calculations in the fourth chapter. A sensitivity analysis of the different excitation transfer paths is presented for the investigated model. This contribution has been produced in collaboration with the WTD 71.

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2. THEORETICAL BASICS

2.1 Basic equation of energy transport

The EFEM is based on an equation that is similar to the heat transfer equation, since the energy flow between different areas or volumes is described proportional to the gradient of energy density between these two domains. The gradient Δe is scaled depending on the group velocity of the specific wave type c_g, the structural damping constant η and the angular velocity ω of the calculated frequency. The transportation of energy is described by the first term on the right hand side of equation 1. Furthermore the dissipated energy in the area or volume is taken into account. This amount of dissipated energy flow is described by the second term on the right hand side of equation 1. Since the amount of energy is not changing in a stationary state of the system, the dissipated and transported energy sums up to the incoming energy Π_{in} for a small domain. This leads to the basic equation of the EFEM and reads

$$\Pi_{in} = \frac{c_g^2}{\eta\omega}\Delta e + \eta\omega e. \tag{1}$$

The unknowns of the system of equations are the energy densities of different wave types on each node of the discretized system (3). Basic elements in an EFEM model are beams, plates and acoustic cavities. For each basic element the energy densities of each wave type are distinguished. In the case of plates longitudinal, shear and bending waves are assumed.

2.2 Structural Coupling

At joint connections, a distinction is made between dissipated and reflected power flows of the coupled energy densities, which also depend on the wave type. These power flows are described by transmission coefficients. The analytical calculation of the transmission coefficients depends on geometrical and material properties and is done before the system matrix is set up. The transmission coefficients are collected in the transmission matrix $\hat{\boldsymbol{\tau}}$, which is further used to calculate the joint matrix $\hat{\boldsymbol{J}}$. The joint matrix $\hat{\boldsymbol{J}}$ finally describes the transmitted power flows depending on the level of energy densities in the connected domains and reads

$$\hat{\boldsymbol{J}} = (\boldsymbol{I} - \hat{\boldsymbol{\tau}})(\boldsymbol{I} + \hat{\boldsymbol{\tau}})^{-1}\boldsymbol{C}, \tag{2}$$

with the identity matrix $\boldsymbol{I}$ and a diagonal matrix $\boldsymbol{C}$, containing the group velocities of the coupled wave types. The joint matrices are added to the system of equations after setting up the element matrices. The solution is further obtained using an LU decomposition.

2.3 Fluid-Structure-Coupling

The coupling between fluid and structure domains is realized in a similar way. In this case the transmission coefficients are used to describe the power flows between the fluid and the structural domain. The transmission coefficient

$$\tau_{fluid,structure} = \frac{\beta c_0^2 \sigma}{c_B f h}, \tag{3}$$

describes the amount of energy that is transmitted from the fluid into the structure and depends on the quotient of the characteristic impedances β, the speed of sound in fluid c_0, the group velocity of the bending wave c_B, the specific calculated frequency f, the thickness of the plate h, and the radiation efficiency σ. The amount of energy that is transmitted from the structure into the fluid is described by the transmission coefficient

$$\tau_{structure,fluid} = \frac{2\beta\sigma}{2 + \beta\sigma}. \tag{4}$$

The transmission coefficients in equations 3 and 4 both depend on the radiation efficiency (4). The determination of the radiation efficiency is a crucial point and its calculation includes several properties of the coupled domain. A direct determination of the radiation efficiency is only possible with a large numerical effort. A direct calculation of the radiation efficiency using the FEM is only suitable in the lower frequency range. The determination in the high frequency range for a complete ship structure is not possible due to insufficient numerical capacities. Fortunately, analytical approaches exist that can be used to determine the desired value based on the equations of motion of basic fluid-structure coupled configurations. In this contribution the analytical approach of Rumerman is used (5). Here, the equations of motion of a simply supported, fluid loaded plate are used to give an estimation of the radiation efficiency. The radiation efficiency depends on the edge width of the coupled structure. Therefore, an algorithm has been implemented to detect the needed geometrical properties of the coupled areas. One specific radiation efficiency is prescribed on each structural panel of the investigated ship.

3. COUPLING PARAMETERS

A comparison between the analytical approach of Rumerman and an FEM determined radiation efficiency is shown in Figure 1. The depicted radiation efficiency is obtained for a rectangular simply supported steel plate that is coupled to a water domain. A strong frequency dependency is observed for the radiation efficiency that is calculated with the FEM. This is due to the modal effects that influence the radiating behavior of the plate. These modal effects are neglected in the approach of Rumerman. Nevertheless there is a good averaged agreement of the analytical approach especially for higher frequencies. In the lower frequency range the radiation efficiency calculated with the analytical approach of Rumerman approaches zero, whereas the FEM calculated radiation efficiency is still in a range between 10^{-4} and $3 \cdot 10^{-3}$.

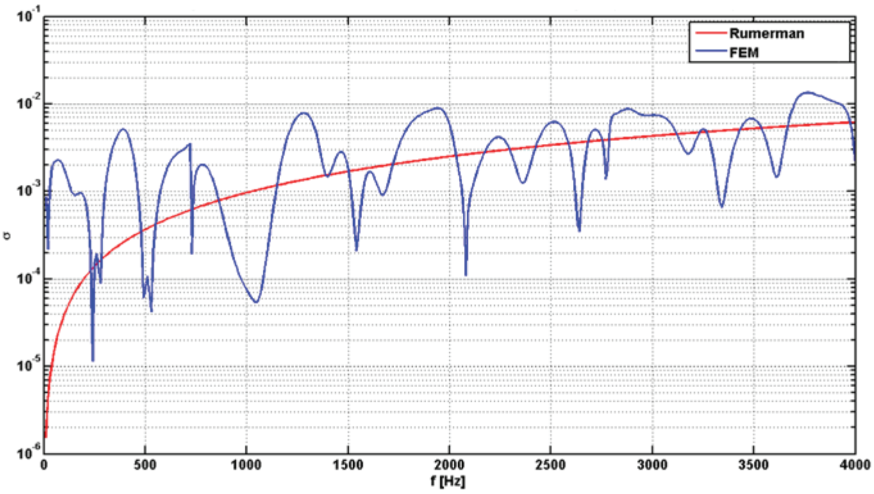

Figure 1 – Radiation efficiency determined with the approach of Rumerman and the FEM

The influence of the radiation efficiency on the transmission coefficient $\tau_{structure,fluid}$ is shown for the frequency range below 1 kHz in Figure 2. All values are in a physically reasonable range. The qualitative deviations are similar to the ones detected in the direct comparison of the radiation coefficients.

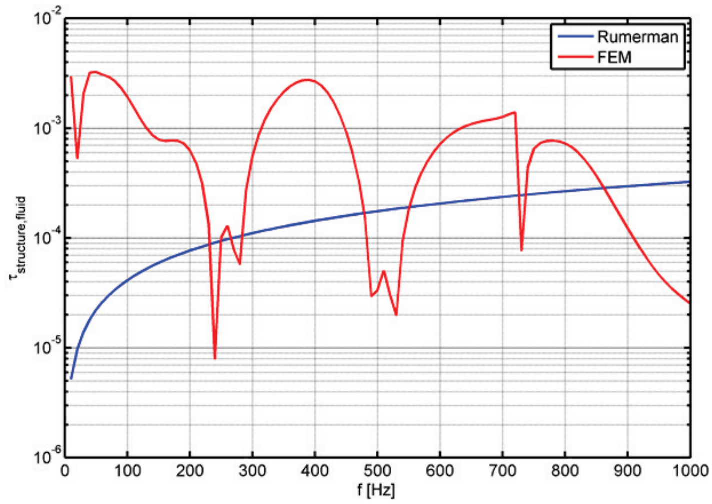

Figure 2 – Transmission coefficient for energy flow from structure to fluid

A comparison of the transmission coefficients $\tau_{fluid,structure}$ with the prescribed Rumerman radiation efficiency and the FEM calculated radiation efficiency is given in Figure 3. Here, the transmission coefficient that is calculated with the FEM radiation efficiency leads to high values in a range of up to 10^3 for low frequencies below 20 Hz.

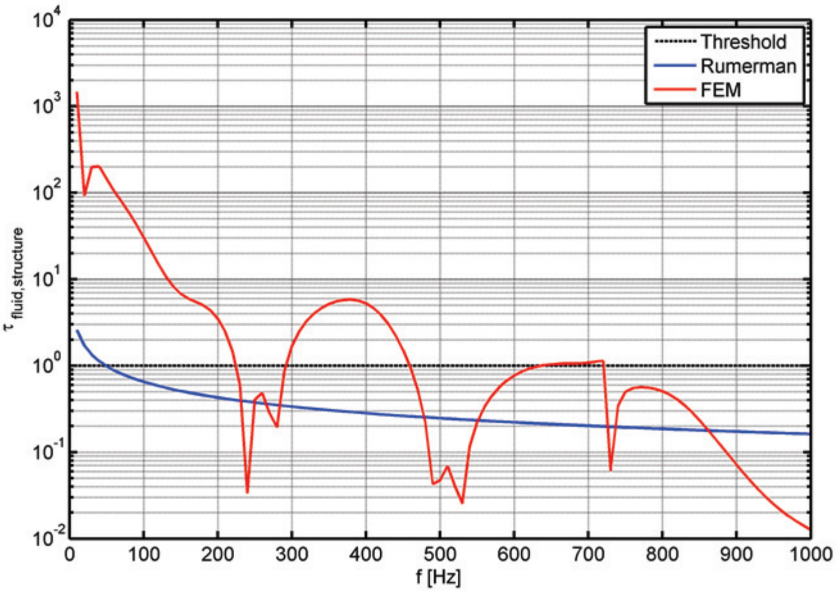

Figure 3 – Transmission coefficient for energy flow from fluid to structure

To assess the solvability of the system of equations a consideration of the largest and smallest eigenvalues of the system matrix is made. The frequency dependent behavior of the largest eigenvalue is shown in Figure 4 for system matrices that are set up for systems in different coupling conditions and with different prescribed radiation efficiencies. Besides the eigenvalues of the coupled system, the eigenvalues of the particular structural and fluid systems are shown. The largest eigenvalue of each system is independent of the prescribed radiation efficiency value. This value is determined by the structural component of the model since the largest eigenvalue of the system matrix of the particular fluid domain is clearly smaller in the whole considered frequency range.

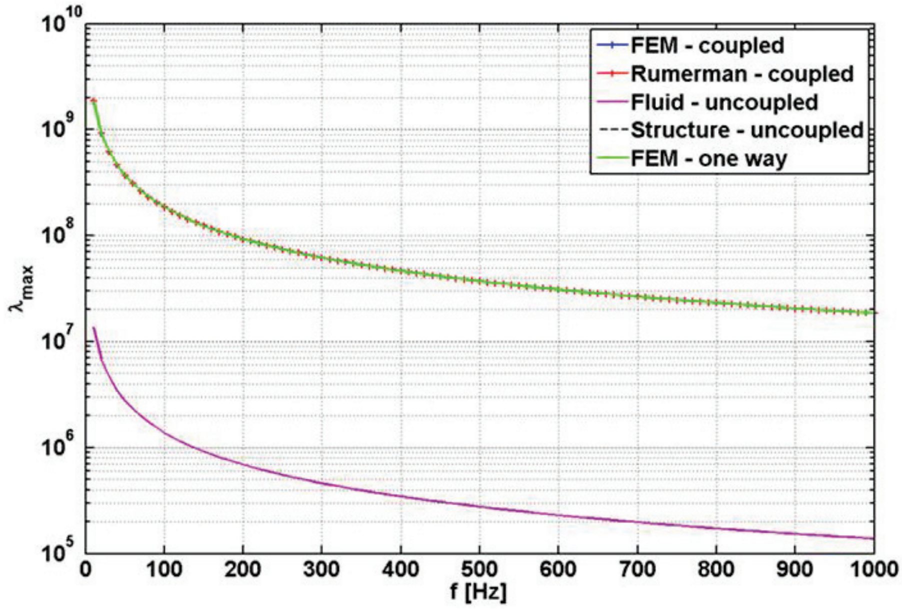

Figure 4 – Largest eigenvalue of EFEM system matrix

The smallest eigenvalue for different coupling conditions is depicted in Figure 5. All eigenvalues tend to zero for very small frequencies. This leads to a poor condition of the system of equations. The unsteady behavior of the eigenvalue of the coupled system with a prescribed FEM calculated radiation efficiency is noticeable. This behavior is not observed if the transmission coefficient $\tau_{fluid,strcture}$ is set to zero, which is depicted in the FEM – one way – curve. Therefore this drop of the lowest eigenvalue is due to the very high values of the corresponding transmission coefficient.

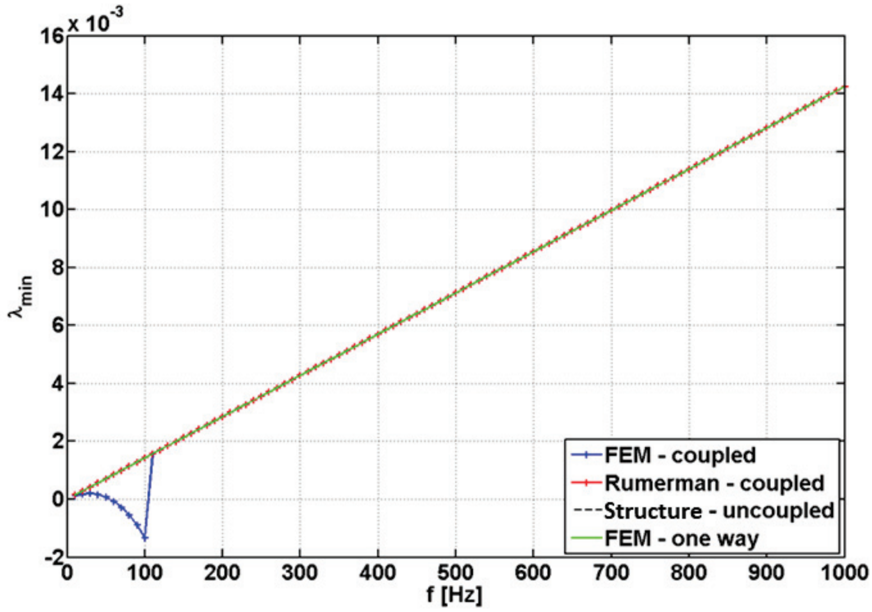

Figure 5 – Smallest eigenvalue of system matrix

This drop leads to eigenvalues that are nearly zero and therefore influences the solvability of the system of equations. For the investigated system a frequency of 100 Hz can be regarded as a lower bound for the solvability of the numerical fluid-structure coupled system of equations.

4. PREDICTION OF SHIP NOISE

The sound emission of a ship mainly results from the structure-borne sound that is transferred over the wetted surface to the surrounding fluid as well as the propeller induced pressure fluctuations. For the research ship Planet of the German Navy the structure-borne sound radiated into the water is mainly generated by the dynamic behavior of the propulsion system which comprises the drive shaft and the electric main and auxiliary engines. Due to the SWATH design of the ship the generators can be positioned on deck between the twin-hull and thus they contribute less to the generated underwater noise than the propulsion. The propeller induced sound emission results from different mechanisms. Besides the turbulence noise and the sound radiated at the blade passing frequencies, the occurring pressure fluctuations on the hull lead to vibrations of the submerged structure and thus to sound emissions induced by the propeller. Additional noise is generated if propeller cavitation occurs. The equivalent sound source exhibits a broadband spectrum which varies with the rotational speed of the propeller.

In the following example the previously outlined vibro-acoustic formulation of the EFEM is applied for a sensitivity analysis of the sound emission of the ship in the mid and high frequency range with respect to the above mentioned main contributors. According to measurements the sound emission of the ship is defined in terms of the maximum sound pressure level while crossing a hydrophone, as illustrated in *Figure 6*.

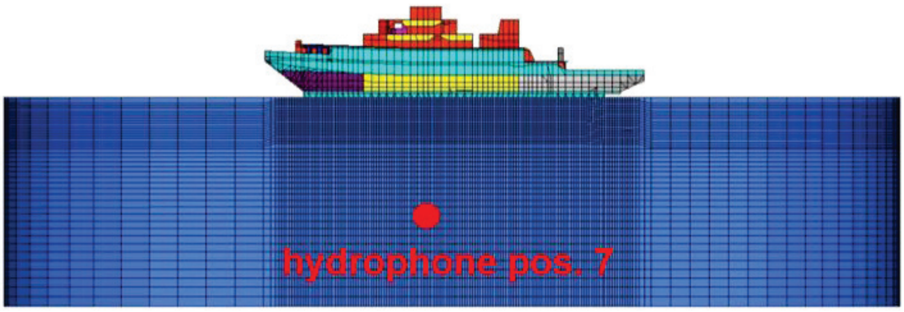

Figure 6 – Vibroacoustic EFEM model of the research ship Planet

Provided that the hydrophone is passed with a constant speed, it can be assumed that the vibrations of the ship are at a steady state. Furthermore, it is supposed that the transfer functions from the sound generating components to the hydrophone are independent of the excitation. Thus, the sound emission of the ship for different passing speeds is determined by scaling the transfer functions with the according input power spectra of the different wave types. The required input powers of the structural components can either be measured or calculated. In contrast, the determination of the sound power generated by the propeller is an even more challenging task since it is, if even possible, hard to measure. Furthermore, appropriate simulation methods, especially for the high frequency range, are entailed with very high computational costs. The development of more efficient numerical methods is the subject of current research (6, 7).

Figure 7 shows the corresponding transfer functions for a hydrophone located midships at a depth of about 25m below the water surface as depicted in Figure 6. By comparing the transfer functions of the structure-borne sound sources it is obvious that the hydrophone is least sensitive with respect to the generators. This is due to the fact that the transfer path is longer than for the other structural components, thus more energy is dissipated before it propagates through the hull into the water. The engine units and the thrust bearing of the shaft are mounted on the submerged part of the ship which results in a shorter transfer path to the hydrophone and hence yields a higher noise sensitivity. In order to point out the effect of the reinforced propulsion foundation with respect to the sound emission an additional transfer function for a direct excitation on the hull is determined. The according transfer function is denoted as shaker and shows a sensitivity regarding the hydrophone that is up to 100 times higher than for the propulsion components. Furthermore, by comparing the transfer functions of the structure-borne sound sources it turns out that the structural damping leads to a decreasing acoustic sensitivity especially at higher frequencies. The longer the transfer path, the faster the acoustic sensitivity decreases.

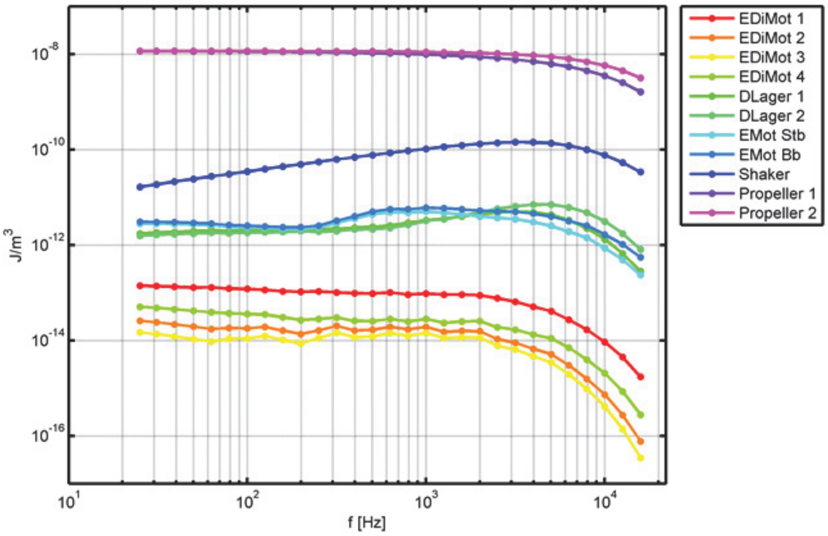

Figure 7 – EFEM transfer functions with respect to hydrophone no. 7 for different excitation components

However, the highest acoustic sensitivity is observed for the propellers which are modelled as acoustic sources including the interactions with the ships structure. The direct transfer path to the hydrophone results in a sensitivity, which is up to 10^8 times higher than for the less sensitive generators.
In addition to the transfer functions the energy distribution in the entire acoustic domain provides further information about the sound emission properties of the ship. Among other things directivity characteristics as well as the sound power can be determined. Figure 8 shows the resulting energy density distribution in the acoustic domain due to a unit load at the mounting points of the backboard engine at 1.6kHz.

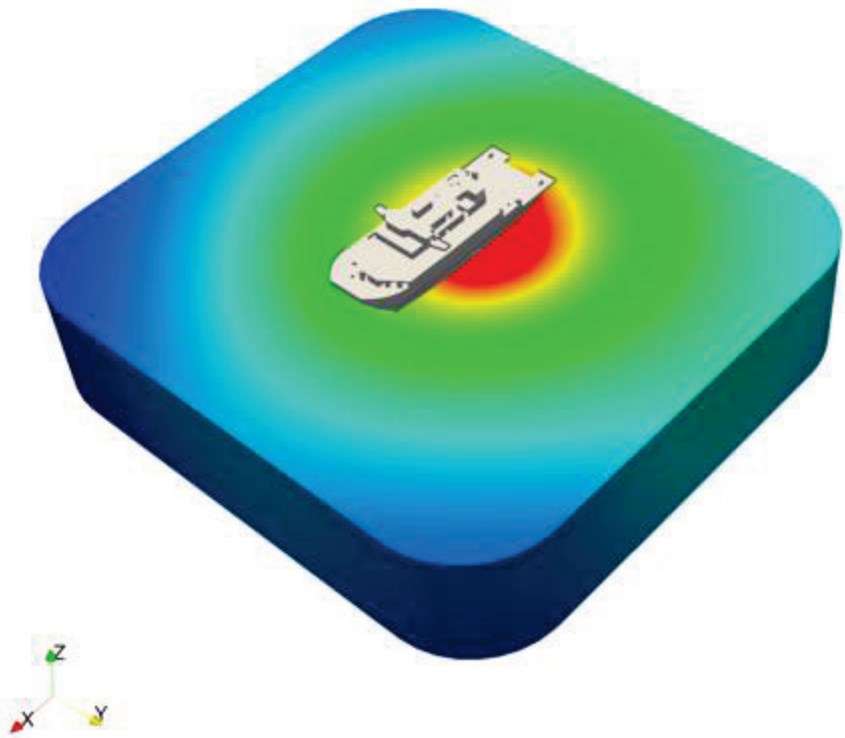

Figure 8 – Energy density distribution in the fluid domain at 1.6kHz due to a unit load at the mounting points of the backboard engine

5. CONCLUSIONS

In this contribution a limiting lowest frequency is shown for a simplified model of a single plate that is coupled to a fluid volume. In the frequency range below this limiting frequency an influence of the radiation efficiency on the numerical solvability of the system of equations is observed. For high radiation efficiency values unphysical transmission coefficients in a range greater than one are detected in this lower frequency range. In this case the solution of the system of equations is possible but should not be interpreted as physically correct.

An application of the coupling parameters is studied for the calculation of sound radiation of the research ship Planet. The sensitivities of the sound pressure level to the different excitation components are shown via the frequency dependent transfer functions and qualitative comparisons between the vibration exciting systems are presented.

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