MODELING WATER LOADING AND UNDERWATER SOUND RADIATION OF VIBRATING STRUCTURES IN DEEP AND SHALLOW WATERS

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1 INTRODUCTION

Noise and vibration simulation using classical methods such as Finite Element Method (FEM), Boundary Element Method (BEM) and Statistical Energy Analysis (SEA) are well integrated into standard design processes in the automotive, aerospace and train industry. In the marine industry, simulation starts to occupy a key role in product design as vibro-acoustics and environmental requirements are becoming increasingly demanding. This paper discusses new advances in marine vibro-acoustic predictions and in particular the effect of water loading of hull panels and underwater sound radiation. A newly developed wavelet based sound radiation formulation is used to load hull panels with sea water which allows for the computation of the ship's water loaded natural frequencies and modal damping. This allows proper vibration level predictions on the hull panels for a wide range of frequencies. This paper also presents an innovative use of Fast Multipole Boundary Element Method (FMM-BEM) to predict underwater sound radiation of vibrating structures in deep or shallow waters. Several application examples using a 70 m luxury yacht model are also described.

2 FULL FREQUENCY SIMULATION CHALLENGE

2.1 Traditional vibro-acoustic predictions

The ship industry has relied on empirical models to predict vibration and sound pressure throughout a vessel for many years. This method has proven useful when the ship to be studied is built of similar material, has similar general arrangement plan and has conventional sources as the numerous ships used to build the empirical models. Furthermore, some shipbuilding companies also used FEM to predict first few global modes of the ship and making sure the different sources would not excite the structure with the same frequencies to avoid major resonance problems. Another application of FEM is in the design of the engine foundation. A local FEM model of the engine foundation can be built and the input impedance at the location of the engine and gearbox attachment points can be computed and compared with the impedance of the mounting system. This process ensures a strong impedance mismatch and therefore limiting the amount of vibrational energy getting into the structure [1]. Finally, local FEM models can be used to diagnose local resonance problems by visualizing the mode shapes of certain panels and stiffening or damping as required. One should remember that while stiffening a panel reduces vibration levels, it can significantly increase the sound radiated by the panel and care must be taken when stiffening so one does not create an new acoustic problem while trying to fix a vibration problem.

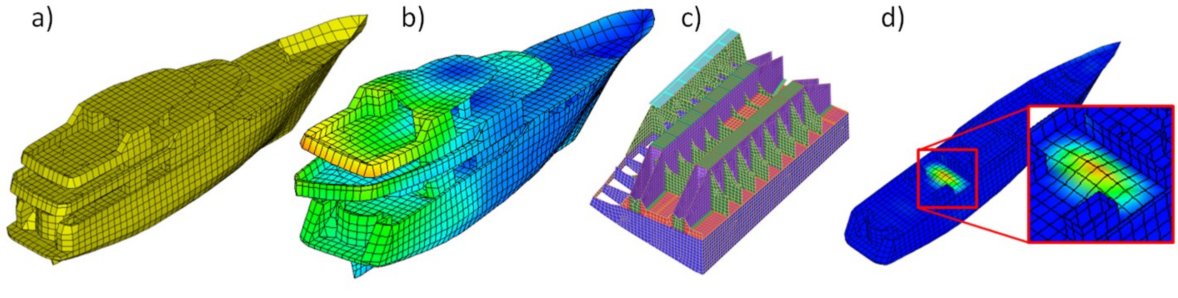


Figure 1 – a) FEM models used to compute first few global modes, b) a torsional global mode, c) engine foundation local model and d) local FEM model to diagnose panel vibration issues.

2.2 Improving sound insulation representation using SEA

SEA has been established in space, aircraft, automotive and train industry for many years now, and this method is increasingly used in the marine sector to design interior insulation [1,2,3,4]. SEA can be applied on a wide frequency range from a few hundred hertz to 10 000Hz. Model building has been greatly simplified by the use of automation (see figure 2). SEA models can now be built from 2D general arrangement drawings. Accuracy and predictivness of SEA has been widely published for other industries and in the marine industry, the number of publications increases each year.

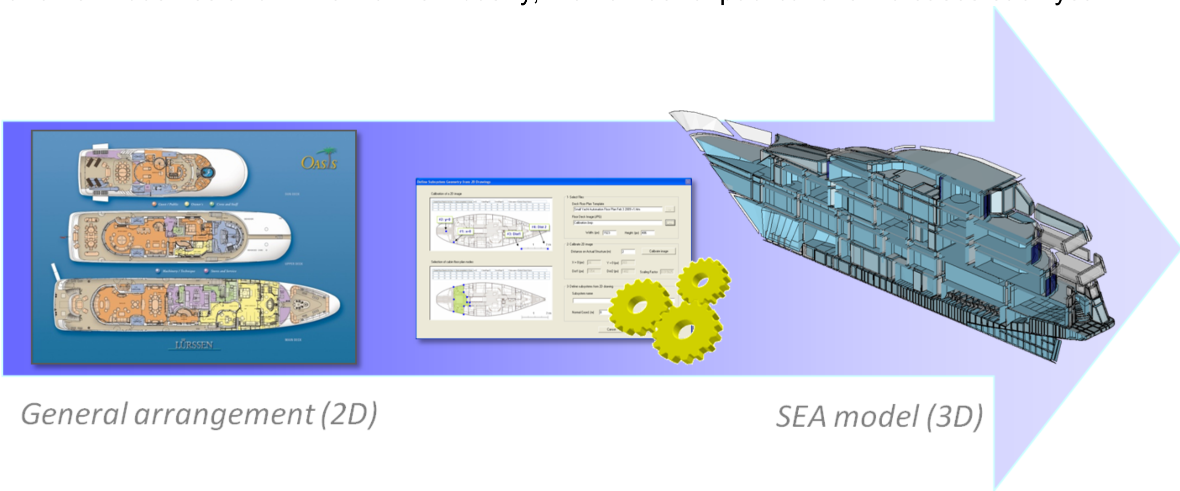


Figure 2 – Model building process greatly simplified by automatic model building from 2D general arrangement drawings.

Examples of SEA models are presented in figure 3. All ship images and results in this paper are from a model created by ESI from a 70m luxury yacht 2D general arrangement drawing found on the internet. Computations are made with commercial software VA One.

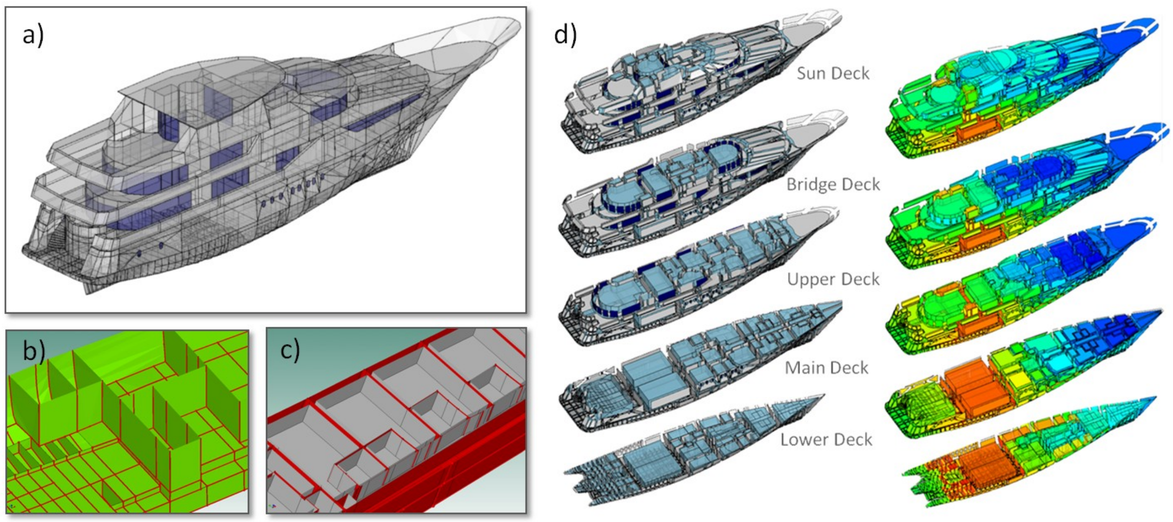


Figure 3 – a) SEA model built automatically from 2D drawings, b) structural point and line junctions (in red) between structural panels (in green) automatically created when node connectivity is enforced, c) area junctions (in red) automatically created between panels and acoustic cavities (in grey), d) images of different decks (left) and contour plot of panel velocity and cavity SPL

2.3 Full frequency vibro-acoustic analysis: Coupling FEM and SEA

A critical aspect of ship design is the modeling of the structure where the structureborne sources are attached. Since this part of the ship is usually stiff and composed of small thick panels, FEM is more appropriate for frequencies up to ~200Hz. This paper proposes a method that allows engineers to build predictive models for the full frequency domain (0-10000Hz). As previously mentioned, in the marine industry, it is common to build a FEM model of the ship for low frequency structural analysis. A SEA model can cover the high frequency domain. For mid-frequency, (20 to 200 Hz for a 70m luxury yacht) a FEM/SEA model provides a good representation of the ship's physics: FEM for stiff below water line structure and SEA for the remainder of the structure. All acoustic cavities (cabin volume of air) can be modeled as SEA (Figure 4).

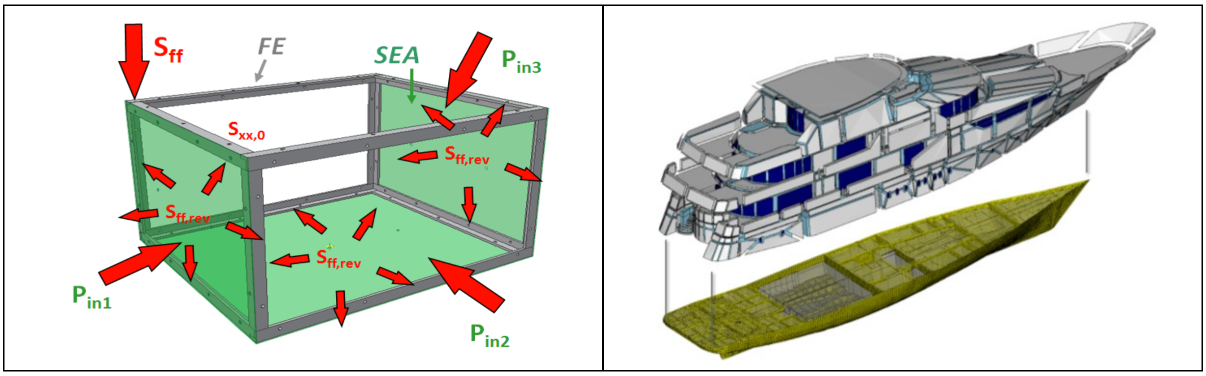


Figure 4: Left: FE/SEA Coupled: SEA subsystems in green and FEM stiff beam structure in grey. Right: Application to a luxury yacht, the stiff tightly coupled plate network at bottom of ship in FEM and large flexible panels in the upper part in SEA.

The conventional structural FEM formulation in equation 1 uses the dynamic stiffness $\boldsymbol{D_o}$ of the system to compute displacements at all FEM nodes $\boldsymbol{x}$ for a given excitation $\mathbf{f}$. Equation 1 also includes an extra term added to the dynamic stiffness $\boldsymbol{D_o}$ of the system to account for the direct field dynamic stiffness that the SEA content adds to the FEM content of the model. Actually, all modes of the system are represented: either by mode shape and natural frequencies (FEM) or as a probability of finding a certain number of modes into a frequency band (SEA). The added direct field dynamic stiffness is an average value that represents how the SEA panels and acoustic volumes load the FEM panels and beams of the model.

$$\left[\mathbf{D}_0 + \sum_i \mathbf{D}_{i,dir}\right]\{\mathbf{x}\} = \{\mathbf{f}\} \qquad 1)$$

Equation 2 shows that the total response at each FE node equals the sum of external excitations applied directly onto the FEM parts ($\boldsymbol{S_{xx,0}}$) and the reverberant energy contained in the SEA panels and acoustic volumes($\sum_{\mathrm{i}} (S_{\mathrm{xx,rev,i}})$).

$$\mathbf{S}_{xx} = \mathbf{S}_{xx,0} + \sum_i \mathbf{S}_{xx,rev,i} = \mathbf{R}\left[\mathbf{S}_{ff,0} + \sum_i \mathbf{S}_{ff,rev,i}\right]\mathbf{R}^H \quad \text{where} \quad \mathbf{R} = \left[\mathbf{D}_0 + \sum_i \mathbf{D}_{i,dir}\right]^{-1} \qquad 2)$$

See [5,6] for a detailed description of full coupling between FEM and SEA parts.

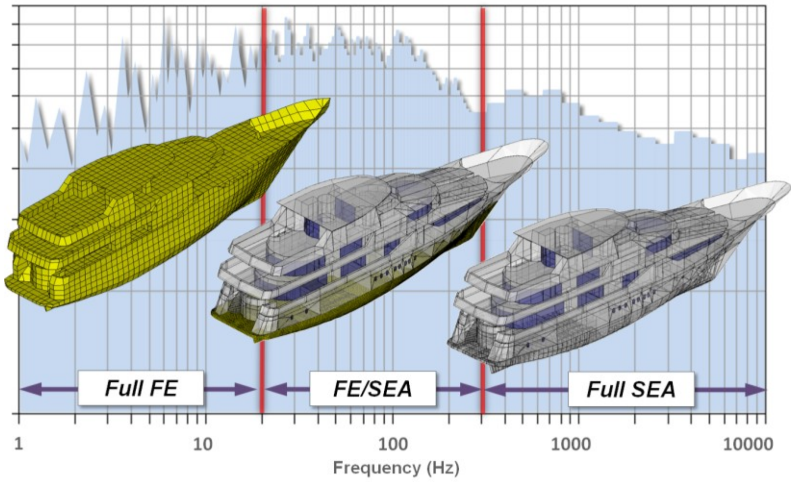


Figure 5: Full frequency analysis concept: From deterministic/narrowband low frequency FE model to statistical 1/3rd octave band high frequency SEA model.

This new formulation allows for a full frequency vibro-acoustic analysis of a ship using FEM for low frequency, FE/SEA coupled for mid-frequency and SEA for high frequency.

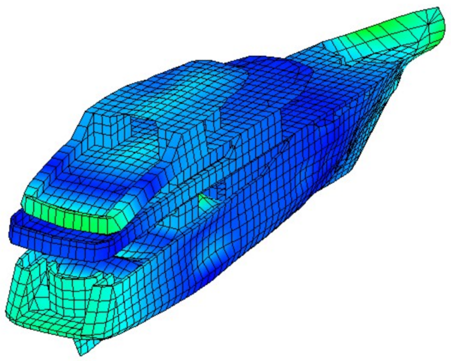
3 FLUID LOADING

3.1 New Formulation based on wavelets

Fluid loading plays an important role in the vibration of the hull, especially at low frequency. The loading actually changes natural frequencies and mode shapes in a significant way. Therefore, one cannot ignore the fluid loading in any predictive model of vibration and noise radiation for hull panels as well as all tanks (water, fuel, waste) in a ship. A new efficient fluid-structure analysis method [7] makes use of wavelets to compute the acoustic radiation from baffled, unbaffled, or partially baffled planar structures. The surface displacement and the surface pressure are expressed in terms of wavelets, and the acoustic dynamic stiffness (baffled case) or the acoustic receptance (unbaffled case) between any two wavelets is derived in closed form. This formulation is implemented into the commercial software VA One. In the present work, this formulation is only used to compute velocity on the hull panels. Underwater radiation computation is done using FMM -BEM and is presented in the following section.

3.2 Numerical examples

The 70m luxury yacht model was used to compute hull panel velocity with the new wavelet formulation (SIF in all graphs), with traditional BEM and finally with SEA. Figure 6 show the effect of water loading on natural frequencies of the structure modes. It can be seen that loading the hull panels with sea water decreases the natural frequencies by as much as 4.5 Hz, from 12.8 Hz to 8.3 Hz (mode 9). Figure 7 shows a comparison between the wavelet method (SIF) and traditional BEM. It can be observed that water loading decreases hull panel vibration at the node where the force is applied by as much as 20 dB at low frequency. The wavelet approach (SIF) tracks reasonably well the reference BEM response. One has to remember that the final goal is to compute the acoustic sound radiation and that the average panel velocity is therefore most important. A small nodal variation can be tolerated without major effect on acoustic radiation when a computational speedup is needed. In this example, a gain of a factor of 5 can be achieved (from 2.8 sec/freq to 0.58 sec/freq) by using the wavelet approach. Finally, the effect of water loading was also evaluated for higher frequency on the average velocity of one hull panel close to the excitation and a difference of only a few dB can be observed for the frequency range between 200 Hz and 10 kHz.



Node ID	Uncoupled (Hz)	Coupled (Hz)	Damping (-)
1	6.2	5.2	0.034
2	7.4	5.9	0.049
3	7.7	6.9	0.015
4	9.5	7.4	0.010
5	11.0	7.5	0.010
6	12.0	7.8	0.015
7	12.2	7.9	0.011
8	12.3	8.2	0.011
9	12.8	8.3	0.016
10	13.1	9.2	0.021

Figure 6: Left: First bending mode shape, Right: Uncoupled and coupled natural frequency and coupled modal damping

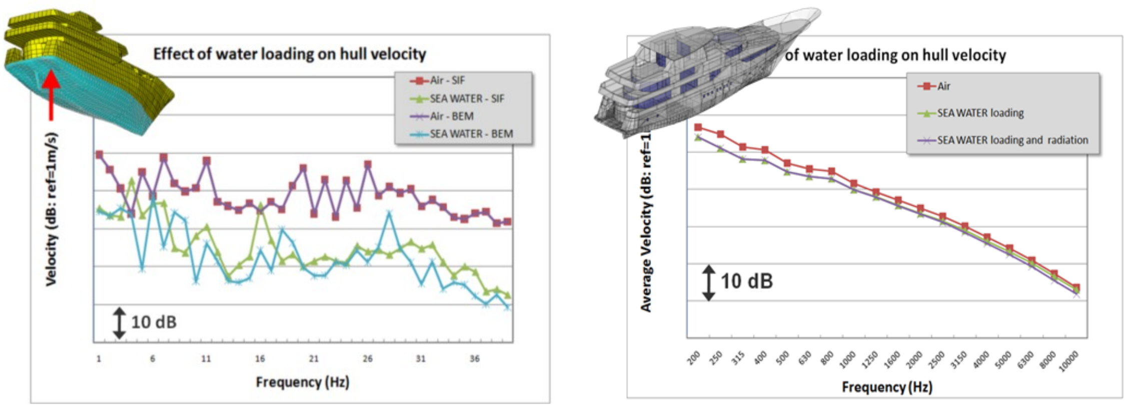


Figure 7: Left: Comparison between BEM and Wavelet approach (SIF) shows that water loading decreases hull panel vibration at a single node by as much as 20dB. Right: At higher frequency (SEA) water loading reduces panel average velocity be only a few dBs.

4 UNDERWATER RADIATION

4.1 Method used: FMM-BEM and SEA

To represent the fluid around the yacht hull, the FMM-BEM method has been selected. This method provides a detailed description of the wave propagating from the hull, the scattering of waves around the complex contour of the hull and is therefore appropriate to predict underwater radiated noise. The FMM-BEM formulation from Gumerov and Duraiswami [8,9] coupled to an ILUT pre-conditioner was used to compute the response at keel aspect and to generated the contour plots of underwater pressure distribution. The following sections describe results for deep and shallow waters.

4.2 Numerical example for deep waters

The 70m luxury yacht model was used to demonstrate both underwater radiation from the vibrating hull and the scattering of waves from a point source located at propeller blades. Figure 8 shows pressure distribution for both cases. On the left, the hull panels wetted by the water are vibrating and velocity is plotted. In the fluid domain, the pressure fluctuation generated by the vibrating panels is also shown. On the right side of figure 8, the hull is considered rigid and wave travelling from propeller position to the environment are shown. Computation time for FMM-BEM underwater radiation is 60 sec/freq up to 200 Hz as opposed to 720 sec/freq for the diffraction problem.

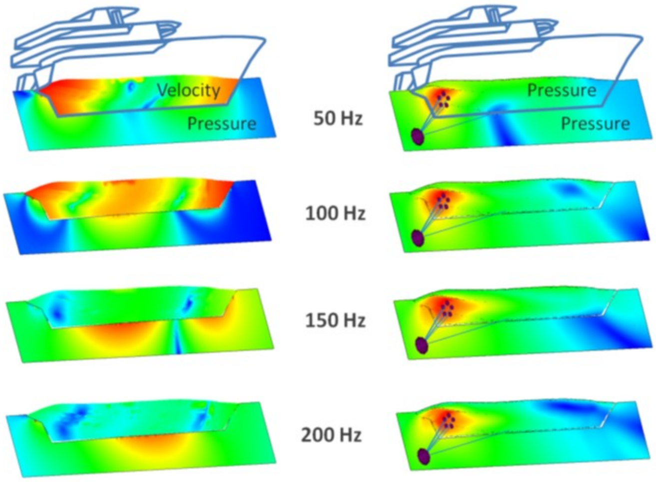


Figure 8: Left: hull underwater panel velocity and radiation into sea water.
Right: underwater wave propagation from 5 monopoles at propeller blade locations.

Figure 9 shows underwater sound radiation from the vibrating hull panels for frequencies 20 and 50 Hz. It also shows the directivity pattern of the yacht at frequencies from 8 to 50 Hz. Note the p=0 condition at water surface on contour plots and the strong directivity of pressure distribution around the yacht at these frequencies.

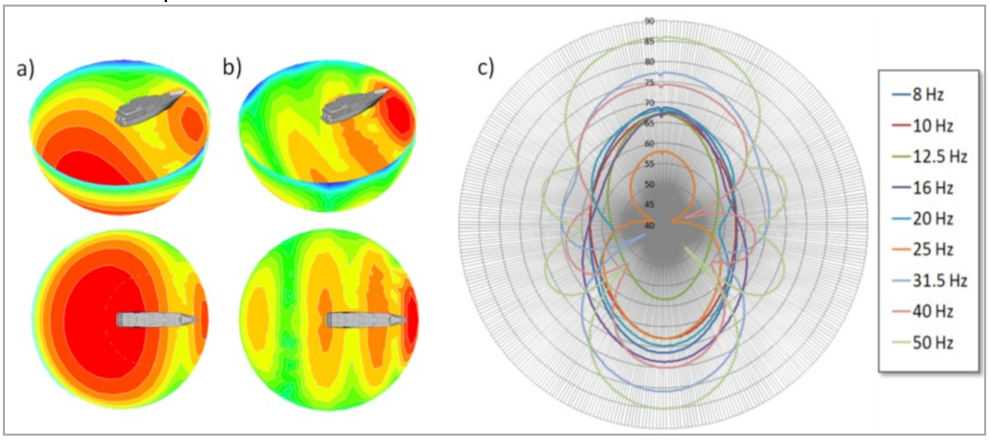


Figure 9: Deep water sound radiation with water surface baffle (p=0). a) 20 Hz b) 50 Hz
c) sound radiation directivity pattern from yacht hull from 8 Hz to 50 Hz

4.3 Numerical example for shallow waters

To analyze shallow water problems the following approach was adopted: a half sphere with water surface at top was used. The plane that represents the bottom of the sea was created at 50m and the half sphere and bottom plane were combined to create the shallow water fluid domain. A full convergence study should provide the proper radius of the half sphere to use to insure the smallest possible model can be used to reduce computation time/memory usage or to increase maximum upper frequency. In this study, only two different radii are compared: 200m and 600m. The benefit of this method is that one can add impedances to the bottom of the sea to represent the different materials composing the sea bed. One can also match the sea bed topology since the model does not use an infinite flat plane but a real BEM surface. Full convergence analysis and various topology/material for the sea bed will be presented in future publication. Figure 10 illustrates the modeling concept.

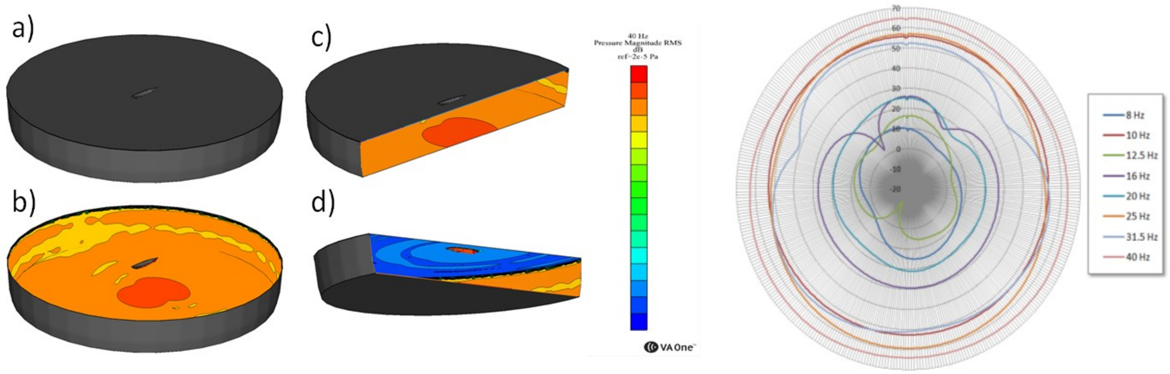


Figure 10: Left: a) external view of shallow water fluid BEM domain: Half sphere of 600m radius cut at 50 m from water surface. b) water surface removed to view pressure distribution on sea floor and at lateral edges of BEM domain. Notice yacht in the middle of water surface. c) BEM domain clipped to reveal pressure distribution on the inside of the domain. d) BEM domain clipped to reveal pressure distribution under the water surface. Right: sound radiation directivity pattern from yacht

Surfaces in gray are outside the BEM domain: in a) to d) the BEM domain is shown and the ship can be best seen in b). Note the water surface with pressure close to zero. On the surfaces of the BEM domain, impedances have been used to represent the proper boundary conditions: the water surface impedance is close to zero, the lateral walls of the water domain have impedance that corresponds to sea water and finally the sea bed has impedance twice as large as sea water (arbitrary choice for this particular example). A representative impedance of the sea bed materials should be used when available.

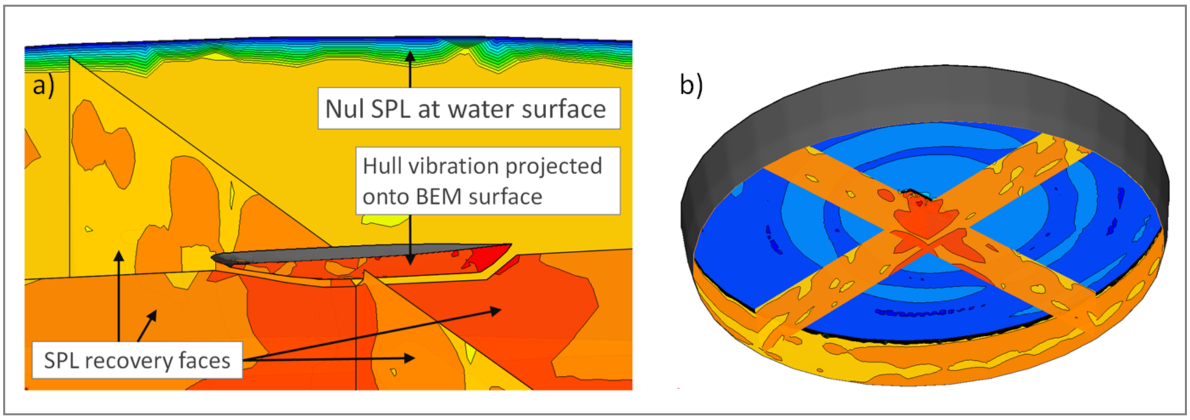


Figure 11: a) Detailed view of pressure distribution within shallow water BEM fluid domain: Notice pressure distribution on hull and propagation onto the recovery faces. Also note the p=0 condition at top of BEM domain (water surface). b) Water surface pressure distribution clearly visible

Figure 10 also presents the directivity pattern of the yacht for underwater radiation into shallow waters. They differ greatly from the deep water case. To visualize the pressure distribution, recovery planes have been created and are shown in figure 11. Finally, figure 12 shows comparisons of pressure distribution between the case with radius of 200m and radius of 600m on the left. A red circle on the 600m case represents the boundaries of the 200m case. One can see that the pressure distribution within this red circle and the 200m results are comparable. On the right of figure 12, the keel aspect SPL is shown. A small difference between the 200m and 600m cases suggest that the model can be relatively small and that the impedance approach used doesn’t introduces significant unwanted effects. These are preliminary results and a full convergence analysis will be published at a later time.

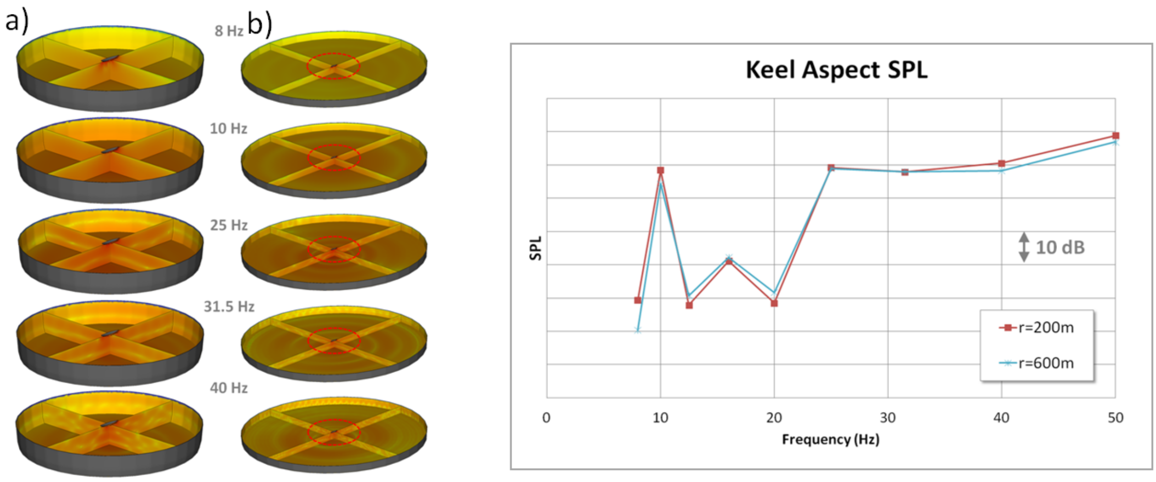


Figure 12: Left: Pressure distribution for shallow waters BEM fluid domain of a) 200m and b) 600m. Right: Keel aspect SPL suggest the smaller radii would be sufficient to properly predict SPL

5 CONCLUSION

This paper has presented an application of a new wavelet approach to water loading on a 70m luxury yacht. It has also demonstrated that computing underwater sound radiation using FMM-BEM shows promising results and that if the model can be kept relatively small the upper frequency limit of the analysis could be pushed upward significantly. Finally, this paper presents a new approach to predict underwater sound radiation for shallow waters based on FMM-BEM and the use of a bounded fluid domain in combination with surface impedances to represent the water surface, water lateral surfaces and the sea bed.

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