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FULL FREQUENCY NOISE AND VIBRATION CONTROL ONBOARD SHIPS

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The marine industry has in the past extensively used empirical models to predict transfer functions between source locations and noise sensitive cabins. These empirical methods work well for standard construction types, materials and small number of cabins. Today's tendencies are to use complex construction methods, exotic material such as composite and build larger and larger yachts with cabin layouts and numbers not easily represented in an empirical way. This paper presents an approach to build predictive vibro-acoustic models for full frequency analysis (0-10kHz). The approach makes use of several modeling methods and coupling such as FEM (Finite Element Method), FMMBEM (Fast Multipole Method-Boundary Element), SEA (Statistical Energy Analysis) and "FE/SEA Coupled" to represent the yacht structure, interior cabins, fluid tanks, underwater fluid loading and noise radiation. This approach also permits the representation of the acoustic insulation and the optimization of its content to achieve required targets while reducing mass and insulation cost. This paper also discusses the source models to be used to represent the airborne, structureborne and waterborne contribution of major excitations. This approach is applied on a 70m luxury yacht where these vibro-acoustic concepts are discussed and illustrated.

1. Introduction

This paper describes in section 2 the traditional vibro-acoustic predictions methods in the marine industry and proposes ways to improving sound insulation representation using SEA and then introduces an advanced method of predicting full frequency vibro-acoustic response by coupling FEM and SEA. In section 3, a new wavelet heavy fluid loading method is describes and illustrated by a numerical example. Section 4, presents numerical examples for shallow waters sound radiation computation using the FMM-BEM method. Section 5 discusses how sources can be represented in the models discussed.

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 (Figure 1).

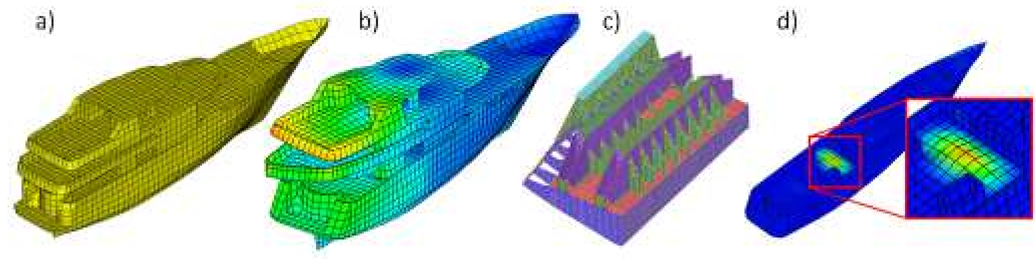


Figure 1: a) FEM models used to compute first few global modes, b) a torsional global mode, c) engine foundation local model and d) local FEM model to diagnose panel vibration issues.

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 a new acoustic problem while trying to fix a vibration problem.

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 (Figure 2).

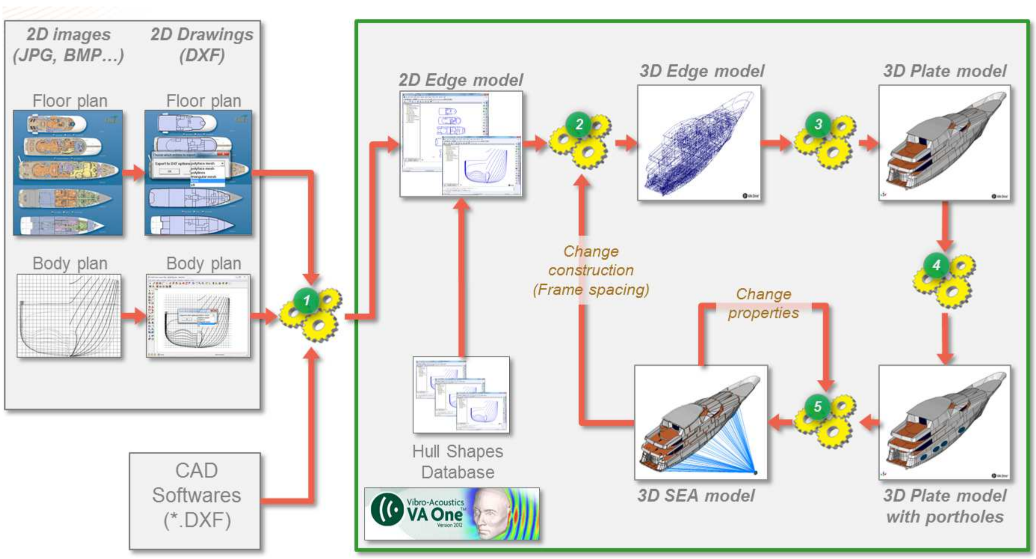


Figure 2: Model building process greatly simplified by automatic model building from 2D general arrangement drawings.

SEA models can now be built from 2D general arrangement drawings in a few days. This automated method is based on a 2D cabin layout and body plan drawing. From these, a process has been developed to define an 2D edge model that is later used to create a 3D edge model (Figure 3).

From the 3D edge model, a Doubly-Connected Edge List (DCEL) approach is used to find all possible faces within an edge network. This data structure provides an efficient manipulation of the topological information associated with nodes, edges and faces in a network. It is used to find all possible closed faces within a plane.

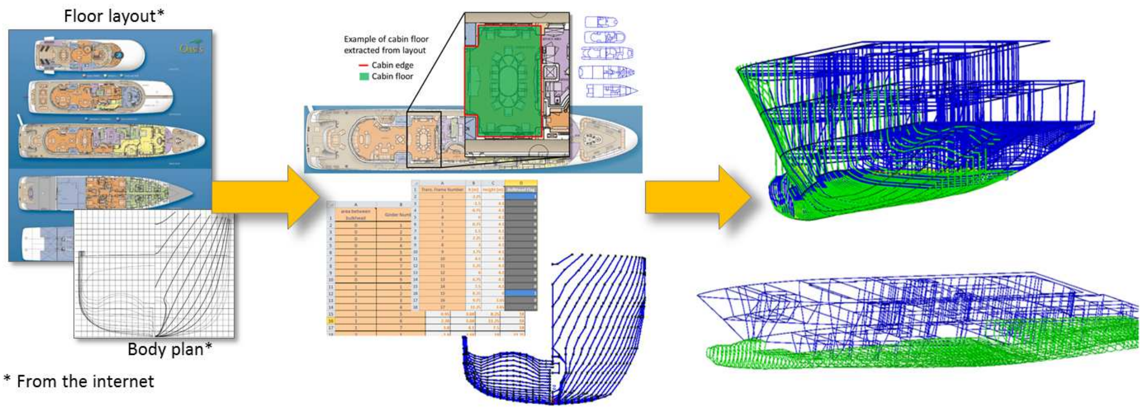


Figure 3: Creating a 3D edge model from 2D general arrangement and body plan drawings.

Therefore, the 3D edge model is broken down in planes defined by 3 nodes associated with a pair of connected edges. The algorithm loops through each possible plane that exit in the 3D edge network. For each plane, the algorithm travels along the edges until it closes a face. If end is open because current edge is travelling away from the current plane surface, the algorithm discards the current face. When all possible faces are found in all possible planes, the SEA plates are created (Figure 4).

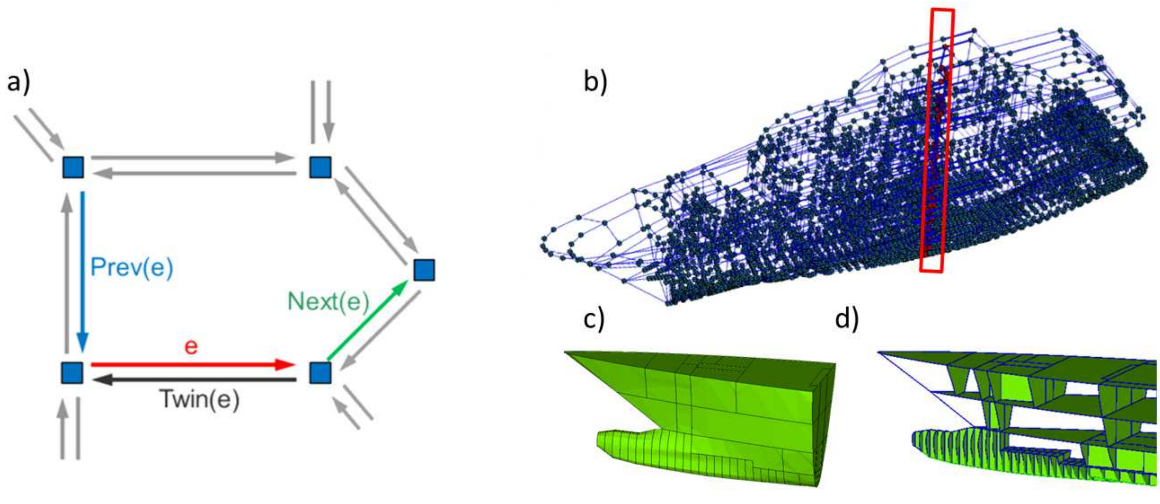


Figure 4: Creating a 3D plate model from 3D edge model using DCEL approach: a) DCEL diagram, b) Looping through all planes in 3D edge network to find all possible closed faces, c) Hull panels created from an unwrapped edge network of the hull and d) interior panels created from faces

The insulation content in a yacht can be described by either a treatment layup describing each layer of a noise control treatment or as an insertion loss. Many insulation designers use a condensation model that uses an insertion loss and a damping spectrum to represent the insulation treatment. This makes model building even easier since the insulation does not have to be explicitly created in a structural model but only as a treatment applied on a base panel. Examples of SEA models are presented in figure 5. 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 and post-processing are made with the commercial software VA One. Accuracy and predictiveness of

SEA models have been widely published for other industries and in the marine industry, the number of publications is increasing each year.

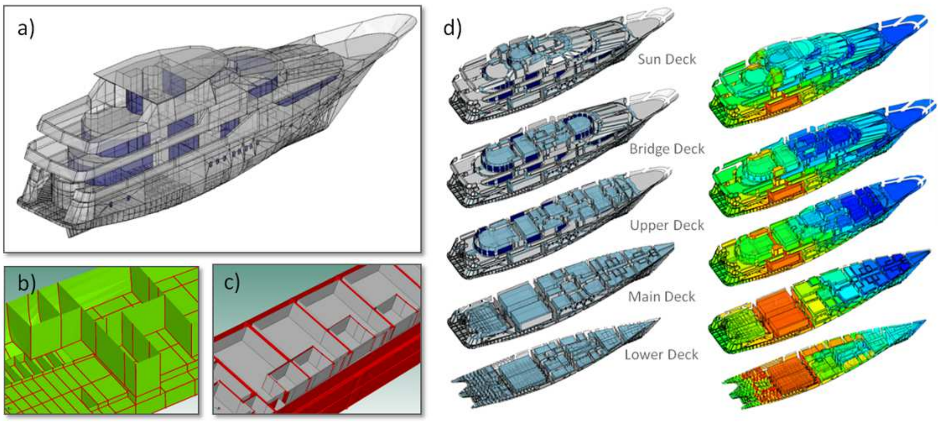


Figure 5: a) SEA model built automatically from 2D drawings, b) structural point and line junctions (in red) between structural panels (in green) automatically created when node connectivity is enforced, c) area junctions (in red) automatically created between panels and acoustic cavities (in grey), 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 modelling 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 modelled as SEA (Figure 6).

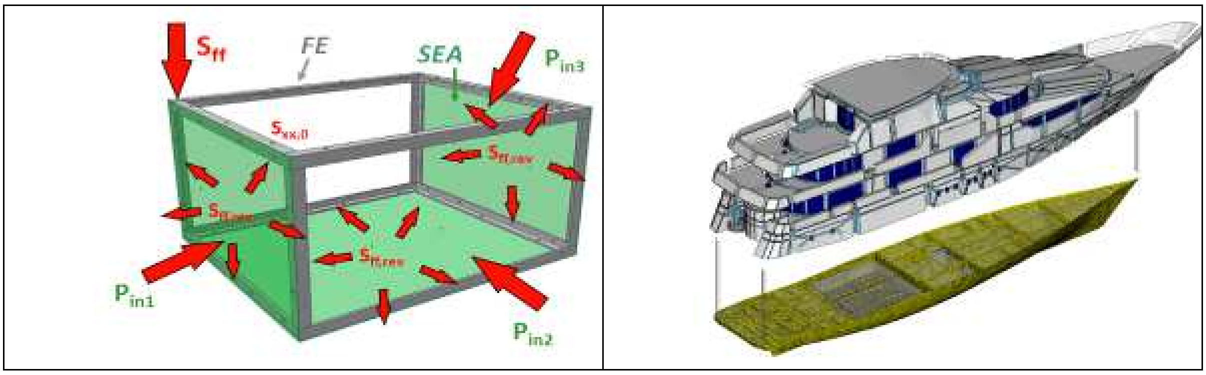


Figure 6: Left: FE/SEA Coupled: SEA subsystems in green and FEM stiff beam structure in grey. Right: Application to a luxury yacht, the stiff tightly coupled plate network at bottom of ship in FEM and large flexible panels in the upper part in SEA.

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 ($S_{xx,0}$) and the reverberant energy contained in the SEA panels and acoustic volumes($\sum_i (S_{xxrev,i})$).

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

See [5,6] for a detailed description of full coupling between FEM and SEA parts. 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 (Figure 7).

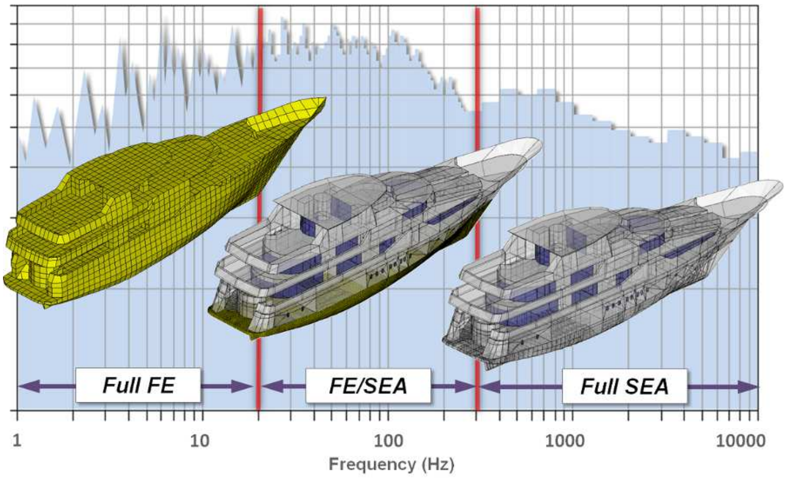


Figure 7: Full frequency analysis concept: From deterministic/narrowband low frequency FE model to statistical 1/3rd octave band high frequency SEA model**.**

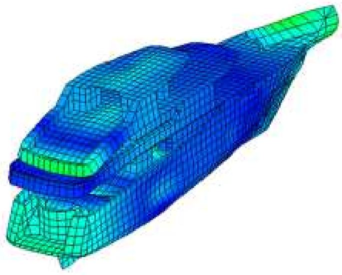
3. Fluid loading

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 8 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).



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 8: Left: First bending mode shape, Right: Uncoupled and coupled natural frequencies

Figure 9 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 time speedup is needed.

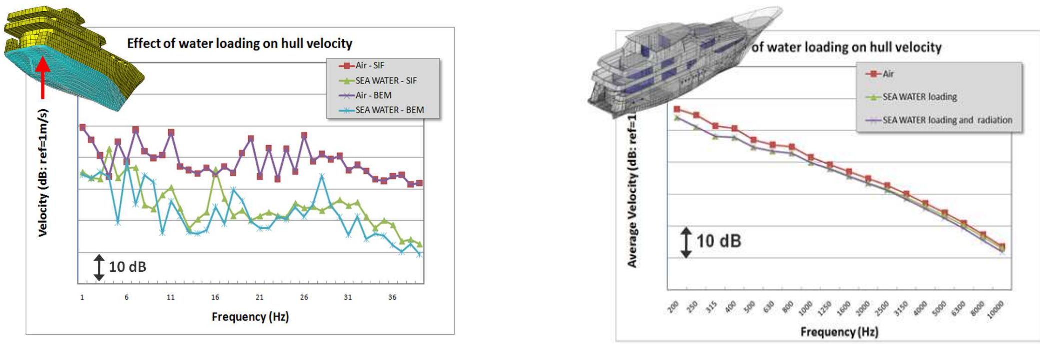


Figure 9: Left: Comparison between BEM and Wavelet approach (SIF) shows that water loading decreases hull panel vibration at a single node by as much as 20dB. Right: At higher frequency (SEA) water loading reduces panel average velocity be only a few dBs.

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.

4. Underwater radiation

4.1 Method used: FMM-BEM and SEA

To represent the fluid around the yacht FEM 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] is an advanced algorithms in order to solver large scattering and radiation problems well suited for forced response computation. The solver is based on a multilevel *Fast Multipole Method* to solve the Helmholtz boundary integral equation for the acoustic unknowns of pressure and velocity. The FMM BEM solver is intended for mid-sized to large scattering and radiation problems, typically problems involving from approximately 10,000 to 2,000,000 nodes in the rigid and elastic faces defining the domain. The solver is expected to give accurate results for $0.0025 < kD < 500$, which corresponds to 4.10^{-4} to 80 acoustic wavelengths throughout the BEM domain. The solving time is expected to scale in $(kD)^3$; part of this behavior is due to the expected increase of GMRES iterations as kD increases. The FMM-BEM is coupled to a ILUT pre-conditioner to compute the response and to generated the contour plots of underwater pressure distribution. The following sections describe results for shallow waters.

4.2 Numerical example for shallow waters

To analyse 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 20m 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. 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. Figure 10 illustrates the modelling concept and typical results.

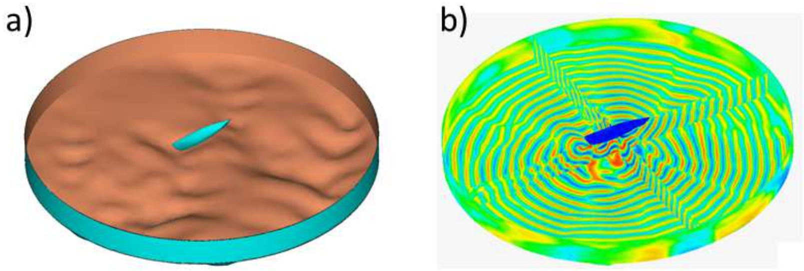
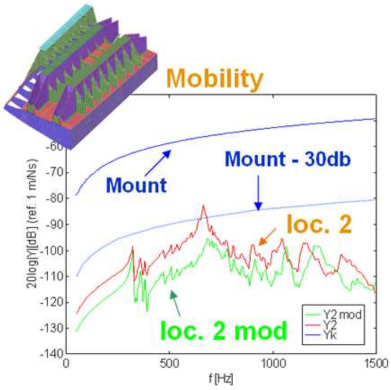


Figure 10: a) Topology used in the analysis, b) Underwater wave propagation at 200 Hz

5. Sources

Typical sources are of 3 types: airborne, structureborne and waterborne. Main sources of noise and vibrations are engines, gearbox, generators, HVAC, bow thrusters and propellers. To keep the ship model as simple as possible, a side model is built in finite element (FE) to compute the structure input mobility at the source location (Figure 11). These mobilities are compared with idealized mount mobilities. Structural design changes are recommended if i) less than 30dB between mount and structure ii) structural mobility is higher than -100 dB or iii) resonance peaks occur. From force and input mobility, a power input is computed. It is then applied to the appropriate subsystems in the SEA model. An FE model of the source structural foundation is used to set adequate input mobility at attachment points. Right) Airborne and structureborne sources are applied to the model as either constraints or power sources. When no power input is available, a force spectrum or an acceleration constraint is used. Propeller is another important source that will be addressed in a separate communication. In addition to these sources, the HVAC systems power input can be assigned in the location of the HVAC unit and additional HVAC power sources added to individual cabin to represent flow noise.



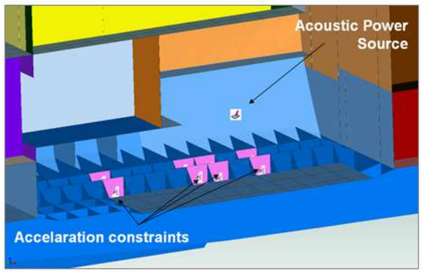


Figure 11: Left) A FEM model of the source structural foundation is used to set adequate input mobility at attachment points. Right) Airborne and structureborne sources are applied to the model as either constraints or power sources.

6. Conclusion

This paper presented an approach to build predictive vibro-acoustic models for full frequency analysis (0-10kHz). The approach makes use of several modeling methods and coupling such as FEM (Finite Element Method), FMMBEM (Fast Multipole Method- Boundary Element), SEA (Statistical Energy Analysis) and "FE/SEA Coupled" to represent the yacht structure, interior cabins, fluid tanks, underwater fluid loading and noise radiation. This approach also permits the representation of the acoustic insulation and the optimization of its content to achieve required targets while reducing mass and insulation cost. This paper also discussed the source models to be used to represent the airborne, structureborne and waterborne contribution of major excitations. In the near future, cabin absorption computation based on surface absorption of various furniture and equipment present in a cabin will be investigated. ESI is also in the process of defining source models for propellers and HVAC systems.

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