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A numerical model calibrated by in-situ measurement for predicting the underwater radiated noise of an azimuth thruster structure of a passenger ship

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ABSTRACT

In a global context of limiting the impact of noise pollution on the behavior of marine fauna, the understanding of acoustic radiation phenomena from underwater sound sources of a passenger ship is of primary importance. Although diesel engines and propellers are considered to be one of the main sources of ship noise, our observations have shown that immersed motors contribute significantly to the noise radiated into the water by passenger ships at low speeds. This contribution summarizes a semi-empirical method for estimating the noise radiated into the water by an azimuth thruster-type structure. An analysis of the underwater vibroacoustic behavior of the structure excited by an internal acoustic source is detailed. Beyond the complexity of performing full-scale measurements in a noisy environment, the analysis of the measurements allows to estimate the vibration modes of the immersed structure. A numerical model of the dedicated structure, calibrated on the basis of an experimental modal analysis, is used to estimate the pressure field radiated by the structure into the water. As a result of these analyses, a simplified model of the radiated noise in the water is developed for use during nominal thruster operation.

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1. INTRODUCTION

In the last decade, the increasing underwater noise pollution has led many stakeholders in the maritime industry to address the impact of underwater radiated noise (URN) on marine life. As a result, several projects have been initiated at the European level with the aim of quantifying this noise and finding solutions for its reduction in the water [1]. At the international level, the International Maritime Organization (IMO) has published guidelines for the reduction of underwater radiated noise from shipping to address adverse impacts on marine life, with annual review of methods and deadlines [2]. Growing awareness has prompted shipowners and shipyards to study the issue and explore possible methods to reduce ship noise. Among all the challenges involved, one of the most important is to understand how a ship generates underwater radiated noise and then to characterize each of the sound sources that contribute to the overall noise of the ship. If the propeller noise source has been known for many years as the main underwater noise source due to various cavitation phenomena [3], underwater radiated noise measurements on passenger ships have highlighted the contribution of another source at low speed: the azimuth thruster. Electric azimuth propulsion systems are becoming increasingly popular for maritime propulsion due to their energy efficiency. It is important to thoroughly investigate these propulsion systems, especially in terms of their impact on underwater radiated noise, in preparation for their widespread use. This contribution outlines different approaches to characterize this noise source. The characterization of vibroacoustic sources involves various methods, but typically requires experimental investigations. The nature and complexity of these studies vary depending on the context and the intended objectives. While experimental protocols are well defined in certain cases, addressing the industrial constraints often require the development of innovative approaches and the refinement of specific methodologies [4]. This is the case in this study where we have to face with various environmental and structural challenges, which implies the development of an appropriate methodology. The present work includes an in-situ experimental modal analysis coupled with radiated water pressure measurement of the structure. A numerical approach such as the finite element/boundary element method is used to determine the radiated pressure in the far field is conducted. The numerical model has been calibrated based on the experimental results to develop a simplified method for estimating the radiated noise in water of these structures.

2. EXPERIMENTAL APPROACH

Measurements are performed on the actual structure at the end of the shipbuilding process, in an uncontrolled industrial environment. A brief overview of the structure and the measurement environment helps to understand the measurement constraints and complexities.

An azimuth thruster is a large immersed steel structure located at the stern of a ship beneath the hull (see Figure 1).

Measurements are conducted in the following industrial context :

- Dockside of an operational port (relatively calm on the day of measurement)
- Very shallow water (less than 35 m)
- Very limited access to the interior of the azimuth thruster (restricted to the manufacturer)

Two simultaneous measurement approaches were carried out: one aimed at characterizing the dynamic behavior of the structure through an Experimental Modal Analysis (EMA), and the other, in parallel, focused on characterizing the acoustic radiation efficiency of the structure through the radiated pressure field in the water resulting from the vibration measurement of the structure.

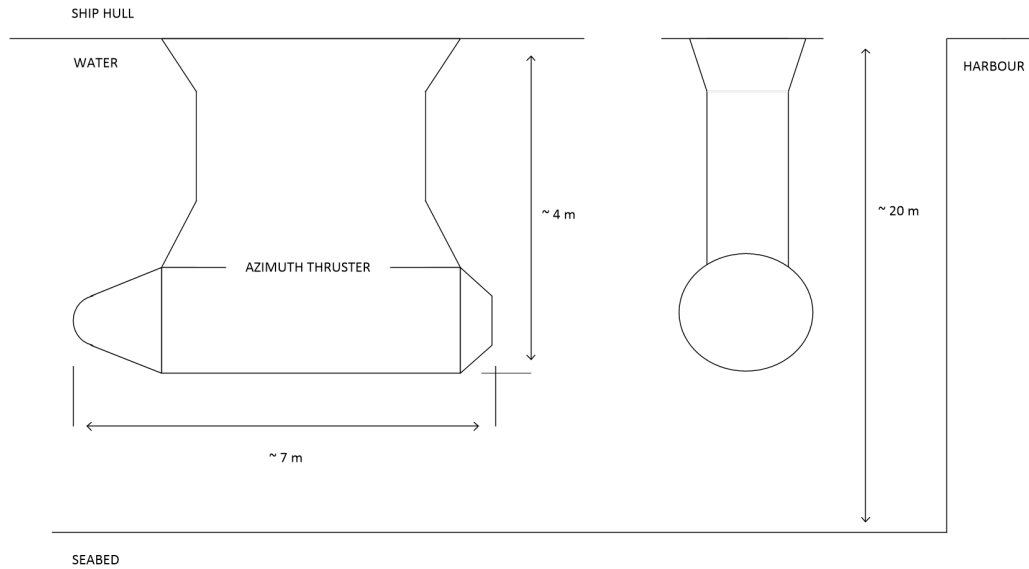


Figure 1: Structure and environment characteristics.

2.1. Experimental Modal Analysis

Experimental modal analysis is a well-known approach to evaluate the dynamic behavior of a structure based on the knowledge of its natural vibration modes, which are intrinsic characteristics of the system: it provides information on the vibration modes coupled with a heavy fluid (water). To evaluate the FRF required by the EMA, traditional structural excitation tools, such as impact hammers and shakers, have proven to be efficient. However, it is important to note that these tools do have their limitations in our case. To excite a large, stiffened surface with limited space for an excitation system, we investigated an alternative solution: an acoustic monopole source.

Modal analysis using acoustics can provide accurate results for structures with flat surfaces, although it is a less commonly used technique [5]. In his study, *Hugues* [6] performed an acoustic radiation analysis of a cylindrical structure that was excited by a monopolar acoustic source. While the main objective of the study was to identify approaches that provide a good balance between accuracy, computational cost, and parameter flexibility, it was found that an acoustic source can effectively excite a cylindrical structure over a wide frequency range and allow estimation of the radiated pressure field in water. Based on the configuration of the structure, the environment, and the available methods, we decided to use an acoustic monopole source as the input for our experimental modal analysis.

2.1.1. Experimental protocol

In a confined environment, such as an azimuth thruster, the most effective way to excite the structure is to suspend a source inside the structure. To accurately measure the induced vibrations, the only viable option is to measure the structural vibrations externally, in the water, using a waterproof accelerometer and a diver. Thus, a MIMO (Multiple-Input Multiple-Output) structural response measurement is performed using a rowing accelerometer in combination with a loudspeaker as a multiple-input (same excitation but input and output measurement at each point). An omnidirectional loudspeaker, '01dB LSO1', generates pink noise in the frequency range [100 – 3000 Hz]. At the same time, a PCB PMR 426 microphone measures the induced acoustic pressure. Vibrations on the structure are measured at 19 points spaced approximately 1 m apart. Due to the size of the structure and the measurement conditions (underwater by a diver and in an industrial environment with time constraints), it was not possible to refine the mesh of vibration measurements.

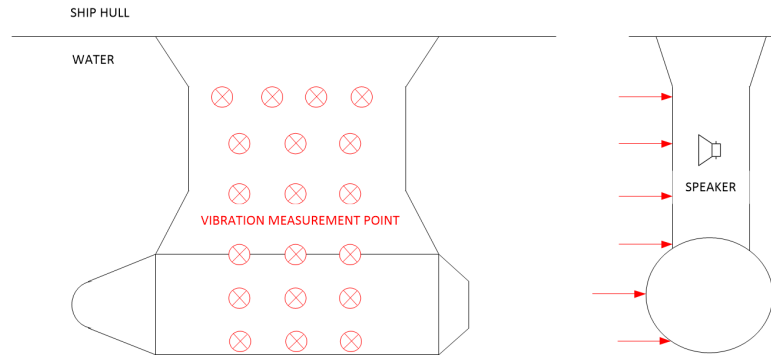


Figure 2: Experimental modal analysis protocol with each vibration point measurement (output) and the excitation position in the azimuth thruster

The excitation is considered significant enough to stimulate the structure at each measurement point, with a level of approximately 110 dB on the frequency band. This hypothesis was confirmed by measuring the vibration background noise. Within the framework of our study, the analyses will focus on the frequency range of [100 Hz – 1000 Hz].

2.1.2. Modal parameter estimation

Various curve fitting methods, including local, global and poly-reference methods, are used in modal testing [7]. Several mode identification tools are described in the literature, and the most common ones are listed below:

- Summation function: amplify predominant peaks within frequency response functions. Used as primary tool when modes are sufficiently separated [7]
- Mode Indicator Function (MIF): help to determine the number of modes in a set of experimental data [7]
- Complex Mode Indicator Function (CMIF): singular value decomposition of frequency response function matrix [7]
- Stability diagram: use to distinguish physical mode and spurious mode [8]

In experimental modal analysis, determining the number of modes within the frequency band of interest is critical. While the frequency response analysis, with primary tools such as summation function and MIF, provide a first indication, distinguishing multiple or closely spaced modes remains complex. We decided to use the stability diagram tool to overcome this problem. A stability diagram succinctly represents the poles of the system by progressively fitting the model with an increasing number of poles. It helps to identify convergent real modes from inconsistent or random numerical modes, thus facilitating pole selection based on the stability of natural frequencies, modal damping between successive model orders [7]. The Least Square Complex Frequency (LSCF) method is the type of model used to extract the modal parameters.

The choice of a high-order model (polynomial order high = 100) for estimating stable poles in high-frequency and high-damping scenarios is justified by the use of noisy FRFs. We identify more than 40 poles that are stable in frequency and damping in the range [100 - 1000 Hz]. An excerpt is summarized in Table 1.

The natural frequencies extracted from our experiment are used to calibrate the dynamic behavior of the numerical structure model. Once calibrated, the model is used to numerically determine the pressure level radiated by the structure. The radiation factor tool is then used to compare the pressure field measurements and the numerical model.

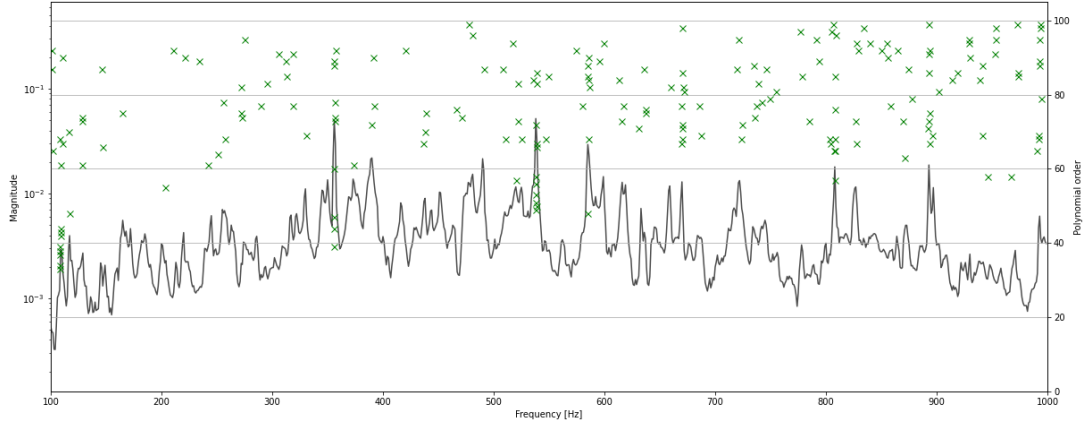


Figure 3: Stability diagram a mean of FRFs. Green crosses correspond to frequency and damping stables poles.

Table 1: Partial data from azimuth thruster natural frequencies experimental modal analysis

Frequencies (Hz)	108.6	146.6	...	895.6	945.9
Damping (%)	<1	> 0.1	...	> 0.1	> 0.2

2.2. Radiation efficiency measurement

While experimental modal analysis can be used to determine the wet modes of the structure to calibrate a numerical model and to understand how the structure behaves with such an added mass of heavy fluid, the primary objective of this study is to characterize the source in terms of radiated noise in water. As mentioned above, a measurement of the pressure field radiated by the structure was performed using a B&K microphone immersed near the structure to measure the radiated pressure field (see figure 4).

The acoustic radiation efficiency factor of the structure is determined to characterize the vibroacoustic behavior of the immersed thruster. Acoustic radiation efficiency factor is defined as the ratio of the radiated sound power to the equivalent power of a baffled piston with the same surface area and space-averaged vibration response [9].

$$\sigma_{rad}(f) = \frac{W_{rad}(f)}{\rho c S \langle v^2(f) \rangle} \quad (1)$$

where σ is the acoustical radiation efficiency factor, W_{rad} is the radiated sound power, ρ is the structure density (kg.m^{-3}), c the fluid celerity (m.s^{-1}), S is the vibrating surface (m^2), f is frequencies (Hz) and $\langle v^2 \rangle$ is the spatially average vibratory response ($\text{m}^2.\text{s}^{-2}$).

Campbell [9] recalls that the common method for determining the radiated sound power is the sound intensity scanning. The protocol for measuring the sound intensity scanning is given in ISO-9614-2. Hazelwood and Robinson [10] explain that, in an environment with poor echoes, it is possible to assume that the sound source energy is dissipated uniformly in all directions. As a result, the radiated acoustic power is equal to:

$$W_{rad} = \frac{4\pi P^2 r^2}{\rho c} \quad (2)$$

where P is the spatially average RMS pressure (Pa) and r is the spherical radius (m).

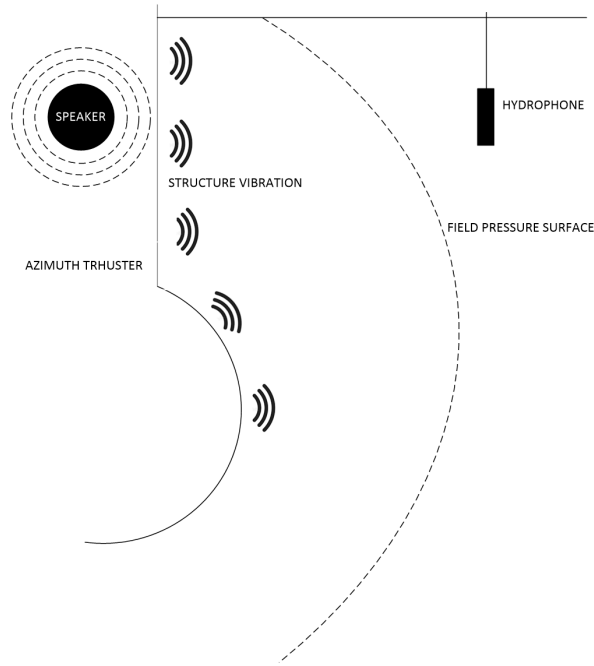


Figure 4: Hydrophone immersed near the structure to measure the radiated pressure field from the azimuth thruster

The acoustic power is related to the acoustic pressure measured on the surface of a sphere with radius r , which is the distance at which the measurement is made. In our experimental setup, only one measurement position was evaluated and we assume hemispherical radiation to account for the measurement position (see Figure 4). For a first experimental approximation of the radiated acoustic power, we also consider the hypothesis of an omnidirectional pressure field without reflection. From the surface velocity field and the radiated pressure measurements, the radiation efficiency factor in the range [100 - 1000 Hz] is estimated using Equation 1.

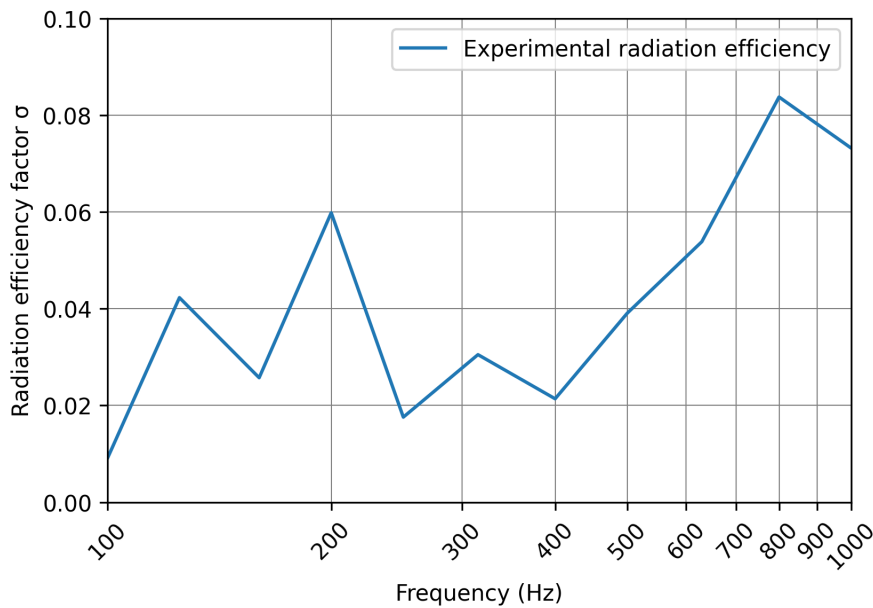


Figure 5: First experimental radiation efficiency factor estimation in the frequency band of interest

Hazelwood and Robinson [10] proposed an alternative approach for determining underwater acoustic power in a totally reverberant environment. This assumption is commonly employed

in tank testing and is well-documented in the literature. In our complex environment, it may be imperative to take into account reverberation, as sound is reflected by interfaces such as the sea surface, the seabed, and the harbour wall. Reverberation time data was not collected during in-situ measurements, therefore it is not possible to confirm this assumption without numerical simulation.

3. NUMERICAL APPROACH

Although our experimental approach allowed us to identify structural modes in the water of the azimuth thruster and to estimate an acoustic radiation efficiency factor, it is evident that conducting these measurements in a complex environment presents numerous constraints and limitations. Further investigation requires numerical modeling. The work presented below is a first step in this direction.

3.1. Structural wet modes

Studies have shown that the combination of acoustic radiation mode calculations and surface velocity measurements provides an accurate method for measuring the radiated sound power from vibrating plates [11]. A FE solver with a simplified numerical model of the structure is used to compute the wet modes. To meet the meshing requirement of 6 elements per wavelength for FE calculations and to compute the modes of the steel structure up to at least 1 kHz, we estimate the size of a tetrahedral element to be 0.07 m, resulting in 29,592 shell elements for the structural mesh. For the water volume, 1,764,791 acoustic elements of 0.2 m are used (see figure 6).

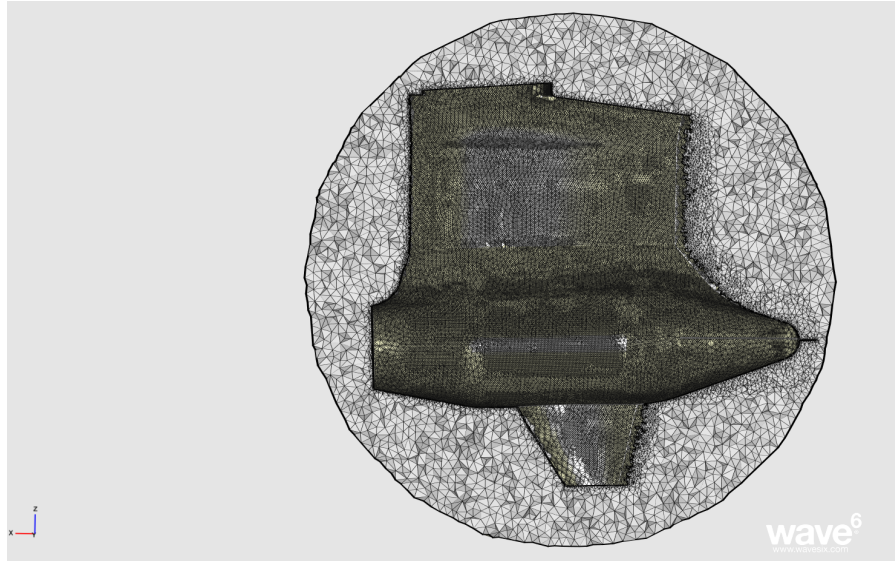


Figure 6: Structure and water volume mesh for coupling modes computation.

Modeling the coupled structure-water system can be done by solving a coupled structure-cavity system. This involves solving the following system [12]:

$$(\mathbf{K} - \omega^2 \mathbf{M}) \mathbf{x} = \mathbf{f} \quad (3)$$

where:

$$\mathbf{K} = \begin{bmatrix} \mathbf{K}_s & \mathbf{C}_{sa} \\ \mathbf{0} & \mathbf{K}_a \end{bmatrix}, \mathbf{M} = \begin{bmatrix} \mathbf{M}_s & \mathbf{0} \\ \mathbf{C}_{sa}^\top & \mathbf{M}_a \end{bmatrix}, \mathbf{x} = \begin{bmatrix} \mathbf{u} \\ \mathbf{p} \end{bmatrix} \text{ and } \mathbf{f} = \begin{bmatrix} \mathbf{f}_s \\ \mathbf{q}_a \end{bmatrix} \quad (4)$$

\mathbf{M} and \mathbf{K} are the mass and stiffness matrix of the coupled FE model, \mathbf{M}_s and \mathbf{K}_s are the $N_s \times N_s$ mass and stiffness matrix of the structure (N_s : number of structural dofs), \mathbf{M}_a and \mathbf{K}_a are $N_a \times N_a$

mass and stiffness matrix of the acoustic domain (N_a : number of acoustic dofs), u_s is the $N_s \times 1$ vector of the degrees of freedom of the structural FE model, p_a is the $N_a \times 1$ vector of the degrees of freedom of the acoustic FE model and C_{as} is the $N_a \times N_s$ acoustic-structural coupling matrix. f_s is a $N_s \times 1$ vector of the structural excitation while f_a is a $N_a \times 1$ vector of the acoustic excitation. f is the $N \times 1$ excitation vector for the coupled model where $N = N_s + N_a$ is the total number of dofs of the model. u is the $N \times 1$ vector of the degrees of freedom of the coupled FE model. The structural parameters and model properties are summarized in Table 2.

Table 2: Material properties of the structures, water and numerical model properties.

Density (structure)	7800 kg.m ⁻³
Young's modulus (structure)	210 GPa
Poisson's ratio (structure)	0.3
Density (sea water)	1025 kg.m ⁻³
Boundary conditions (structure)	No displacement (embedded) at the top of the structure
Boundary conditions (environment)	Free surface (representative of the water surface)

In this study, we are only interested in structural wet modes, i.e. we only consider the mass added to the structure by the fluid. For this purpose, it is sufficient to impose an arbitrarily high sound speed. A total of 2300 eigenvalues are computed in the frequency range [60 - 1170 Hz]. Verification and manual calibration are performed to achieve a realistic dynamic behavior of the structure. Considering the simplified surface model used, the calibration of the model was done by varying main parameters of the shell. However, due to the high modal density in this frequency range, the selection of modes corresponding to experimental measurements becomes challenging.

3.2. Preliminary results : radiated pressure field

The acoustic radiation was computed numerically using a FEM/BEM coupling technique. The dynamic behavior of the structure was calculated using FEM based on the experimentally measured vibration field and the previously computed wet modes. BEM was used to estimate the acoustic radiation in the water caused by the vibrations of the structure. The model included a free surface condition (water-air interface) above the structure to simulate real-world conditions. The pressure field measurement was performed by a single measurement placed at the same position as the hydrophone during the experiment. The primary objective of this initial numerical computation was to match the same experimental conditions and observe the results. This first approach allows obtaining a satisfactory numerical estimation of the pressure field compared to the experimental radiation factor within the band [160 - 500 Hz]. In the 125 Hz frequency band and the [630 - 1000 Hz] band, the numerical radiation efficiency is underestimated. These differences could be due to a variety of assumptions:

- Use of an inaccurate structural model (geometry, mass, stiffness, material);
- Neglect of seabed and quay edges.

In the study, the seabed and the harbor walls were modeled to answer the above questions. The results indicate that the walls have a minimal influence on the radiated pressure field when measured by only one hydrophone (see Figure 6). Therefore, the hypothesis of an omnidirectional field without reflection seems to be confirmed. Any discrepancies between numerical and experimental radiation factors would be due to the lack of data to build the structural model.

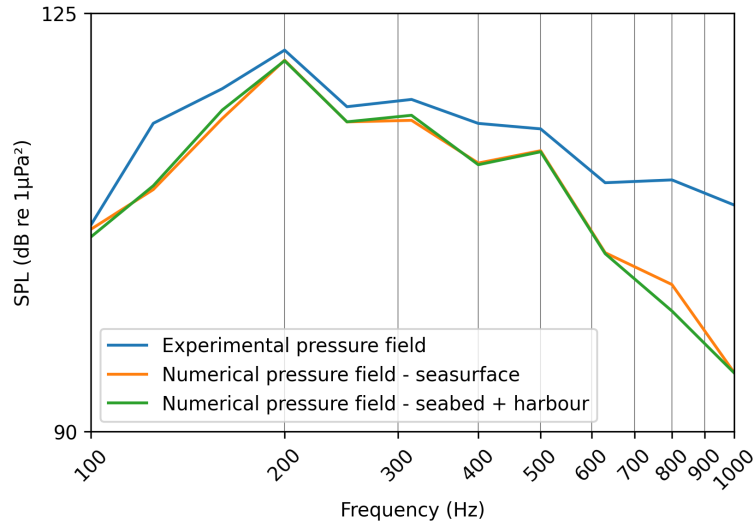


Figure 7: Comparison between experimental radiated pressure field and numerical radiated pressure field in the frequency band of interest with different boundaries conditions.

4. ANALYTICAL APPROACH

Another approach to characterize the vibroacoustic behavior of the structure is the use of analytical solutions. In the present work, we apply the formulas developed by Szechenyi for cylinders [13], in which no mode coupling selectivity is considered since the space lacks discrete modes for radiating cylinders in an infinite space. Although estimation using Szechenyi's formulas

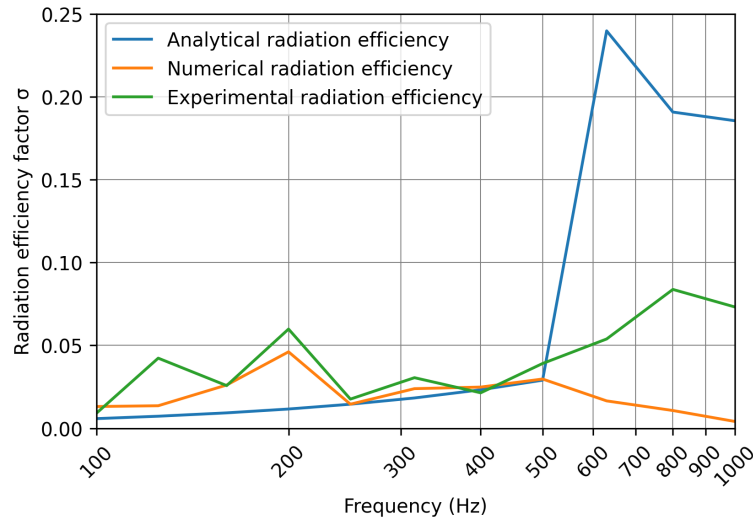


Figure 8: Comparison between analytical, numerical and experimental acoustical radiation efficiency factor for azimuth thruster.

provides an idea of trends, it is too far removed from numerical and experimental radiation factors to be used to predict the underwater radiated noise of an azimuth thruster.

5. FINAL COMMENTS AND CONCLUSIONS

Many methods are available to characterize vibroacoustic sources. This study has highlighted 3 methods for estimating the vibroacoustic behavior of an azimuth thruster: experimental, numerical and analytical. Although analytical approaches are fast and easy to use, the results

could be inaccurate when applied to model the vibroacoustic behavior of industrial structures. The experimental method provides insight into the real behavior of the structure, but it is subject to numerous sources of uncertainty due to environmental and industrial constraints. This method is generally complex and lacks flexibility. Numerical methods offer greater flexibility and the ability to test multiple parameters. However, their applicability depends on the proper calibration of the model with respect to the real system. Nevertheless, we were able to achieve a satisfactory correlation between experimental and numerical estimates of the radiated pressure field estimation within the frequency band of [160 – 500 Hz]. From these results and the determination of the radiation efficiency factor, it could be used with onboard measurements during sea trials to estimate the URN of the thruster.

This preliminary work will be completed by numerical modeling the vibroacoustic behavior of the structure, directly integrating the acoustic source. With the preliminary promising results, this characterization method will make it possible to estimate the vibroacoustic behavior of an azimuth thruster before testing. The final step is to understand the causes of structural vibrations in order to make accurate predictions in the early stages of ship design. Finally, the method used can be adapted to other sources of ship noise.

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