The use of numerical prediction tools inside the "TSI-Noise and Vibration Comprehensive Management" methodology to build "Silent Ships"

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Abstract

Due to its impact on marine life, the abatement of the Underwater Radiated Noise (URN) by ships has become the most outstanding novelty and difficult challenge that the Shipbuilding industry has ever faced. Therefore, the industry is trying to provide new solutions in order to comply with the new directives and requirements having been recently developed and promoted by the EU, Marine Institutions and the scientific community.

In the framework of AQUO project specific full scale URN measurements of a fishing vessel have been carried out and compared with the prediction performed for the same ship with VA-One. This paper will present the correlation between the prediction and the measurements, describe how these prediction tools can be used to evaluate and optimize abatement solutions, and finally, it will detail how these tools are used inside the "Noise & Vibration Comprehensive Management" methodology developed and currently used by the authors for the Dynamic and Acoustical Design focused on building "Silent Ships" in compliance with the most strict Underwater Radiated Noise Requirements.

Introduction

In the framework of the AQUO project, the need to predict underwater noise generated by vessels has been identified as a key point to enable the shipbuilding industry to comply with the forthcoming directives building quiet vessels. With the aim to cover this aim, TSI has developed a complete set of engineering activities in a Research Vessel mainly focusing on a numerical underwater prediction and full-scale measurements for validation.

Full-scale measurement campaign

The selected vessel to address the study is a research vessel of the CSIC (Spanish national re-search center). The main particulars of this vessel and main views are shown below.

- Total length = 24m.
- Breadth = 5.80m
- Draught = 2.60m
- Deadweight = 31.20 ton.
- Displacement = 170.52 ton
- Hull material = Steel

The characteristics of the main underwater noise sources are described below.

- Main Engines: 2x287kW Diesel engines rigidly mounted.
- Auxiliary engines: 2x 53kW Diesel-Gen set resiliently mounted.
- Propeller: 2 Fix-Propellers

The most relevant ones for this study are the main engines and the propellers.

The measurement campaign was specifically designed to record all the required data to feed the other work packages as well as to validate the new European underwater measurement standard proposed in AQUO project. To cover these aims, specific developments and TSI equipment adaptation were required with the result of the customised buoy shown in Figure 1.

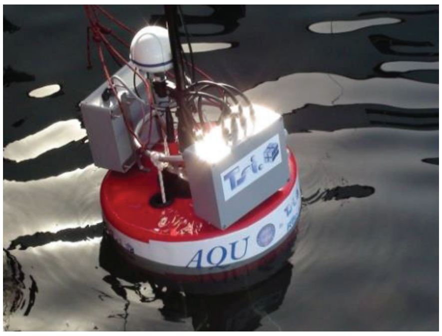

Figure 1: Customized buoy to measure the underwater raidated noise of the ship.

The set of measurements carried out during the project is was defined for a larger scope within AQUO but in particular, the study described in this paper used the data provided by the following measurements:

- Machinery vibration measurements: They were used to assess the dynamic forces caused by the engines and the contribution of the rest of the sources.
- Airborne noise: The noise in different points of the machinery room was measured to help estimate the airbone noise of the main engines and diesel gen-sets.
- Hull vibration: In a first stage these measurements were used to correlate the results of the model in the stern area after applying the dynamic force of the engine, and in a second stage, they were utilised to try assess the propeller underwater noise (see Figure 2).

- Underwater radiated noise measurements synchronized with GPS data: In order to correlate it with the results of the model.

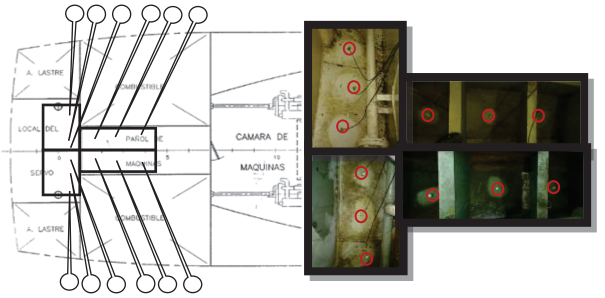

Figure 2: Location of the accelerometers for the hull vibration measurements.

Description of the vibroacoustic model

According to the required need to predict the noise generated by the vessel in low, medium and high frequency, two vibroacoustic models were developed, FEM-BEM for the low frequencies and SEA for medium and high frequencies. SEA models are valid once the modal density of the radiating plates are higher than 3. FEM-BEM models are used in the low frequencies as the SEA models are not accurate for such frequencies. However, in mid-frequencies these models requires a lot of computational resources and are really sensitive to the uncertainties of the properties of the structure (masses, thickness, …).

In both cases the load considered were the engine load, obtained thanks to the machinery vibration measurements, and the effect of the propeller in the hull, measured thanks to the vibration measurements carried out in the stern area of the ship above the propeller. Regarding this source, the propeller, there were a lot of uncertainties as there was no experimental data of the propeller noise. However, the most important underwater noise source for this vessel was the machinery, so a good correlation was still expected.

FEM-BEM + SEA model: As said before, for low frequencies a FEM-BEM model was used. Actually, it was a hybrid model. Below the waterline a FEM model was used, and the structure above the waterline was modelled by SEA plates. The main aim of this modelling approach was to save computational resources and yet modelling with enough accuracy. The dry mode shapes and eigenfrequencies of the FEM part were computed in an external software and imported to VA-one. The Figure 3 shows the model and Figure 4 some results.

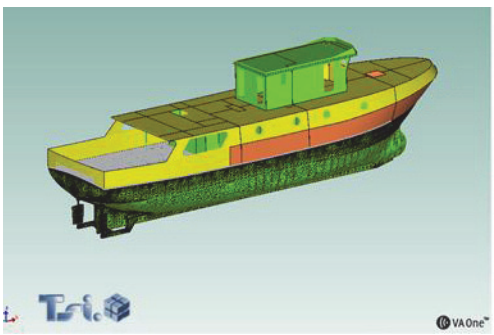

Figure 3: Overview of the FEM-BEM + SEA Model .

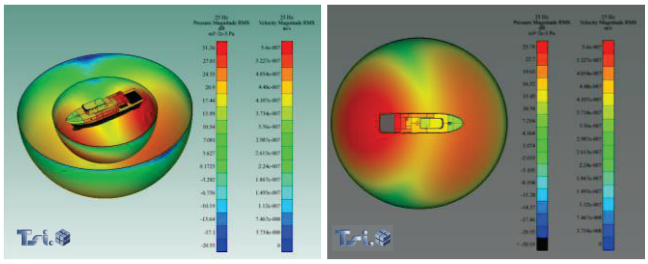

Figure 4: Some results of the FEM-BEM + SEA Model .

SEA Model: As said before, SEA model becomes accurate for those bands where the modal density of the radiating hull plates is larger or equal than 3. In particular, for this vessel this holds from 125Hz. However, due to the computational resources required no mode shapes above 63Hz could not be obtained, which is why the SEA model were used to compute the underwater acoustic signature in the 80Hz, 100Hz and 125Hz bands. As we will see later, the results for these frequencies were reasonably correlated with the measurement.

In order to model the sound propagation from the ship outwards, SEA cavities with null damping were used achieving reasonable transmission loss. This way, we avoid the uncertainty associated with the conversion of the noise levels measured into source noise level at 1m of the ship. The Figure 5 shows the model used and the Figure 6 show some results.

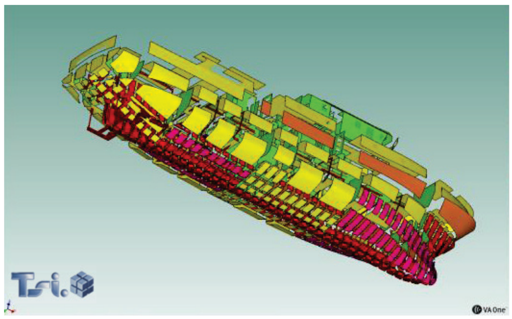

Figure 5: Overview of the SEA Model .

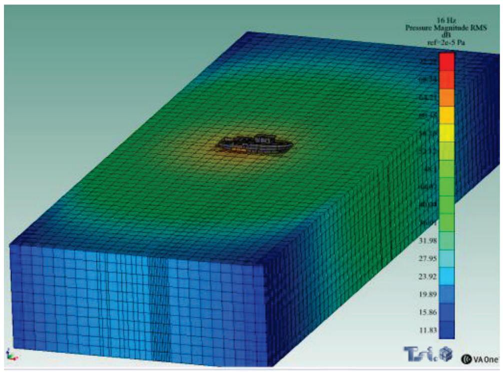

Figure 6: Some results of the SEA Model .

One important factor to be taken into account is the damping of the hull plates. In order to test their influence, two damping loss factor were considered:

- A constant damping loss factor of 1%
- An experimental damping loss factor of a representative hull plate.

Modelization of the noise sources

The airborne noise of the main machinery was modelled as an acoustic diffusive field in the SEA cavity of the machinery room taking advantage of the airborne noise measurements taken in the machinery room during the trials. This accounts for all the items located in the machinery room.

The structure-borne noise of the main engines was modelled using a FEM Model of their foundation (Figure 7). Thanks to this model, the transmission between the main engines and their foundation was properly characterized by the computation of their mobility. Indeed, thanks to this model, the necessary dynamic force to reproduce the vibration measured in the feet of the engines was computed.

The structure-borne noise of auxiliary engines was despicted due to the fact that they are elastically mounted and their rating power is much lower than for main engines.

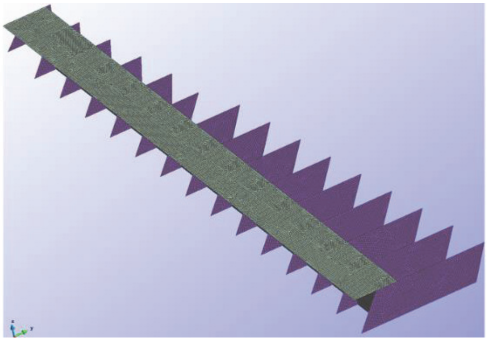

Figure 7: Details of the FEM model of the main engine foundation .

The rest of the machinery was modelled thanks to the vibration measurements performed on some of them and considering a representative mobility of a 50mm plate.

The propeller noise was estimated using the vibration measurements taken on the hull plates above the propeller and experimental transfer functions. Due to this fact, the biggest uncertainty of the model was this source. However, the propeller noise of this ship was not relevant in the range of interest compared to the machinery noise for this ship.

Correlation between measurements and the numerical prediction

The Figure 8 shows the correlation between the results obtained from the FEM+BEM model, SEA model and the measurement considering a constant damping loss factor. The results of the model presented in this figure are the average, post-processed in the same way as the time-signals recorded in the measurement, of the sound levels of a line of cavities to simulate somehow the data window that the followed measurement procedure defined in [1].

The Figure 9 shows the correlation between the results obtained from the FEM+BEM model, SEA model and the measurement considering the experimental damping loss factor.

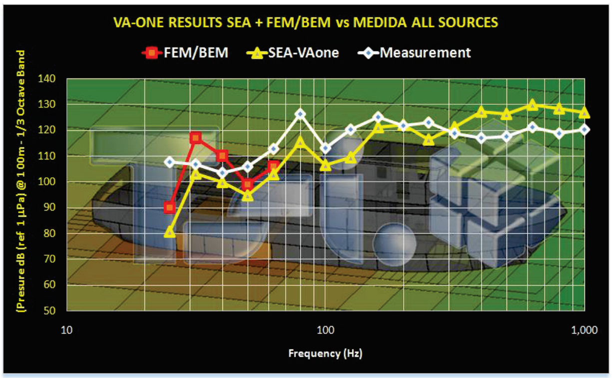

Figure 8: Correlation between numerical results and the measurement considering a constant damping loss factor for the hull plates.

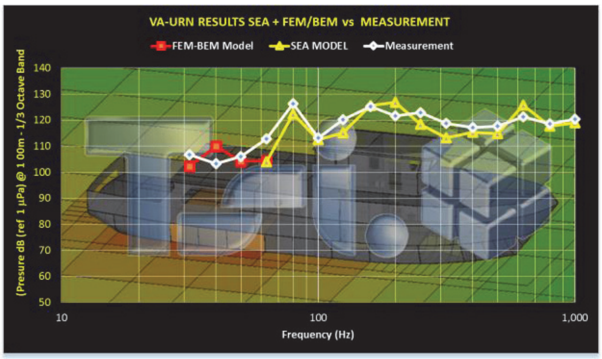

Figure 9: Correlation between numerical results and the measurement considering an experimental damping loss factor for the hull plates.

Clearly the figure shows that the agreement between the FEM+BEM Model and the SEA model is really good. This is an indicator that the SEA model works reasonably well at this low frequency probably thanks to the good characterization of the source by a FEM model of the foundation.

On the other side, the correlation significantly improves with the experimental damping loss factor. In addition to that, the correlation between the numerical results and the measurements are quite good with an average deviation of about 2-3dB.

Use of these tools inside the TSI-Noise & Vibration Comprehensive Management

Achieving low levels of underwater noise is a challenge for the shipbuilding industry but achievable, as we will see later, if the right actions are taken in due time. There is a clear principle in the vibro-acoustic design of a ship: it has to start at the beginning of the project and to continue through all its stages. TSI has developed a methodology to assist shipyards in the design and building of “quiet” ships. It consists of four phases:

- **Assistance in the specification of the main underwater noise sources** (in particular, machinery and propeller). Analysis of the vessel in terms of noise & vibrations with special emphasis on the effect of its underwater noise signature.

- **Numerical prediction** to verify the correct design of the ship. In particular, to avoid possible resonance phenomena, assure a correct design of the engine foundation, assure the correct insulation and predict the underwater radiated noise of the ship to propose cost-effective solutions whenever it is necessary.
- **Factory Acceptance Tests** to verify that the suppliers fulfil with the requirements imposed during the purchase stage. Specific tests in different stages of the building of the ship to verify its correct design.
- **Sea trials** to measure the underwater radiated noise of the ship.

The first block of activities is aimed at setting control mechanisms to the most important suppliers by means of proved dynamic and acoustic requirements so they can be included in the corresponding purchase specifications.

The second block of activities is made up of different prediction tools like the one shown in this paper (FEM/BEM + SEA models) to assure the correct design of what is the shipyard competence (ship structure, noise insulation, main machinery foundation,..). These tools can also assess the improvement associated to different solutions like the example below, where the effect of applying damping tiles was assessed.

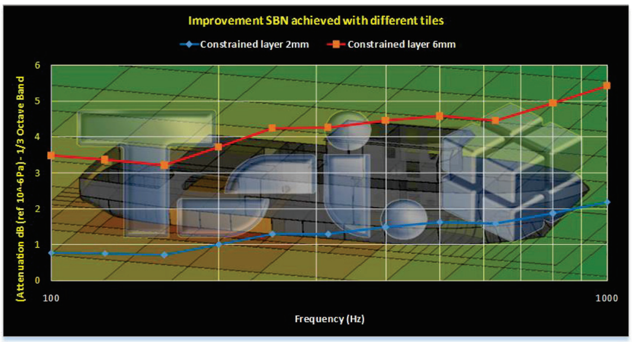

Figure 10: Assessment of the noise reduction achieved by different damping tiles.

During the third block of activities, different tests are defined and carried out to verify the fulfilment of the different noise sources with the requirements set in the first block of activities and the correct design of sensitive parts of the ship (engine foundation, insulation, etc...).

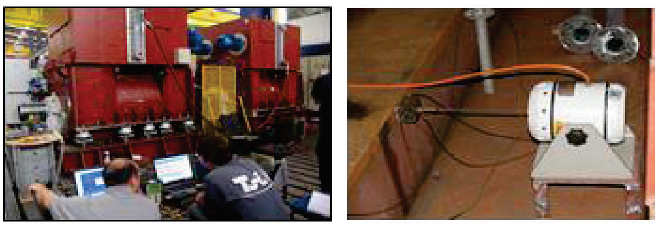

Figure 11: Overview of some factory acceptance tests

Finally, a measurement campaign is carried out to verify the compliance of the ship with the limits imposed.

Bibliography

[1] NR 614 of Bureau Veritas Underwater Radiated Noise October 2014

[2] Publio Beltrán, Eric Baudin et al. A comprehensive framework to address ship underwater radiated noise: from Bureau Veritas Class Notation to validation of numerical prediction tools.