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PREDICTING SOUND PRESSURE LEVELS ON A CRUISE SHIP

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Advanced computer programs allowing for numerical solving of mechanical problems are often used as design tools. Effective use of packages based on the finite element method allows for predicting the dynamical behaviour of the structure already at the design stage. This study is concerned with numerical analysis of the hull in a passenger ship whose FEM model is based on the technical specifications. Estimated values of acoustic pressure are compared with measurement results. Furthermore, the applicability of computer models to predictions of dynamic behaviour of complex structures is evaluated.

INTRODUCTION

Advanced computer programs allowing for numerical solving of mechanical problems are often used as design tools [2,10,12]. Effective use of packages based on the finite element method allows for predicting the dynamical behaviour of the structure already at the design stage [4,11]. Model accuracy has to be constantly verified, using a series of tests, to ensure reliability of the predictions. Thus numerical solutions have to be compared with already existing, even approximate, analytical solutions. The final verification procedure involves the comparison with measurement data in situ.

This study is concerned with numerical analysis of the hull in a passenger ship whose FEM model is based on the technical specifications. Estimated values of acoustic pressure are compared with measurement results. Furthermore, the applicability of computer models to predictions of dynamic behaviour of complex structures is evaluated.

1. SCOPE OF WORK

Analysis of vibrations and sound levels on a three-decked passenger ship was carried out. The ship has cabins for passengers and crew members, also the rooms housing ancillary equipment, such as fans, pumps, etc. There is one sun deck with the captain's bridge and passengers' cabins. There is also an upper deck, a bulkhead deck and a lower deck where passengers' and crew members' cabins are situated, as well as rooms housing the necessary equipment.

The actual layout of rooms and the way they are adapted are determined by vibration and acoustic parameters in the given region.

Underneath the lower deck is the platform deck with the engine room, a pilot house and rooms housing fans, pumps, power generators, and the like. Still beneath is the internal bottom of the ship. On that level we can find the main engine with the ancillary units, power generators, compressors, pumps, gear-boxes. Between the internal and external bottom of the ship is some free space where fuel tanks and ballast tanks are kept.

2. COMPUTER MODEL

The 3D FEM model developed here is based on detailed technical specifications. The hull is represented as a complex truss system, made of vertical, lateral and longitudinal beams which act as bearing elements, at the same time joining the ship's sides plates with floor plates. Depending on their function and position, truss reinforcements have different cross-section and profile structure. The strength of this structure is further improved through application of longitudinal and deck beams.

Longitudinal (T-profiles) run along the hull under the deck. They improve the torsional and bending strength. Deck beams are arranged laterally and improve the torsional rigidity. Decks are supported by pillars, i.e. bearing elements made of vertical tubes of variable cross-section (ranging from ϕ 82.5 x 8 to ϕ298.5 x 12.5 mm).

The model of the hull is presented in Fig 1. It consists of 2269 elements (beams and tubes) and has 1020 nodes, which corresponds to about 6120 DOFs.

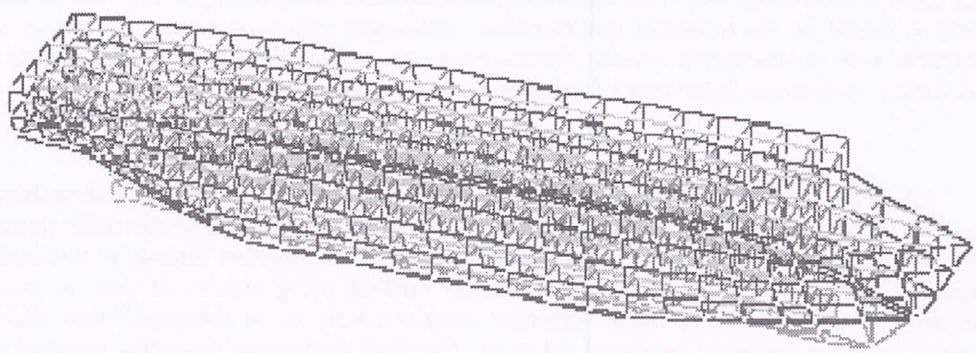

Fig 1. A truss model of the ship hull

The truss model is supplemented with the plating 9-14 mm thick and floor plates having the thickness of 5 mm on the sun deck and 8 mm on the platform deck and the ship bottom.

3. SELECTION OF BOUNDARY CONDITIONS

Several models are presented below, the main difference between them being the number and positions of subtracted DOFs. Boundary conditions are difficult to model as it is not easy to determine the actual conditions in the contact area between the shell plating and the water environment [3,8]. Several options of determining the number of DOFs in order to model the hull position on the water surface and in the shipway are presented in the study.

* Model with nodes at points of maximal rigidity

This model corresponds to the critical loading known as the "sagging moment'. In this model DOFs at the hull bottom are subtracted. These points are selected because this part of the ship is more rigid than the remaining portion of the hull. In these regions are floor plates

12 mm thick. The hull cross-section at that point is smaller than in the other hull regions. Rigidity of the structure depends on the type of profiles used for support in accordance with the design objectives.

* Model of hull support on the shipway

This model reproduces the conditions created by the shipway. Its distinctive mark is that DOFs are subtracted at equally distant nodes, every 10 m along the ship length. The aim of this model is to compute and experimentally verify the natural hull's frequencies already at the construction stage. Accordingly, the design engineer would be able to select the main engine so that its natural frequency would not coincide with that of the hull.

* Determining the boundary conditions in the stern section

In this model all DOFs in the stern nodes that determine node displacement and rotation are subtracted. Hence the model resembles a beam rigidly fixed at one end while the other is free to move.

* Model of a hull fixed on bars whose rigidity equals the value of water table deflection due to ship draught

The truss model is fixed on vertical beams whose rigidity is derived from the deflection produced by hull draught. These beams are sectioned into finite elements. The nodes can move in the vertical direction. That is an attempt to reproduce the conditions produced by ship displacement and draught while the aquatic environment is modelled as a compliant medium.

4. MAIN EXCITATIONS AND ANALYZSIS OF NATURAL FREQUENCIES OF THE HULL

There are several sources of noise and vibrations in the analysed structure. There are two sources of low-frequency vibrations, the main engine being the predominant one. Vibrations excited by the engine are imposed as concentrated forces at the point the engine is mounted [1]. Passengers' comfort and the strength of the ship structure depend on variable hydrodynamic pressure, being the exciting force. These vibrations are generated by water turbulence produced by the operating screw propeller. Variable pressure can be sensed in the stern part, and when yielded through structural elements it can be measured and felt even on the top decks. Variable pressure due to water movement caused by the operating screw propeller is imposed on the stern planking as a constant value 1.5 kPa (pressures at the stern 1.0-1.8 kPa were recorded during model tests in a Danish shipyard).

Apart from dynamic loading, it is necessary to consider the constant hydrostatic pressure resulting from the ship draught.

As it is extremely difficult to reproduce the real boundary conditions of the submerged hull, several methods of representation were testified. It is also considered how the boundary conditions impact on natural frequencies of the hull and on dynamic behaviour of the structure. The results were then verified by comparing the natural frequencies of the fixing options with measured values and also with results obtained for similar structures, as can be found in literature on the subject [5,9].

Results of the analysis of natural frequencies of hull vibrations are summarised in Table 2. In model 1 DOFs are subtracted at points of the maximal rigidity. DOFs in model 2 are subtracted in the same manner as when the ship is fixed on the shipway. DOFs in model 3 are subtracted in the stern part. Model 4 is fixed on vertical, elastic beams whose rigidity equals the value of water table deflection due to ship draught.

Table 1. Results of the analysis of natural frequencies of hull vibrations for several options of boundary conditions

Number of mode	Boundary conditions at points of maximal rigidity [Hz]	Boundary conditions as in the shipway [Hz]	Boundary conditions with DOFs subtracted in the stern part [Hz]	Model fixed on flexible /compliant beams [Hz]
Mode 1	0,98	0,11	0,09	0,06
Mode 2	2,8	1,23	2,98	2,28
Mode 3	3,84	2,56	3,7	3,8
Mode 4	4,2	4,72	4,42	4,59
Mode 5	5,8	5,23	5,8	5,8
Mode 6	6,13	6,63	7,9	6,4

5. COMPARING THE RESULTS OF NUMERICAL ANALYSES WITH THE MEASURED VALUES

Numerical analysis involves dynamic analyses in time and frequency domains. Thus obtained amplitude - frequency characteristics for each node are then surface- averaged over the analysed frequency bands. The spectrum of upper deck node vibrations (over the main engine) is presented in Fig 2, where characteristic frequencies of the engine (2.8 Hz) and the screw propeller (3.8 Hz) can be easily distinguished.

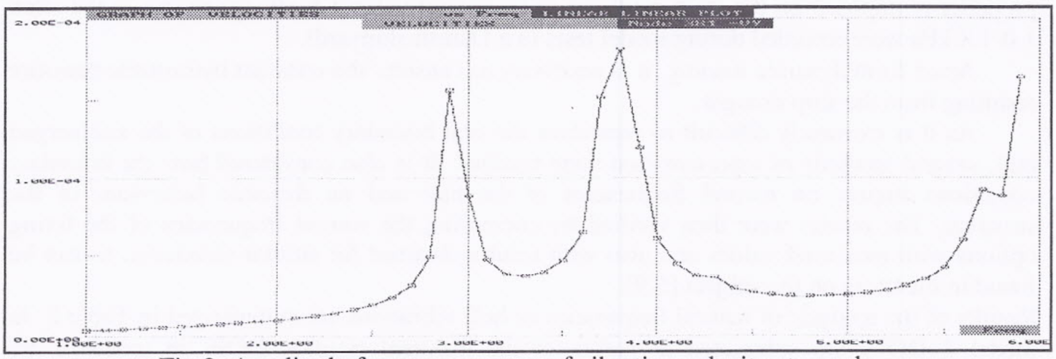

Fig 2. Amplitude-frequency pattern of vibration velocity at a node

The value of radiation efficiency of steel plates was assumed constant in the next stage of computations. Anticipated levels of the acoustic pressure on the upper deck can be predicted using the averaged results of vibration velocity measurements. Predicted distributions of acoustic pressure along the hull are presented in Fig 3. Beside the curves obtained from computer analysis there is also a curve representing the measured values.

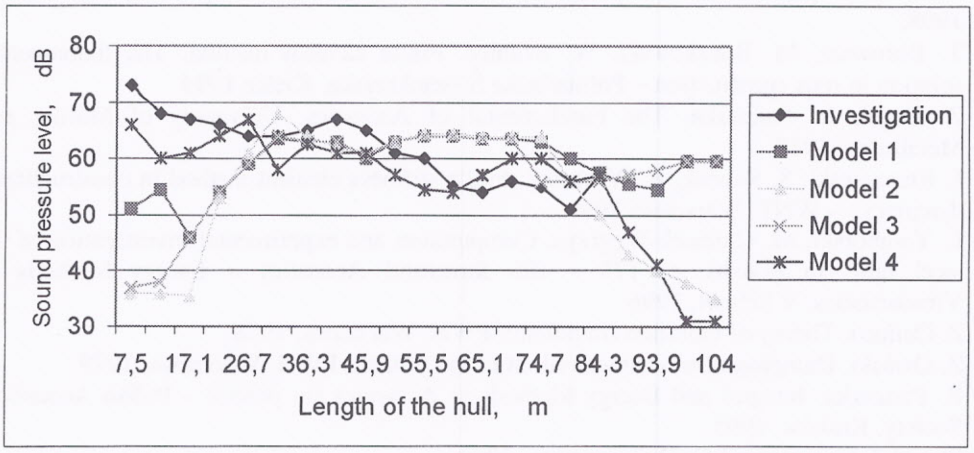

Fig 3. Acoustic pressure distribution on the upper deck.

These predictions follow from numerical analysis performed on models with different boundary conditions. The results obtained for Model 3 best agree with the measured data. Depending on the distance from the stern, the difference between the calculated and measured values of acoustic pressure ranges from 1 to 5 dB, while in the middle section it is as low as 3 dB. As far as other estimates are concerned, the most reliable predictions are for middle hull sections, between 25 and 90 m in the direction of the bow. In this case the difference between the predicted and measured values varies from 1 to 5 dB, depending on the point where it was measured. At the distance 7.5 to 25 m, this difference can be as high as 30 dB.

6. CONCLUSIONS

Accurate fitting of boundary conditions is of primary importance in FEM modelling. Boundary conditions and exciting forces ought to reproduce hull load combination due to ship maintenance and operation. Amplitude - frequency characteristics allow for determining the dynamic state of the chosen ship section for variable excitations and boundary conditions. Hence it is possible to alter hull designs, to choose the optimal drive systems and best dislocate the equipment, depending on their functions and vibration levels.

FEM model with boundary conditions corresponding to the shipway position allows for validation of the FEM model of the hull by comparing the data with results of measurements taken at the stage of ship construction. This model supplemented with boundary conditions that reproduce the actual operating conditions allows for fairly accurate prediction of vibro-acoustic parameters during normal operations at the full sea.

The error involved in predictions of acoustic pressure levels was the smallest when the hull is modelled as fixed on rigid beams whose rigidity equals the value of water table deflection due to ship draught. In relation to the measurement data the error does not exceed 3 dB over the major part of the hull length. This model allows for fairly precise prediction of dynamic parameters, which up till now were mostly determined using less accurate calculation techniques, or from measurements of completed structures.

REFERENCES

1. K. Adamowicz–Perska, M. Podfigurny, The elastic mounting influence on structural vibration propagation in hull, p. 13-20, Structural Acoustics in Ship Designing, Gdańsk 1998.
2. T. Borowicz, M. Buczkowski, W. Szaniec, Finite element method. The fundamental solution in rods construction – Politechnika Świetokrzyska, Kielce 1994
3. Z. Engel, R. Panuszka, The Fundamental of Acoustics. University of Mining and Metallurgy, 1980.
4. J. Kruszewski, S. Sawiak, E. Wittbrodt, Finite stiffness element method in constructional dynamics, – WNT, Warszawa 1999.
5. L. Tomborski, M. Gniewek-Wegrzyn, Computation and experimental investigation of the steel hull (in polish), p. 173 - 186 Structural Acoustics – Energy Methods in Vibroacustics, V School, 1996.
6. Z. Osiński, Theory of vibration (in polish), PWN, Warszawa, 1978.
7. Z. Osiński, Damping of mechanical vibration (in polish), PWN, Warszawa, 1979.
8. R. Panuszka, Integral and Energy Methods in Acoustics (in polish) – Polish Acoustical Society, Kraków, 1995.
9. Strand 7, User manual G+D Computing, 2000.
10. T. Uhl, W. Lisowski, Modal analysis - practical problems in (in polish), University of Mining and Metallurgy, Kraków 1996
11. G. Rakowski, Z. Kacprzyk, Finite Element Method in Mechanics Constructions (in polish) – OW Politechniki Warszawskiej, Warszawa 1993
12. J. Wiciak, R. Panuszka, Software for Calculations of Propagation of Vibroacoustical Energy (in polish). Bull. of Mechanics, T. 14, Z.2 p. 205-220, 1995.
13. O. C. Zienkiewicz, Finite Element Method, Arkady, Warszawa 1972.