Ocean Engineering 155 (2018) 310–323

Contents lists available at ScienceDirect

Ocean Engineering

journal homepage: www.elsevier.com/locate/oceaneng

Vibration analysis for the comfort assessment of a superyacht under hydrodynamic loads due to mechanical propulsion

Pais Tatiana *, Boote Dario, Vergassola Gianmarco

DITEN, Università di Genova, Via Montallegro 1, 16145, Italy

A R T I C L E I N F O

Keywords:
Comfort analysis
Finite elements models
Frequency response function (FRF)
Pillars
Tuned mass damper

A B S T R A C T

Yacht designers and builders are continuously looking for new solutions to reduce construction costs and to improve the quality and innovation of their vessels. In the case of superyachts over 30 m in length, performances are no longer a primary goal and the efforts of technical offices are mostly addressed on other aspects related to the aesthetic impact of the project and to the on board comfort. From this point of view, vibrations and noise represent a difficult issues to deal with for designers, both in the initial phase of the project, when it is necessary to have preliminary information about the response of the structure not yet defined, and during construction, in case some critical behaviors arose in any part of the structure. In this paper, the dynamic behavior of a 40 m motor yacht is investigated. The Vibration Velocity Spectra obtained by the numerical model of the yacht are compared with the value of level limits imposed by the ISO6954:1984. Different structural solutions are investigate to improve the comfort level in same area not in compliance with the rule.

1. Introduction

The commercial success of recreational boats of any dimension depends on several aspects related, on one side, to the objective, technical tasks and, on the other side, to more subjective aesthetical characteristics. Before the recent financial crisis, the very positive situation of yacht market allowed the designers to exercise all their creativity giving rise to as innovative as disputable projects. Beyond the related commercial risk, this philosophy had the objective to attract potential owners and going over the many passionate competitors. In this perspective, however, the on board comfort hold a secondary position, often relegated to the personal feeling of the designer who didn't worry at all to sacrifice it in advantage of other more remarkable performances (Boote et al., 2013). With the rising of the 2008 global crisis, the competition in the superyacht field was further exacerbated in this way, also by troubling minor shipyards while confirming the prestige of other more experienced ones that continued to focus on yacht quality and reliability. This was particularly evident for large motor yachts, for which, due to their high value, design tasks should be achieved with great attention to any detail influencing the final quality of the product or/and the owner's approval. While the innovative technical aspects were dissected in all perspective, onboard comfort remains a matter to be investigated (Moro et al., 2013). The first regulation for verify the conformity of merchant ships crews and later the definition of class notations for passenger ships dates back to 1984, from which the term ISO 6954-1984. The latest review occurred on 2000, so that the code is named ISO 6954-2000. The value limits imposed by the regulation are very restrict and given the objective difficulty in making any change to the dynamic behavior of hull structure after construction, it is extremely important to perform FEM predictive analyses to identify the natural frequency of the hull and of local structures, such as decks and bulkheads, and then their response to exciting loads induced by propellers, engines and waves (Pais et al., 2017). In this paper, the dynamic behavior of a 40 m motor yacht has been investigated, by using the finite element method (FEM). An investigation on the most influencing excitation sources, as well as the influence of the main simplifications adopted in the numerical model, have been also carried out (Biot et al., 2015). The frequency response analysis have highlighted different living areas not complying with ISO 6945:1984. The authors have prefer to use the ISO 6954:1984 because this rule allows verifying the response of the structure at each excitation source. In order to reduce the vibration peak level that would compromise the habitability onboard, three different structural solution have been identified and tested; a thickness increase in the critical areas, the use of stiffened bars and the installation of Tuned Mass Damper (TMD) device (Pais and Boote, 2017).

* Corresponding author.
E-mail address: tatiana.pais@edu.unige.it (P. Tatiana).

https://doi.org/10.1016/j.oceaneng.2018.02.058
Received 6 September 2017; Received in revised form 1 February 2018; Accepted 21 February 2018
Available online 7 March 2018

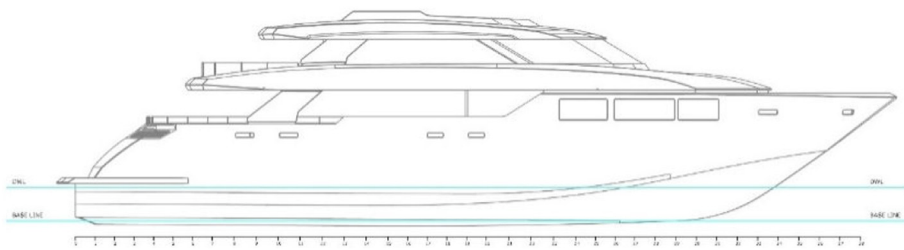

Fig. 1. The motor yacht assumed as a study case.

2. Numerical model

The vessel assumed as a study case is a particular type of craft named "cabin cruiser", 40 m long and 8 m wide in aluminum, composed by large internal areas distributed on three decks (main, upper and sun deck). Generally, these crafts are subdivided in internal and external areas. The internal one is composed by two living spaces: the first, called "*daily*", includes the main saloon, the kitchen and the steering console; the second one is conceived for nighttime, so cabins and toilets characterize it. The external area is provided by a large aft cockpit and an area to enjoy the sun. Above the superstructure, the craft has an open space called "*fly-bridge*" where is placed an equipped external zone and others steering command console (generally it represents the helm position). In Fig. 1, an illustrating picture of the boat is reported.

The structural modelling of this vessel is the most challenging part of any efficient method for vibration analysis prediction; it is very important that structures are reproduced clearly and detailed in order to have an accuracy of results in vibration terms (Boote et al., 2013). By the way, the real structural scheme of the yacht is extremely complicated to be reported in a FE model, hence it necessary to use some basic approximations, that should be carried out in order to have an optimal compromise between reality and the study case. Another aspect that have to be taken into account in this kind of investigation is to achieve an adequate ratio between result reliability and computational time and costs. Being the vibration global analysis the aim of this study, it is important that the structural hierarchy is respected (Asmussen et al., 2001). The principal reinforcements (transverse and longitudinal) and all the principal structure as decks and bulkheads should be modelled in a truthful way as better as possible because these structures influence the frequency response. The secondary reinforcements as longitudinal stiffeners are modelled in a truthful way to have accurate results. The local reinforcements (grate bars, brackets) are neglected because they do not influence results; however, they were take account in the weights distribution as it will be discuss forward. Once the modelling accuracy is established, the discretization of model is performed. As first step, the complete 3D cad model of the hull provided by the shipyard is imported in the pre-processor software. For the dynamic analysis, the fineness of the mesh is decisive (Moro et al., 2013). In principle, the minimum number of grid points to capture a wave shaped deformation is five, but in practice, this is the double (Klyukin, 1963). Any FE model should be generated according to such simple consideration, recalling the fact that larger elements of the mesh grid make the model stiffer and that by limiting the number of grid points a reduction of modelling and calculation time may be obtained. In this study, the dimension of the mesh is defined starting from the equation of flexural vibrations of a thin plate (Corradi, 2012). In this study, six-grid point are chosen to represent the wavelength and with a shell elements dimensions of 300 mm it is possible obtained a wavelength correspond to frequency of 35 Hz. To create a complete model of a superyacht is very difficult, so the principle shell element dimensions is 300 mm, although small variations about the nominal value are possible in the model. The mesh was created using shell elements for plating and main reinforcements such as keelsons, floors and girders. For secondary stiffeners simple beam elements have been preferred to keep the model dimension as low as possible. The yacht has a typical propulsion system for this kind of units, in the specific there are two main diesel engines with 2580 kW power, 3.5 t weight each and the mass moments correspondent are: $J_{xx} = 221$ kg m^2, $J_{yy} = 553$ kg m^2, $J_{zz} = 221$ kg m^2. The engine system has been modelled by using punctual masses, where their weight is described by a 0-D element located in the engines center of gravity. Engines are connected to the hull by resilient mountings, which are located in four different areas in the basement in order to better distribute the engine's weight. The resilient mounts have been modelled by spring linear elements in order to take account to stiffness constants in the three directions. The mounts are T35 HA Sh(55) and the stiffness constant of mounts is the same in all the three direction, i.e. 2500 N/mm (Vulkan, 2014). Dealing with the rigid link between engine and resilient mountings, four RBE2 elements is chosen, which allow to rigidly transferring the inertial forces of the mass with six degrees of freedom. The same approach have been used for linkage two generators at the base. In addition, the shaft line, the brackets and the bearing support of the shaft have been modelled with beam elements. Instead, the propeller has been modelled adding a punctual mass and relative moments of inertia. Its value have been provided by the shipyard and it has been corrected in order to take into account the added mass, i.e. an addition of the 50% of the propeller mass. As mentioned before, the machinery weight, the propulsion system, propellers, generators and engines have been modelled as concentrated masses. At this point, the model has a weight not corresponding to the lightweight ship condition because the weight of the all furniture, insulations, plants, floors, etc. lacks. The shipyard, for the part of the deck inside (for example saloon, rooms etc.) has suggested the value 100–130 kg/m^2 and for the deck outside about 40 kg/m^2. These kinds of loads cannot be treated as concentrated masses, but it is necessary to use uniform distributions, therefore "non structural mass" command is used. The nonstructural mass defined on the property entries is mass that is added to the structure in addition to the structural mass from the elements. The total weight of the lightweight ship condition is 214 tons. The loading condition of this study case was not the lightship one, but 6 tons of liquid in the tanks must be added. Moreover, the liquid is distributed in 2 tons in stern tanks and 4 tons in bow ones. In order to achieve this distribution, non-structural masses have been used.

Compared to other vehicles, ships are more complicate, because they work in a fluid that leads high inertial effects due to a high specific weight of the fluid. When the ship hull vibrates in the surrounding water, pressure fluctuations are excited. This effect could be simulated by adding a mass of water that cooperate with the structure, called added mass (Korotkin, 2007). To take into account the added mass in the numerical model, the Nastran card called *MFLUID* is used and it refers to the Boundary Element Method (Sauter and Schwab, 2010). This function is defined as the properties of an incompressible fluid volume generating a virtual mass matrix. This command can consider the presence of internal (tanks) or external (added mass) fluids.

To solve a vibration problem on ship structures, a high level of uncertainties is associated to the total damping forces governing the amplitudes of oscillations of the structure (De Silva, 2007). Effects of energy dissipations are usually accounted by a damping coefficient ξ, i.e. the critical damping. It is define as the ratio between the actual damping c and the critical damping c_r: