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Benchmark study of slamming and whipping

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ABSTRACT

Throughout the maritime world, considerable efforts have been spent on predicting loads associated with slamming. Up to now, little attention has, however, been paid to the accuracy of the translation from these loads to the structural responses. An important reason for this is that, in general, it is assumed that the uncertainties in the modeling of the hydrodynamic properties are larger than those related to the structural responses. To address this topic, the ISSC 2012 Dynamic Response committee, performed a benchmark study. The goal of this benchmark was twofold: on the one hand, the degree of variation in estimates produced by different methods and organizations was revealed; on the other hand, the deviations of the analyses were investigated by comparison with responses measured during model tests. From the results presented, it may amongst others be concluded that the shapes and frequencies of the two and three node, dry and wet and horizontal and vertical flexural vibration modes determined by the participants, were well in line with experimental results for four of the six participants. Computations considering an impulse induced by a regular head wave showed significant differences between the experiment, the different participants, and applied methods.

1. Introduction

Ships sailing at high speeds or in harsh environment can experience slamming loads. These loads occur because either the ship impacts the wave or the wave impacts the ship. Here the relative angle and velocity between the wave surface and the ship hull at the point of impact as well as the duration of the impact are important parameters. The classical explanation for the high loads is the sudden acceleration of the fluid close to the interface. Particularly in case of ships of 200 m and longer, slamming loads can result in a transient dynamic structural response of ships called whipping. Whipping vibrations increase both the extreme and fatigue load effects occurring in ships. Vertical bending moments induced by the combined effect of wave bending and whipping can be double that of the former. This was for instance shown by Zhao et al. (2004). During a period of four years, extensive experimental research on slamming induced whipping responses of large ships was done at the model test facilities of the Marine Technology Centre in Trondheim. In total, 80 different bow and stern geometries were tested. It was found that the whipping component of the vertical bending moment in large cruise ships had a magnitude of 30–150% of that of the wave component, in extreme wave conditions. For a 294 m long containership of newer design trading in the North Atlantic, Drummen et al. (2008a) showed that the wave-induced vibrations contributed approximately 40% of the total fatigue damage. Results were obtained using a 6.5 m long flexible backbone model of the ship. Using a similar setup, Storhaug (2007) showed an increase of about 25% for a 300 m long bulk carrier in World Wide trade. Here it should be noted that wave induced vibrations cover both vibrations due to whipping and springing. Springing is the steady-state resonant vibration due to continuous wave loading.

Throughout the maritime world, considerable efforts have been spent on predicting loads associated with slamming. For instance Kapsenberg and Thornhill (2010) developed an efficient and accurate method for predicting slamming loads for ships in waves. The method is based on momentum theory and accounts for the pile-up effects due to the immersing bow as well as the draft dependent static bow wave. The approach is tuneable to specific ship characteristics, but it captures the effect of different wave conditions and different headings well. Tuning can be done using a dedicated set of experiments, either in the towing tank or by CFD. Tuitman (2010) coupled the hydrodynamic and the structural model using generalized modes. He determines the slamming force using two different 2D methods, the Generalized Wagner model and the modified Logvinovich model. Kapsenberg (2011) presents a thorough review of the literature published on the problem of ship slamming in waves.

With the work regarding the prediction of slamming loads ongoing, little attention has been paid to the accuracy of the

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translation from these loads to the structural responses. An important reason for this is that, in general, it is assumed that the uncertainties in the modeling of the hydrodynamic properties is larger than those related to the structural responses. To address the uncertainties in predicting structural responses, the ISSC 2012 Dynamic Response committee performed a benchmark study. The goal of this study was twofold: on the one hand, the degree of variation in estimates produced by different methods and organizations was revealed; on the other hand, the absolute error made in the analyses was investigated by comparison with responses measured during model tests.

2. Benchmark setup

There were six participants: two research organizations (Marin and TNO), two class societies (Germanischer Lloyd and Indian Register of Shipping), one university (Norwegian Technical University), and a consulting company (The Glosten Associates). The benchmark was blind and consisted of three different stages. Not all participants delivered results for each stage. The tasks for each stage, as well as the results, are discussed in the subsequent sections. Participants were free in choosing methods for obtaining the results. The used methods are also described in the subsequent sections.

In order to investigate the absolute error, use was made of results from tests performed at MARIN with a model of a 173 m long RO/RO ferry (see Fig. 1). The hull's flexibility was accounted for by transversely cutting the model in several segments and connecting these segments by means of a backbone (i.e., Drummen, 2008b).

The main particulars of the ferry at full and model scale are shown in Table 1. The scale was 1:36. The data was provided by Cooperative Research Ships (CRS). In CRS, MARIN brings together a group of companies with a common interest in non-competitive research.

The aluminum circular backbone of the test model had a diameter of 0.110 m and a thickness of 0.005 m. Each of the hull segments was connected to the beam through two bulkheads, one at the forward part of the segment and one at the aft part. The beam was instrumented with strain gauges to measure, amongst others, the vertical bending moment in the different sections. The bow was built as a separate segment and connected to the forward hull segment through a six-component force transducer. This bow segment was instrumented with 23 pressure gauges to measure the detailed pressure distribution and six accelerometers to measure the local vibrations. Of the pressure gauges, five were located on the centre line of the model, 17 on the starboard (windward) side, and one on the port (leeward) side. Ten accelerometers were fitted inside the model, one laterally and one vertical accelerometer in the centre of each segment.

For the first stage of the benchmark, participants were asked to determine the shapes and frequencies of the two and three node, dry and wet, and horizontal and vertical global flexural vibration modes of the test model. The next step in the benchmark was to apply analytical yet realistic pulses to the numerical model. The final task was to predict responses, given an experimentally measured wave.

3. Stage 1: modal response

3.1. Experimental data

As mentioned in the previous section, for the first stage of the benchmark, participants were asked to determine the shapes and frequencies of the two and three node, dry and wet, and horizontal and vertical global flexural vibration modes of the test model. At this stage, participants were provided with details of the geometry of the model, including electronic hull description, locations of cuts, dimensions of the backbone, and mass distribution.

The modal parameters of the physical flexible backbone model were determined by MARIN using stochastic subspace identification (Van Overschee and De Moor, 1996). The parameters of the two and three node global vertical flexural vibration modes are given in Table 2.

The parameters in water were determined by performing hammer tests in still water. For determining the dry parameters, the fully instrumented model was suspended in air in a soft spring system. The precision error (Coleman and Steele, 1989) of the wet modal parameters was reported by MARIN to be very small. The single 95% confidence interval of the mean value of the natural frequencies was less than 2%. Due to the spring system, the uncertainties were larger in air, particularly for the three node mode. Although specific numbers were not given by MARIN it was noted that uncertainties related to determining damping values are larger than those related to the natural frequencies and mode shapes.

3.2. Methods

Participants were free to use a two or a three dimensional model to define the properties of the structure. Both the shapes

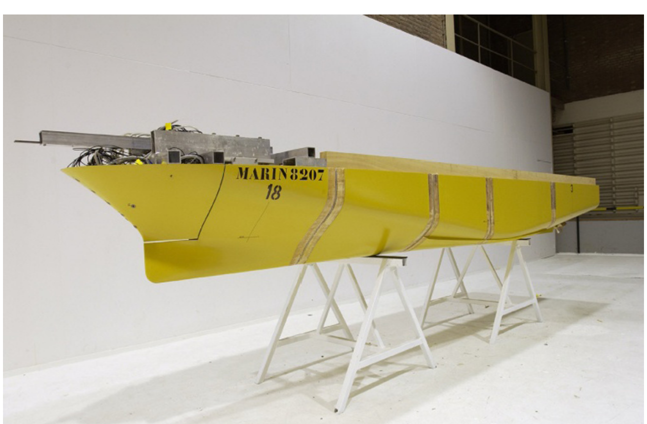

Fig. 1. Model of a 173 m long ferry, courtesy of CRS.

Table 1
Main particulars.

Main particular	Value at full scale	Value at model scale
Length between perpendiculars	173.0 m	4.81 m
Breadth at the waterline	26.0 m	0.72 m
Draft	6.3 m	0.18 m
Displacement	15721 ton	329 kg

Table 2
Modal parameters of the first two global vertical flexural vibration modes.

Modal parameter	Value
Natural frequency wet two node vibration mode	5.1 Hz
Natural frequency wet three node vibration mode	11.8 Hz
Natural frequency dry two node vibration mode	7.1 Hz
Natural frequency dry three node vibration mode	17.7 Hz
Damping ratio wet two node vibration mode	0.8%
Damping ratio wet three node vibration mode	0.7%