Ships and Offshore Structures

ISSN: 1744-5302 (Print) 1754-212X (Online) Journal homepage: https://www.tandfonline.com/loi/tsos20

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To cite this article: Ivan Ćatipović, Maro Ćorak, Joško Parunov & Neven Alujević (2018): Seakeeping experiments on damaged ship, Ships and Offshore Structures, DOI: 10.1080/17445302.2018.1559911

To link to this article: https://doi.org/10.1080/17445302.2018.1559911

Published online: 25 Dec 2018.

SHIPS AND OFFSHORE STRUCTURES
https://doi.org/10.1080/17445302.2018.1559911

Seakeeping experiments on damaged ship

Ivan Ćatipović, Maro Ćorak, Joško Parunov and Neven Alujević

Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Zagreb, Croatia

ABSTRACT

The aim of this paper is to present results of seakeeping experiments performed on a damaged ship hull and to compare them with numerical analysis results. The assumed damage opening in the ship hull, i.e. the damage extent, is determined based on probability distribution functions for damage size and location as specified by the Marine Environment Protection Committee. Two different sets of experiments are conducted in a large commercial towing tank. Firstly, rigid body motions of damaged and intact ship hulls in tow in head seas with small forward speed are measured for irregular waves. Secondly, ship motions and vertical wave bending moments are measured on the segmented model in regular waves. Experimental results are compared with the linear 3D Boundary Element Method using Hydrostar software. Satisfactory correlation of computations and experiments is achieved for ship in tow while discrepancies are found for the segmented model. An important finding of this experimental campaign is that measured vertical wave bending moments of the damaged ship considerably exceeds those of the intact ship.

ARTICLE HISTORY

Received 9 October 2018
Accepted 11 December 2018

KEYWORDS

Damaged oil tanker; seakeeping; tow; vertical bending moments

1. Introduction

Despite advances in navigation systems a large number of ship accidents continue to occur that cause loss of cargo, environmental pollution and even loss of human lives. Therefore, the International Maritime Organization (IMO) in its Goal-Based Standards (GBS) emphasised that ship, besides for the regular intact condition, should be designed to withstand loads in specified damaged conditions as well (IMO 2004). In the past several decades, various methods with different levels of complexity have been developed for the prediction of wave-induced ship motions and loads as described by the Committee I.2 Loads of the International Ship and Offshore Structure Congress (Hirdaris et al. 2014; Temarel et al. 2016). These methods have been developed in the first place for the intact ship condition, although they could be used for the damaged ship hull as well. In the case of an accident, such as collision or grounding, the ship stability could be reduced, the draught will increase and the ship may heel and trim. In such circumstances, the nonlinear time-domain simulations usually provide more detailed and more accurate results compared to the linear frequency domain methods. In other words, effects like varying of average floating position due to progressive flooding of internal compartments cannot be taken into account by linear frequency domain methods so the nonlinear time-domain seakeeping methodology needs to be employed. A nonlinear time-domain simulation method for the prediction of large-amplitude motions and dynamic global wave loads of a Ro-Ro ship in regular oblique waves in intact and damaged conditions is presented by Chan et al. (2002, 2003). Lee et al. (2007) employed a time-domain theoretical model based on the Bernoulli equation modified by a semi-empirical coefficient in order to study the motion of a damaged ship also considering the effects of compartment flooding. Furthermore, a mathematical model in time-domain for motions and flooding of ships in a seaway is described by Santos and Gedes Soares (2008). Hydrostatic forces are calculated over the instantaneous wetted surface taking into account the ship's motion while the radiation, diffraction and excitation forces are calculated for different even keel waterlines covering a range of mean draft variation resulting from flooding. The amount of water that enters each compartment in any given time step is calculated using a hydraulic model based on the Bernoulli equation. Numerical efficiency of this method was recently improved by introducing adaptive meshing by Rodrigues and GuedesSoares (2017).

Despite the advantages of these more advanced methods, linear methods like Strip Theory (ST) or 3D Boundary Element Method (3D BEM) can be convenient for seakeeping analysis of damaged ships, especially if it is assumed that the ship has already reached a stable equilibrium position around which it oscillates in waves. In this case, two practical approaches for analysis of ship motions and global wave loads can be used, as proposed by Downes et al. (2007). The first method is the added weight method, whereas the second one is the lost buoyancy method. In the former approach, it is assumed that the mass of the flooded seawater becomes an integral part of the ship mass and moves with the ship. The later approach is the lost buoyancy method which assumes that the structure of damaged tanks and all of their contents are removed from the vessel. Parunov et al. (2015) compared the two methods of added weight and lost buoyancy for the assessment of vertical wave bending moments (VWBM) of a damaged oil tanker in head waves, using the 3D BEM code Hydrostar (2011). The predicted result showed larger values for added weight and lower values for the lost buoyancy method in comparison with the

CONTACT Joško Parunov jparunov@fsb.hr

intact ship response amplitude operator (RAO). A more extensive comparison was performed by Jafaryeganeh and Guedes Soares (2018) using WAMIT software. The comparison showed the added weight simulation results in an adequate estimation of the maximum value of the vertical shear force as well as the maximum value of the bending moment. However, in the case of small wave loads, it does not provide accurate predictions. This is especially true for the horizontal bending moment. The lost buoyancy approach provides adequate predictions for the maximum values of the horizontal bending moment but underestimates the maximum values of the vertical shear force and vertical bending moment. Due to simplicity of its implementation, the added weight method is usually used for the seakeeping assessment of the damaged ship in linear analysis. Thus, Folsø et al. (2008) used the added weight method by employing linear 3D BEM for the seakeeping analysis of the damaged oil tanker sailing in the full load condition with different heading angles and various speeds. In the case of the flooded ballast tank in the midship area, they obtained RAOs of the VWBM larger than those evaluated for the intact condition. By using linear ST, Lee et al. (2012) also obtained larger VWBM for a damaged warship, confirming by comparison with the seakeeping experiments that the linear ST usually overestimates measurements for both intact and damaged ships.

After an accident, a damaged ship should be removed from the crash site. If the damaged ship loses the ability of self-propulsion, salvage is usually achieved by towing. In such a case, the total resistance of the damaged ship in tow could be an issue (Bašić et al. 2017). In such circumstances, the ship motions are influenced by the towline besides the excitation of the sea waves. In case of flooding, the flooded water exerts additional loads, i.e. sloshing and thus changes the usual behaviour of the ship at sea. At the same time, the dynamics of the flooded water are affected by the ship motions. Jia and Moan (2012) concluded that the effects of sloshing on the vertical bending moment are small, except in beam seas, while the effect of sloshing on the horizontal bending moment is large, especially in beam seas. Although the problem could be solved by BEM, the multimodal approach shows clear advantages, especially in terms of computational speed compared to BEM.

The goal of the present study is to experimentally investigate seakeeping behaviour of the Panamax tanker in both intact and damaged conditions during the towing procedure with small forward speed. Two sets of experiments are conducted in order to analyse rigid body motions, towline forces and vertical wave bending moments for intact and damaged ship hulls. In the first experiment, the model was in free drifting configuration and three ship motions (heave, pitch and roll), as well as the force in towline, are measured for the intact and damaged ship in irregular waves. Furthermore, the yaw motion is measured for the damaged ship condition. In the second experiment, the model was restrained in the horizontal plane and vertical ship motions (heave and pitch) and VWBM are measured in regular waves in order to obtain RAOs. Experimental results are compared with the numerical rigid body motions and VWBM RAOs. The considered response spectrum is based on Tabain's (1997) wave spectrum, which is conveniently used in the Adriatic Sea, while RAOs are calculated by the 3D panel code BV Hydrostar, which has an option to take into account linearised influence of the towline stiffness for the case of rigid body motions.

A specific feature of present experiments and contribution to the state-of-the-art is primarily the fact that experiments are carried out in a large commercial towing tank (L = 276 m) using a relatively small model scale (1:28.8). Previous experiments on damaged ships were performed in 'university' towing tanks using a much larger scale of 1:50 or even 1:100, as may be seen from the literature review presented in Section 2. Also, this is the first experimental campaign on a damaged oil tanker, since experiments published so far were performed on damaged RO-RO ships and warships.

The present paper is organised as follows. Firstly, a literature review of previous seakeeping experiments on damaged ships is presented in Section 2. Description of the towing tank and models used in the present study is given in Section 3. In the same section, the damage extent chosen for the study, which is based on the Marine Environment Protection Committee (IMO 2003) probabilistic model, is elaborated and justified. In Sections 4 and 5 experimental results are presented and a comparison with the numerical analysis is provided.

2. Literature review of previous seakeeping experiments on damaged ships

Various researchers have been conducting model tests on different types of damaged ship hulls. Korkut et al. (2004) performed motion tests in both intact and damaged conditions on regular waves of a Ro-Ro ship with the ship to the model ratio of 1/125. In the same Towing Tank of the University of Newcastle upon Tyne (which is 36 m in length, 4 m wide and has a water depth of 1.2 m) Korkut et al. (2005) have performed systematic measurements of global loads acting on a Ro-Ro model at zero speed.

To evaluate the accuracy of the theoretical model developed by Lee et al. (2007), model tests have been carried out at the Korean Research Institute of Ship and Ocean (KRISO) in a wave basin which is 68.8 m in length, 37.2 m wide with a water depth of 3.4 m. Three different damaged conditions of the Ro-Ro model are studied with the ship to the model ratio of 1/40. The testing is conducted on both regular and irregular waves without the forwarding speed.

Lee et al. (2006, 2012) conducted an experimental campaign on a 1/100 scale model of the DTMB 5415 frigate hull and presented results for the motion and loads of the intact and damaged ships. An experimental investigation of motions and loads of the same DTMB 5415 frigate hull in intact and damaged conditions was conducted by Begović et al. (2011). Furthermore, Begovic et al. (2013) discussed the model scale influence since they conducted the same measurements on the DTMB 5415 frigate hull with a different 1/51 scale. The experiments have been carried out in the Kelvin Hydrodynamic Laboratory at the University of Strathclyde, Glasgow, where the towing tank is 76 m long, 4.6 m wide and has a water depth of 2.15 m. Recently, Domeh et al. (2015) presented an experimental study analysing the influence of permeability and damage size on the vertical motions of the damaged hull. The recent study by Begovic et al. (2017) focused on the measurement of hull girder loads on the segmented model of DTMB 5415 (1/

51 model scale) at zero speed in intact and damaged conditions, studying the nonlinear effects due to the wave height variation.

3. Towing tank and model description

Within the present campaign, experiments are carried out at Brodarski Institut in Zagreb (Croatia). The towing tank is 276.3 m long, 12.5 m wide and has a water depth of 6.0 m. The flap type wave maker is used to produce waves with a height range of 0.08–0.7 m and the period range of 0.1–3.0 Hz. The towing tank carriages are capable of achieving speeds up to 14 m/s with a maximum acceleration of 1 m/s^2.

Two sets of experiments are performed in the study. In the first set of experiments, irregular waves are considered and Tabain's (1997) one-parameter wave spectrum is used with a significant wave height of 3.5 m, with the model speed of 0.67 m/s corresponding to 7 knots at full scale. This particular wave spectrum is conveniently used in the Adriatic Sea for different engineering applications (Katalinić and Parunov, 2018). Tabain's spectrum is presented in Figure 1 (in full scale plotted against wave frequency ω) along with the spectrum obtained in the towing tank based on the measured wave elevation. Power spectral density evaluation of waves in the towing tank is conducted using the Fast Fourier Transform (FFT) and with smoothing of the obtained amplitudes as described by Bergdahl (2009).

The second set of experiments is performed on the segmented ship model in regular head waves with the constant wave amplitude ($H_{FS} = 3.5$ m, respectively $H_{MS} = 0.1215$ m, where indices FS and MS denote the full and model scale, respectively) with 7 different wave periods and corresponding wavelengths specified in Table 1. According to the criterion of the wave surface propagation in deep water ($d/L_w > 0.5$), all tested conditions can be considered as carried out in deep water. For a water depth of $d = 6$ m, and all tested conditions given in Table 1, the ratio d/L_w is larger than 0.5. Ship speed is 5 knots in this experiment.

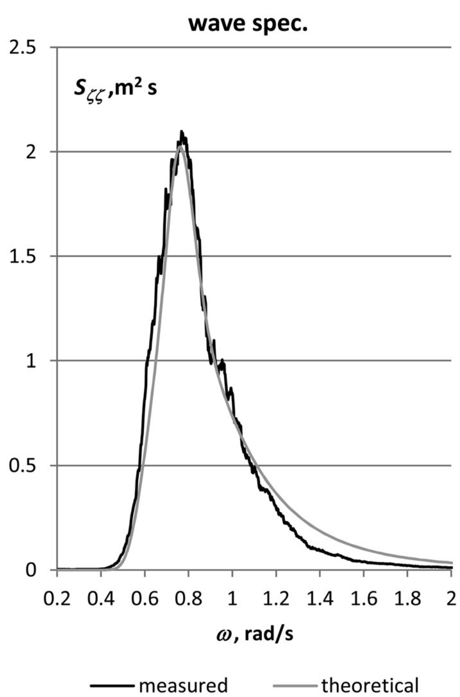

Figure 1. Wave spectrum.

Table 1. Definition of the tested conditions.

	Model scale				Full scale		
Condition	L_w/L_{pp}	L_w(m)	T (s)	v_m (m/s)	L_w (m)	T (s)	V (kn)
1	0.6	3.64	1.53	0.48	104.9	8.20	5.0
2	0.8	4.85	1.76		139.8	9.47	
3	1.0	6.07	1.97		174.8	10.58	
4	1.2	7.28	2.16		209.8	11.59	
5	1.4	8.49	2.33		244.7	12.52	
6	1.6	9.71	2.49		279.7	13.39	
7	1.8	10.92	2.64		314.6	14.20	

3.1. Model for measurement of rigid body motions of ship in tow

The ship model is made from wood where the hull form is offsets of a Panamax oil product tanker with main particulars as specified in Table 2. The hull form of the tanker as shown in Figure 2 is provided with courtesy from Uljanik Shipyard (Pula, Croatia). Bulkheads arrangements for the model within the damaged area are presented in Figure 3 along with the hull opening at the bottom. The procedure for determining the position and dimensions of the opening is described in Section 3.3. Transverse watertight bulkheads (closing the damaged area) are made of plywood with 12.5 mm thickness while the top cover is made from plexiglas 5 mm thick. Each compartment was fitted with a small tube (approx. 30 mm in diameter) to assure air flow during tests in the damaged condition (Figure 3). The floating position of the model in the damaged condition with the water in the flooded tanks is represented in Figure 4. Rudder and propeller are mounted to the model during all the experiments. The rudder is held in the initial position (with the zero rudder angle) while the propeller could rotate freely (due to the inflow of surrounding water).

Table 2. Ship model properties.

Designation	Quantity	Unit
Scale	1:28.8	
Overall length	6.345	m
Length between perpendiculars	6.067	m
Breadth moulded	1.118	m
Depth moulded	0.608	m
Intact condition		
Draft at after perpendicular	0.448	m
Draft at forward perpendicular	0.448	m
Displacement volume	2.479	m3
Total mass of the model	2.479	kg
Centre of gravity above keel	0.355	m
Longitudinal position of centre of gravity from after perpendicular	3.122	m
Roll radius of gyration	0.354	m
Pitch radius of gyration	1.360	m
Yaw radius of gyration	1.393	m
Damaged condition		
Draft at after perpendicular	0.451	m
Draft at forward perpendicular	0.472	m
Displacement volume (including flooded water)	2.562	m3
Total mass of the model (without flooded water)	1.902	kg
Centre of gravity above keel	0.338	m
Longitudinal position of centre of gravity from after perpendicular	2.984	m
Roll radius of gyration	0.379	m
Yaw radius of gyration	1.587	m
Pitch radius of gyration	1.618	m

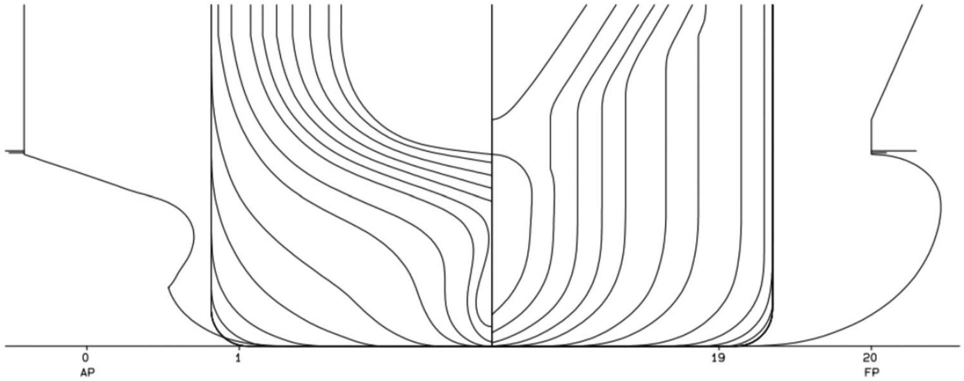

Figure 2. Body lines of the Panamax oil tanker.

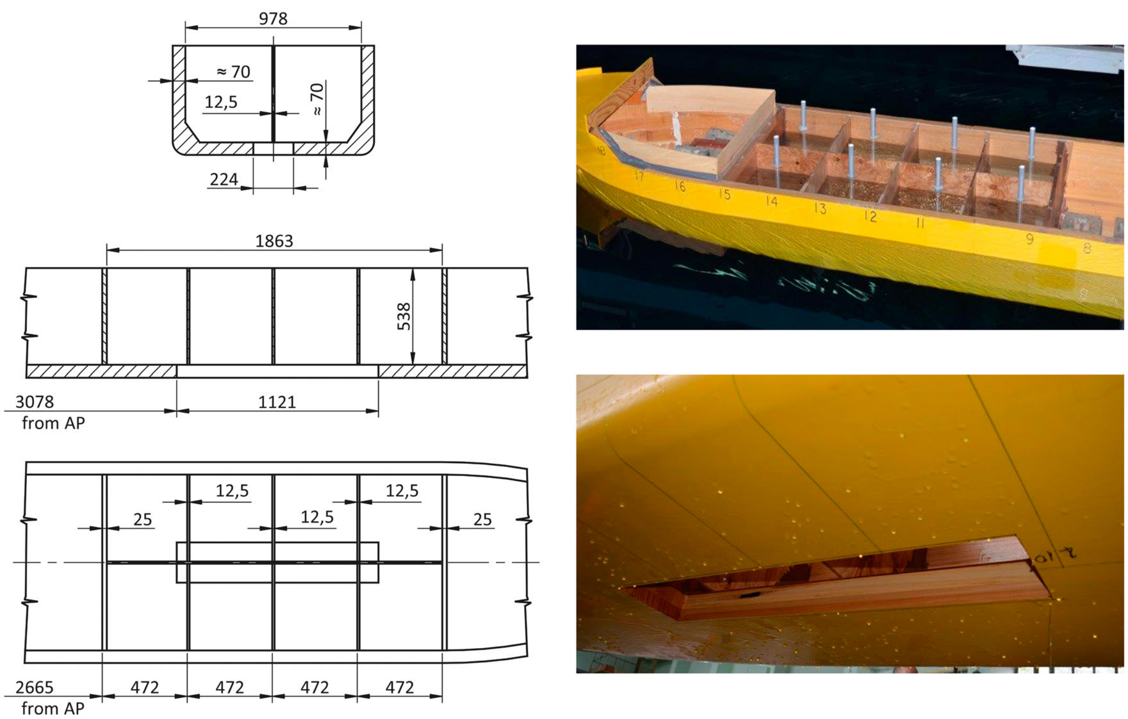

Figure 3. Ship model within the damaged area.

It should be noted that the damaged area of the model contains only compartments that correspond to the cargo tanks (Figure 3). Cargo tanks are made in the model but without ballast tanks in the double hull of the tanker since due to practical reasons it was impossible to include the ballast tanks in the model. Namely, the thickness of the model hull (made from wood) was approximately 70 mm. With the scale of 1:28.8 this thickness corresponds to 2 m length in real scale of the tanker which is a usual height of the double bottom or the width of the double side. If the thickness of the model was reduced (to fit the ballast tanks), the structural integrity of the model would be undermined and it would not be possible to conduct any experimental tests. For the towline, a wire rope with a diameter of 3 mm is used with 6 strand fibre core wire ropes and an approximate mass per unit length of 0.031 kg/m.

3.2. Segmented model for VWBM measurements

For the second set of experiments, the ship model from the first experimental set-up is cut at station 10 (the midship section)

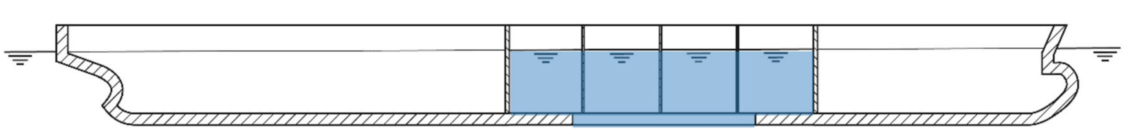

Figure 4. The floating position of the model in the damaged condition.

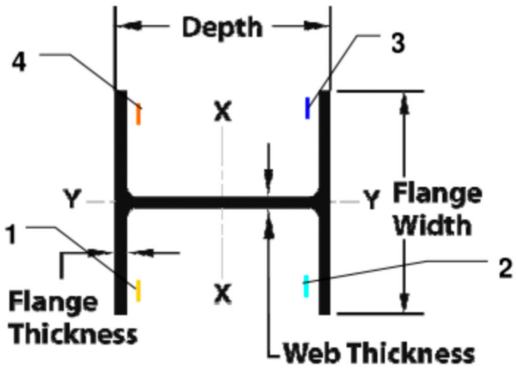

Figure 5. HEA profile of the iron measuring beam with position of the strain gauges (depth = 152 mm; flange width × thickness 160 mm × 9 mm; web thickness 6 mm).

into two segments, connected together with the measuring iron beam type HEA 160, with the length of 2500 mm. The geometry of the beam cross section is given in Figure 5. Four strain gauges for measuring vertical wave bending moment, designated by numbers 1–4, are applied on the measuring beam at the position of Station 10. In order to make the measuring beam more sensitive to bending, the part of the beam flange where strain gages are glued is thinned to 5 mm as presented in Figure 6. The resulting bending moment is calculated by averaging all four measured strains.

The way how the measuring beam was fixed to the ship model is presented in Figure 7(a). Water leakage to the dry compartment is reduced to the minimum in order not to affect the measurements. The gap of 15 mm between the two ship segments is closed with flexible rubber during testing of the intact model as presented in Figure 7(b,c). In the damaged ship testing conditions, the rubber between the segments is removed.

3.3. Definition of the damage extent

For the definition of the damage extent, two important features have to be considered. Firstly, the damage extent should be large enough in order to be able to capture the influence of flooding during experiments. On the other hand, such a large damage extent should be realistic, which means that it should have a relatively high probability of occurrence in case of grounding. It is assumed that the grounding damage occurs at the ship centreline, thus always simultaneously damaging pairs of tanks (portside – PS and starboard – SB). The longitudinal damage location and extent are analysed by Monte Carlo simulation based on damage probability distribution functions specified by the MEPC (IMO 2003) and shown in Figure 8. From these probability density functions and from the location of transverse bulkheads between cargo tanks, it is possible to determine the probability of damaging a certain number of pairs of cargo tanks (PS and SB) during grounding (Figure 9).

From Figure 9, which is based on 1000 Monte Carlo simulations, the probability that four pairs of tanks are damaged during grounding could be estimated to be 11%. Based on this rather large probability of occurrence and the fact that the intention was to capture the effect of flooding during model tests, it was decided to model such a damage which

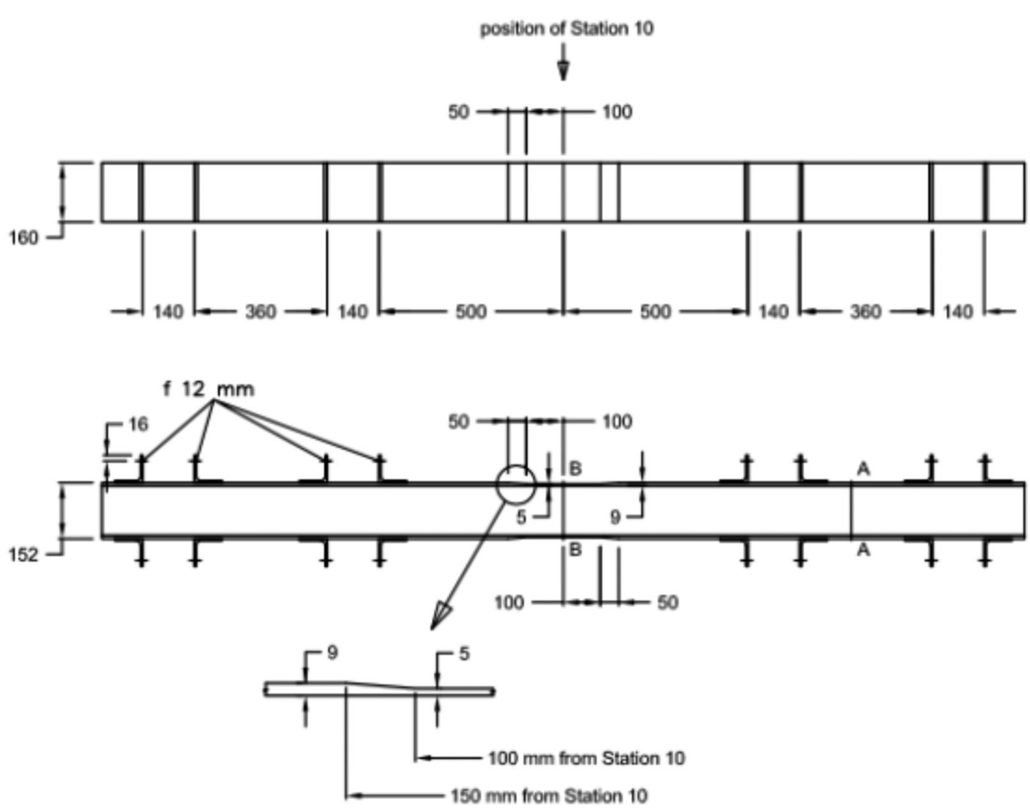

Figure 6. Geometry of the back spline with reference to the Station 10 position.

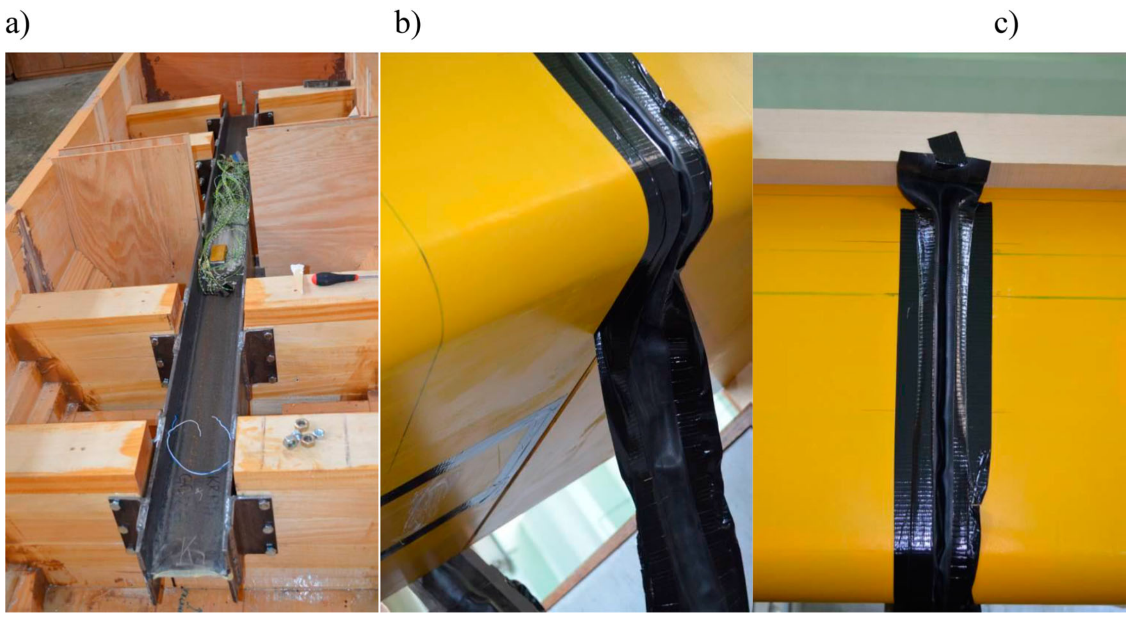

Figure 7. Geometry of the back spline with reference to the Station 10 position.

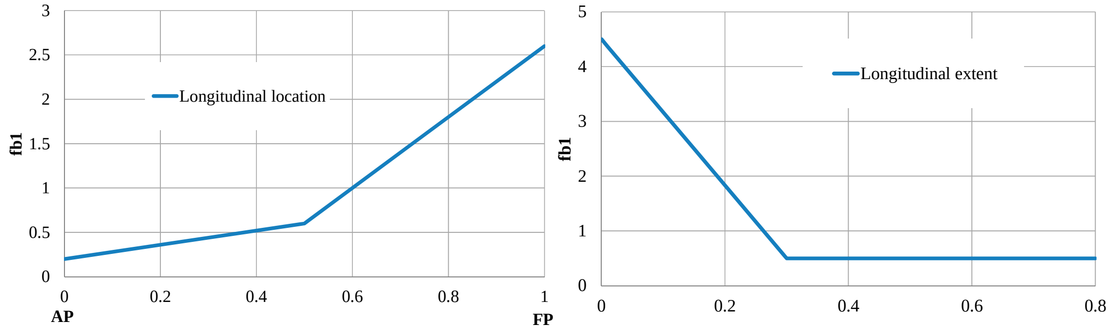

Figure 8. Probability distribution functions of longitudinal extent and longitudinal location for grounding (MPEC, IMO 2003).

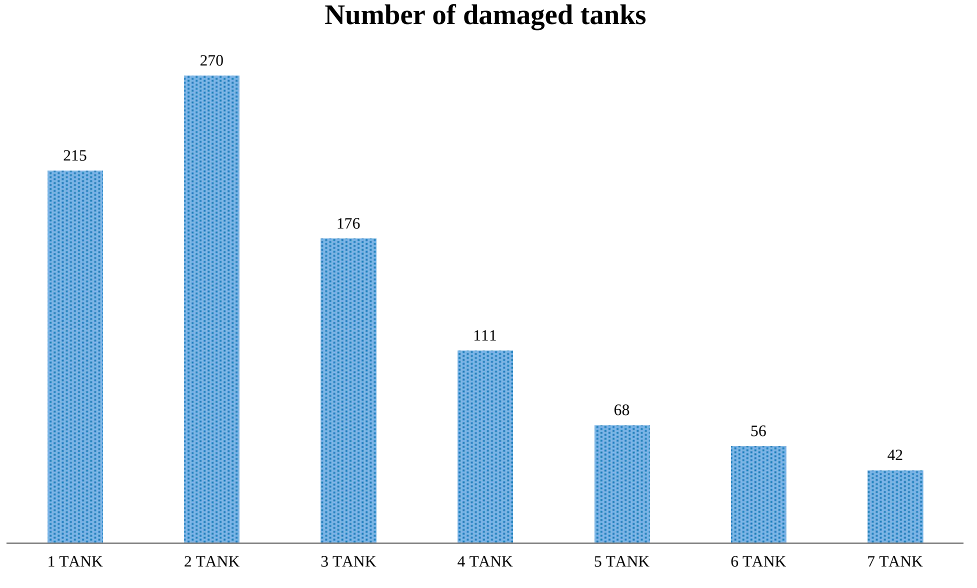

Figure 9. Probability of occurrence of the number of damaged tanks due to grounding (for 1000 random numbers).

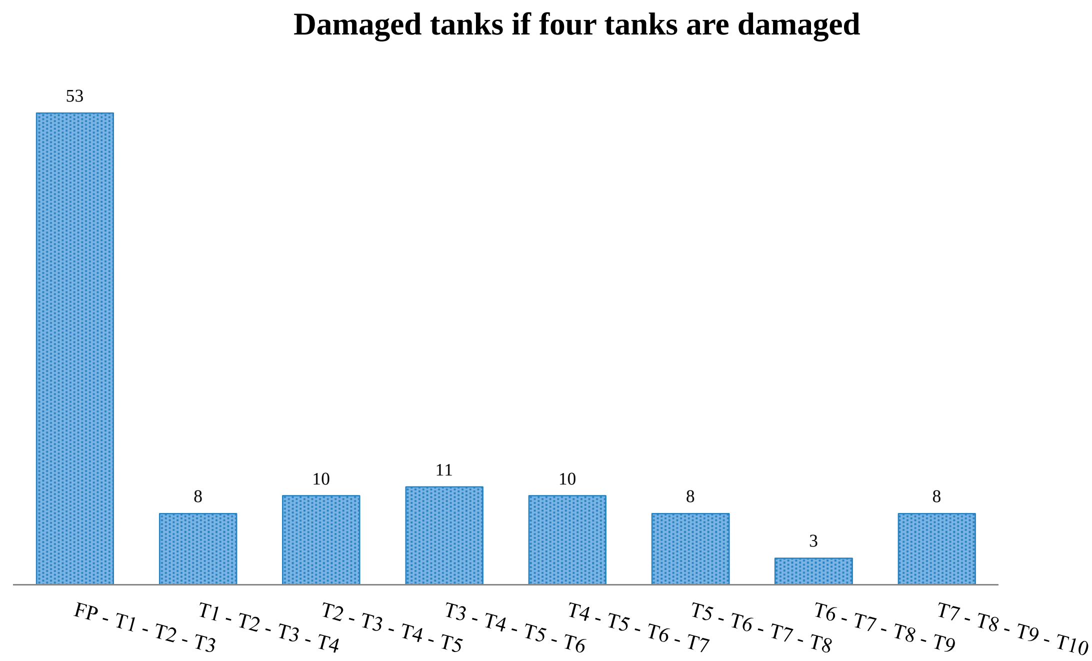

Figure 10. Probability of occurrence of specific combination of damaged tanks if the four tanks are damaged due to grounding.

extends across four pairs of cargo tanks. By further analysis, which is presented in Figure 10, it can be concluded that the combination of damaged tank pairs T3-T4-T5-T6 have the highest probability of occurrence (1.1%) if only cargo tanks in the middle part of the ship are considered (without the Fore peak tank). Since the intention was to maximise bending moments in the midship region, the stated damaged cargo tank pairs are considered. The longitudinal damage on the ship scale extends between frame 110 and frame 148, covering four cargo tanks and having a length of 32.3 m, as shown in Figure 11. If that damage extent is scaled to the ship model, the bottom opening of dimensions 1400 × 226 mm, extending from Station 10–160 mm to Station 14 + 20 mm and placed in the centreline of the hull is obtained (see Figure 3).

The damage width is assumed to be 20% of the ship's breadth which reads as 6.44 m. According to MEPC probability distribution, this damage breadth has 44% probability of exceedance in case of grounding. Vertical damage extent is taken such that the inner bottom is breached and cargo tanks are flooded, having a probability of exceedance of 11% according to MEPC probability functions.

4. Results of the first experimental set-up

As previously described, the first set of experiments is related to the measurement of rigid body motions in the intact and damaged ships in the irregular waves in head seas with a small ship's forward speed of 7 knots. Waves are simulated by Tabain's wave energy spectrum as presented in Figure 1, which has recently been used in structural reliability study of a damaged oil tanker in the Adriatic Seas (Parunov et al. 2017).

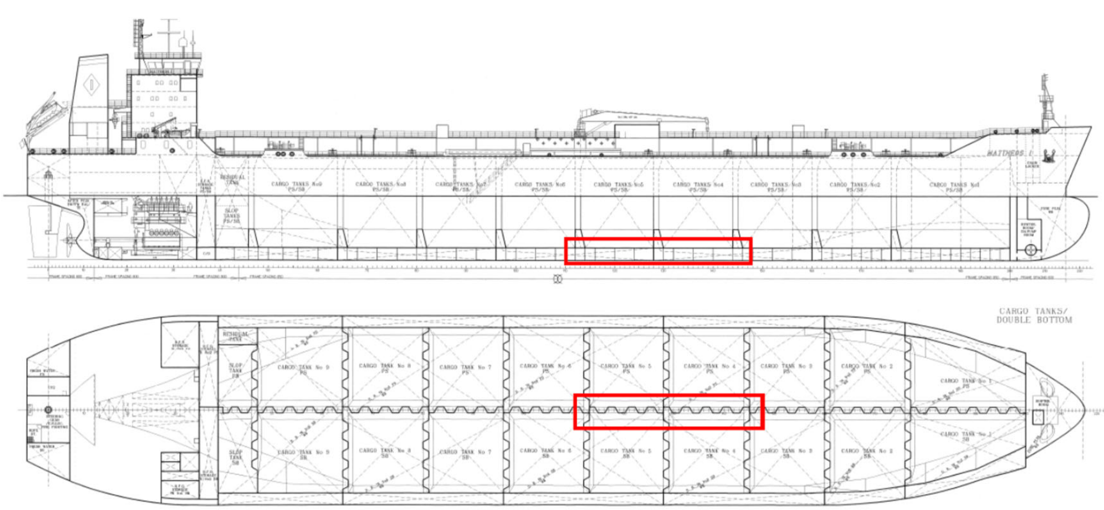

Figure 11. General arrangement of the Panamax tanker and damage extent.

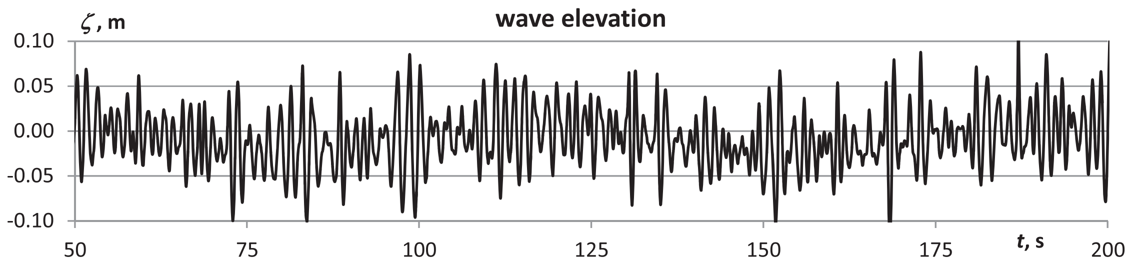

Figure 12. Time history of wave elevation in the towing tank.

A sample of control measurement of the wave elevation (ζ) is presented in Figure 12, corresponding to the model significant wave height of $H_{MS} = 0.1215$ m. Measurements in irregular waves are conducted for three experimental runs with the duration of each individual run of 4 min. Each experimental run resulted in a time series of heave (z), roll (φ) and pitch (θ). The samples of these motions are presented in Figure 13. On the same figure, the sample of the yaw (ψ) is also shown for the damaged case.

Besides the motions, the tension force in the towline was measured as presented in Figure 14. These records are further analysed by the FFT with smoothing of obtained amplitudes as described by Bergdahl (2009), in order to obtain response spectrums. The obtained response spectra are presented in

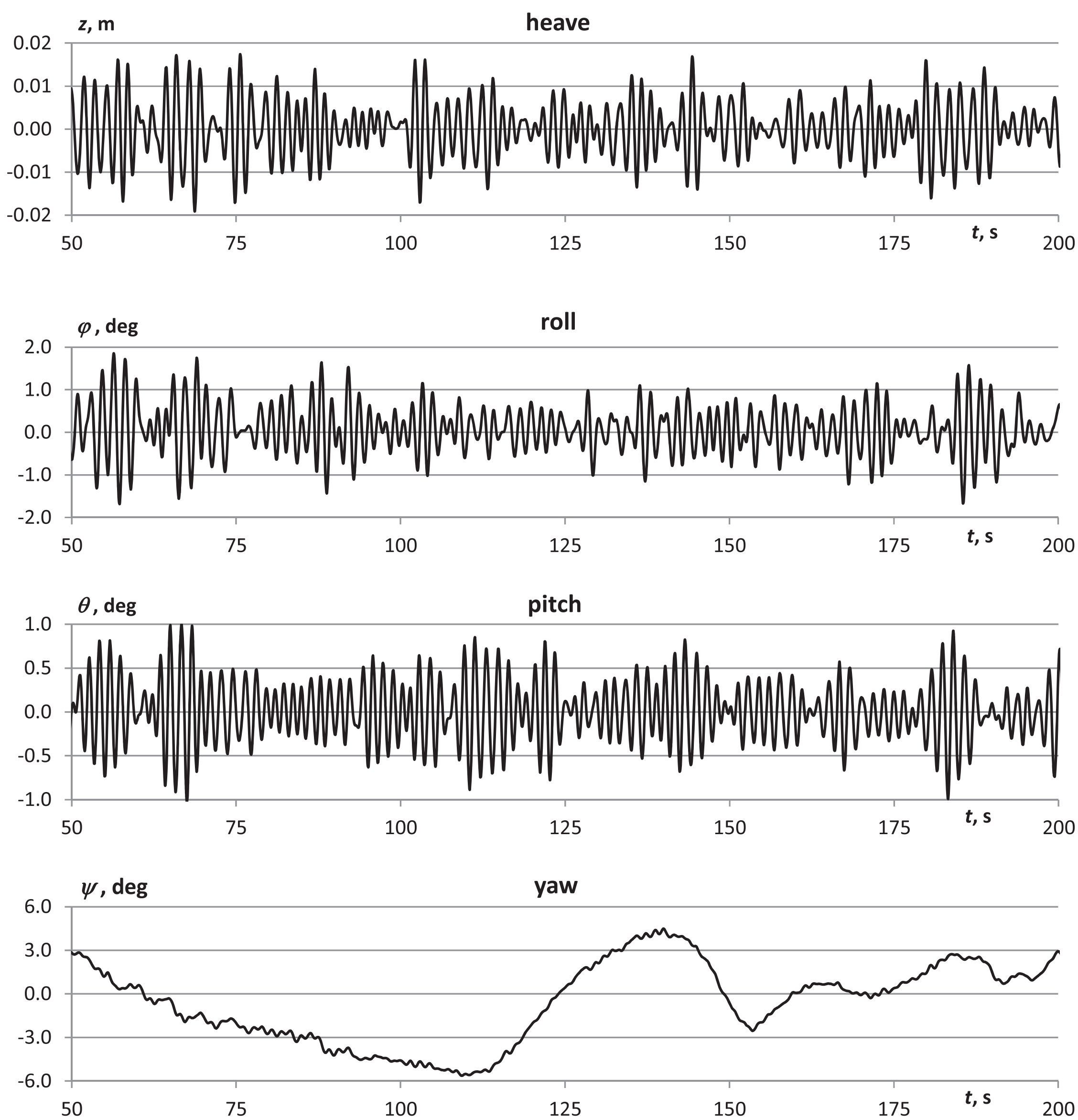

Figure 13. Time histories of the model motions (damaged condition).

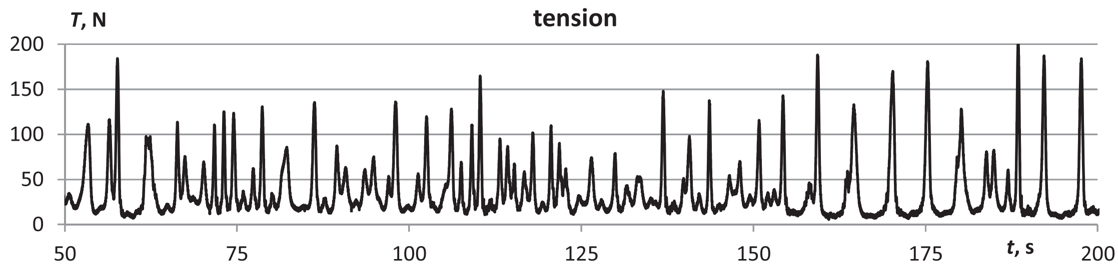

Figure 14. Time history of the tension force in the towline of the model (damaged condition).

Figures 15 and 16 in the ship scale as functions of the encounter wave frequency (ω_e).

The numerical calculation of rigid body motions and forces in the tow line in the frequency domain is performed using Hydrostar (2011) that is based on the potential flow theory and 3D BEM. The flooded area within the damaged case was modelled as a single tank containing the sea water, so the sloshing effects were taken into account. For the sake of simplicity, the hull opening was not modelled. The stiffness of the towline was linearised around the mean value of the measured tension force and according to the simplified mathematical model presented in Ćatipović et al. (2014). The stiffness was applied in the surge direction of the ship. Obtained numerical results for vertical ship motions are compared with experimental measurements, as presented in Figure 16.

Only the heave and pitch are numerically evaluated because of the limitations of the linear numerical tools used. Namely, as can be seen from Figure 13, relatively large values of yaw angle occurred due to the directional instability of the model (which is common for most of merchant ships). Such yaw values cannot be correctly evaluated by usual seakeeping codes. Further, it is well known that in the case of large yaw angles (changing incident wave angle) large changes of roll response occur, which also cannot be predicted by usual seakeeping codes. Therefore, the numerical evaluation of the roll and yaw repose was left out.

By analysing experimental results it can be concluded that the response spectrum for the pitch is larger for an intact condition which is in agreement with numerical results. In the case of the heave motion, the larger response spectrum is obtained for the damaged ship condition. However, this is not correctly predicted by the linear seakeeping analysis. In general, however, by taking into account the complexity of the physical model and the simplicity of the linear seakeeping tools, the prediction of the vertical ship motions may be considered as satisfactory.

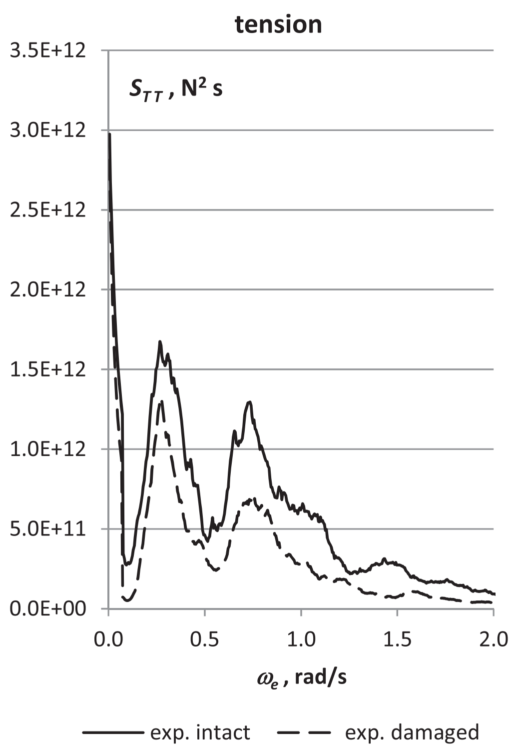

Figure 15. Towline tension spectrum in the ship scale.

5. Results of the second experimental set-up

The second set of experiments is performed on the towed ship model, consisting of two segments connected at the midship section by the elastic bar. Measurements of the VWBM and ship motions signals are performed in head seas in regular waves, for the intact and damaged conditions. Seven wave periods are considered in the experiment as specified in Table 1. Averaged peak to peak values of the pitch, heave and waves are used to calculate the RAOs as recommended by ITTC (2002). The obtained RAO functions of pitch and heave motion are normalised in such a way as to produce dimensionless values. Bending moments are measured with strain gauges at 4 points at the elastic bar (Figure 5). Since the measurements revealed similarity of bending moments between points 1 and 4, and points 2 and 3, only results obtained at points 1 and 2 are considered. Experimental heave and pitch RAOs are presented in Figure 17, while VWBM RAOs are shown in Figure 18. Both figures include results for damaged and intact condition (in the ship scale). The two figures also show the numerical results obtained by Hydrostar (2011). The repeatability of experiments, due to budget limitations, is verified only for $\lambda/L = 1.2$ for the damaged condition. Repeated results of heave, pitch and VWBM are denoted by ○ in Figures 17 and 18, where it can be seen that repeatability of measurements is fairly good for the given frequency.

Numerical results of ship motion for intact and damaged ship condition are almost the same, probably due to limitations of the numerical model used, i.e. the sloshing effect is ignored. By analysing experimental results in Figure 17, it can be concluded that heave and pitch motions are larger for the intact ship compared to the damaged ship for the majority of

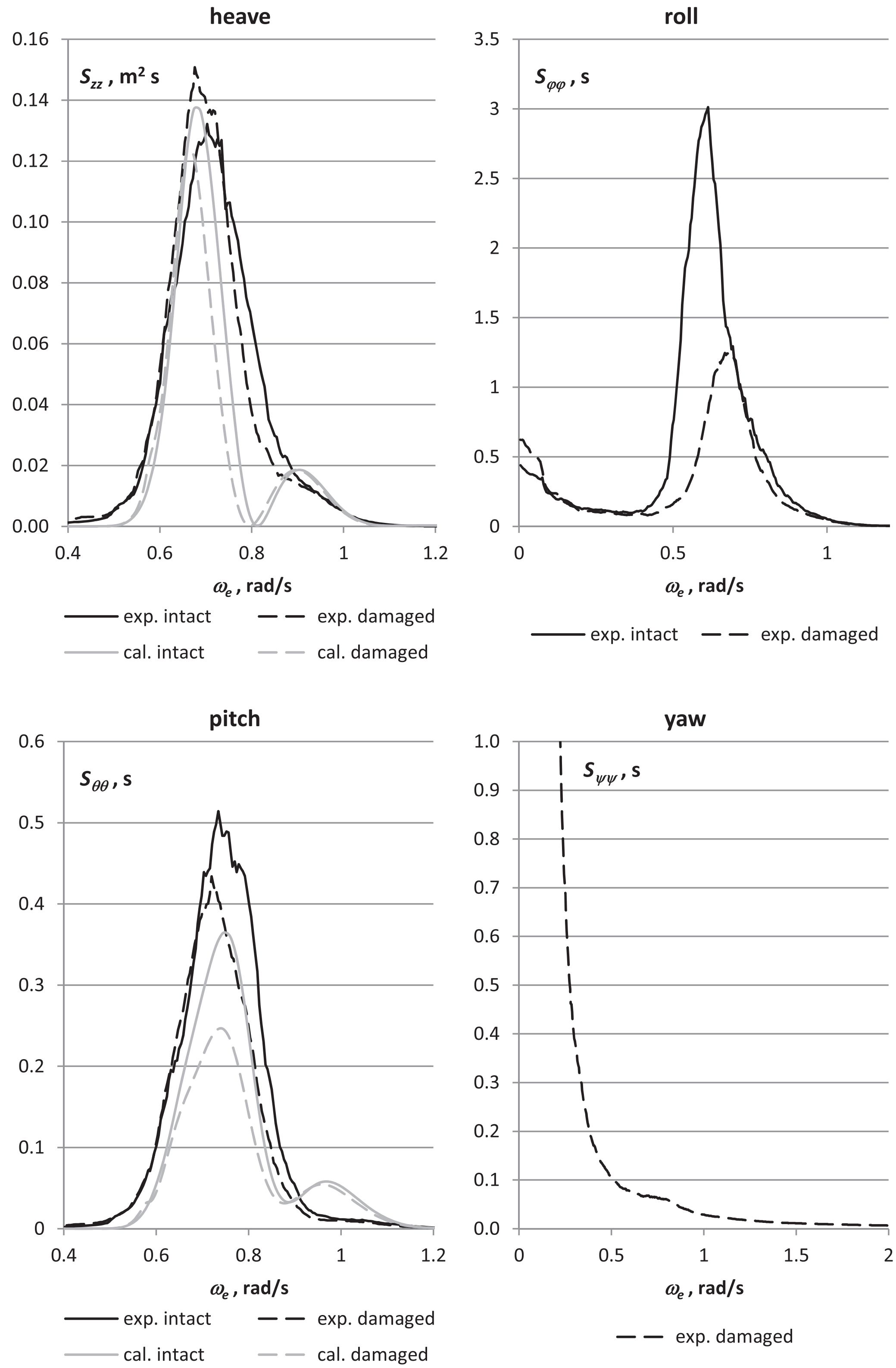

Figure 16. Motion spectrums in the ship scale.

wavelengths. Two resonant-type peaks of RAOs are measured for $\lambda/L = 1.0$ for heave and $\lambda/L = 1.4$ for pitch, for both intact and damaged conditions. Neither of two resonances was found in the numerical analysis. Here, it is important to note that the simplified numerical model is used, which linearises the stiffness of the towline. Moreover, the linearised stiffness was applied only in the surge direction of the ship. This is opposed to the real properties of the used towline which are highly nonlinear and at least two-dimensional. Thus, the numerical model is not able to capture all the physics of the observed problem as well as all the measured peaks. Furthermore, since the sloshing effect is ignored within the numerical

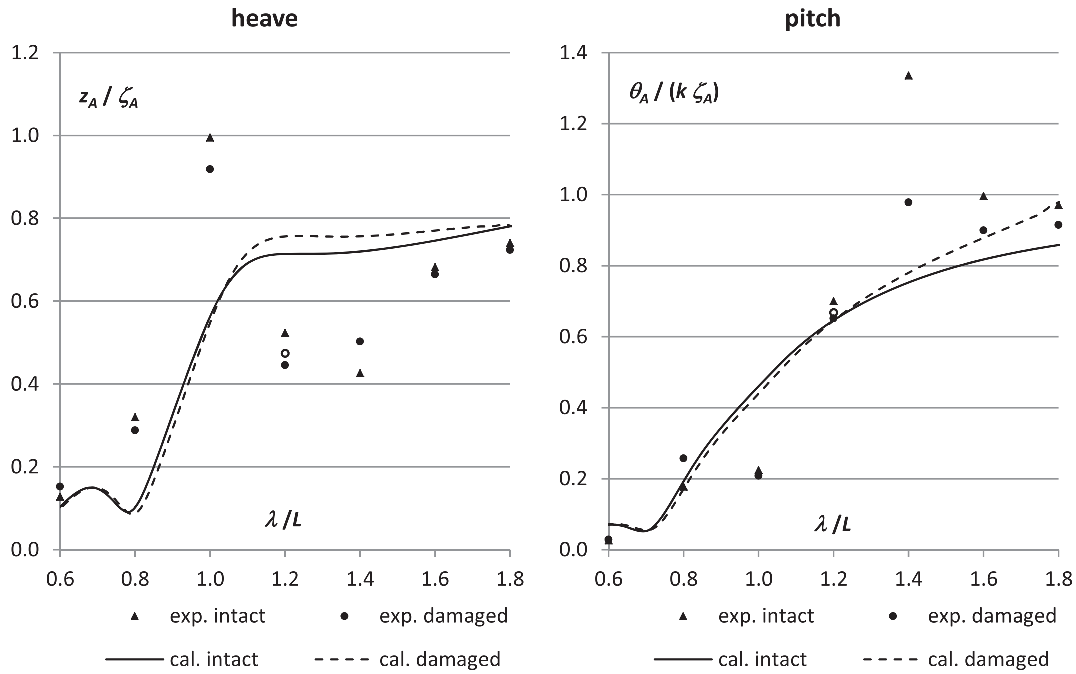

Figure 17. RAO functions of heave and pitch motion (repeated experimental results on damaged ship are denoted by ○).

model, the natural frequencies for sloshing in the flooded tank are evaluated according to Ibrahim (2005). The general equations of motion for a fluid in closed containers are simplified by assuming the container to be rigid and impermeable. Furthermore, the fluid is assumed inviscid, incompressible and initially irrotational. The expression for two-dimensional flows

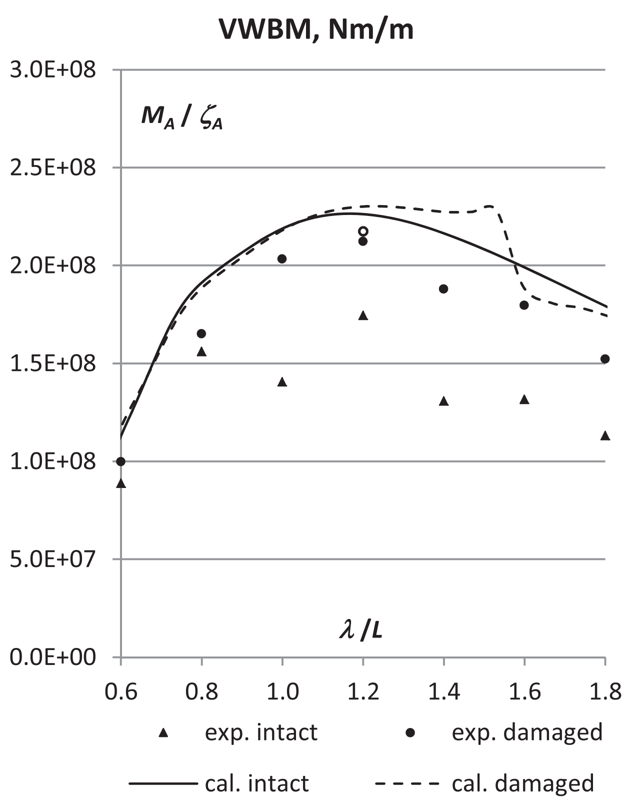

Figure 18. RAO functions of VWBM (repeated experimental result on damaged ship is denoted by ○).

is used and the lowest natural frequency is calculated as 0.58 rad/s which corresponds to $\lambda/L = 0.61$. So, the measured peaks presented in Figure 17 are not caused by the occurrence of sloshing in the flooded tank.

From Figure 18, it may be seen that the measured VWBM RAOs are considerably larger for the damaged than for the intact ship condition. Furthermore, it may be noticed that numerical results overestimate measurements. Similarly, as for the ship motion, the calculated RAOs for the intact and damaged ship are quite close to each other.

6. Conclusion

The paper presents the results of the seakeeping experiments performed on a Panamax oil product tanker damaged in grounding. The measured experimental results are compared with the numerical computations by the hydrodynamic analysis. Two different experimental and numerical set-ups are conducted and compared.

The first set of experiments is related to the measurement of rigid body motions and towline forces for intact and damaged ships. Experiments are performed in irregular waves, in head seas with the small ship's forward speed. Time records of heave, pitch, roll and yaw motions, as well as the tension force in the towline, are analysed by the FFT. The measurements show that the heave motion is larger in case of the damaged than in case of the intact ship, while pitch motions are larger for the intact case. The measured roll motion and towline tension force are larger for the intact ship than for the damaged ship. Response spectra of heave and pitch motions are compared to the numerical results. The agreement of experimental and numerical results is satisfactory.

The second set of experiments is performed on a towed ship model, consisting of two segments connected at the midship section by the elastic bar. Seven wavelengths are considered in the experiment, and averaged peak to peak values of the pitch, heave and VWBM are used to calculate the RAOs. The comparison of experimental and numerical results for ship motions is not satisfactory, especially for two resonant frequencies, one in heave and the other in pitch that significantly deviate from the numerical values. The disagreement of results could be the consequence of the measurement uncertainties, as the repeatability of experiments is verified for only one frequency. The agreement between experimental and numerical results for VWBM RAOs is better for the damaged than for the intact ship. Measured VWBM RAOs are larger in average by 28% for the damaged than for the intact ship. This last conclusion is particularly important since differences in VWBM RAOs between intact and damaged ships are very often ignored and RAOs of the intact ship are frequently used in the safety assessment of damaged ships.

Acknowledgements

Seakeeping experiments are performed in Brodarski Institut (Zagreb, Croatia). The ship model that is used in experiments is provided by Uljanik Shipyard (Pula, Croatia).

Disclosure statement

No potential conflict of interest was reported by the authors.

Funding

The work has been fully supported by the Croatian Science Foundation within the project 8658. Also, this project has received funding from the European Union's Horizon 2020 research and innovation programme under the Marie Sklodowska-Curie grant agreement no. 657539 STARMAS.

ORCID

Joško Parunov http://orcid.org/0000-0002-8566-4927

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