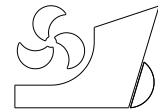


Ivana Martić
Nastia Degiuli
Ivan Čatipović



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ADDED RESISTANCE IN WAVES OF INTACT AND DAMAGED SHIP IN THE ADRIATIC SEA

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Summary

In this paper the ship added resistance in regular head waves and at different sea states in a certain frequency range at different forward speeds was calculated. Calculations were conducted using hydrodynamic software. Since after a maritime accident a damaged ship often has to be removed from the place where the accident occurred, the seakeeping characteristics of the damaged ship were also calculated. The response of the damaged ship was simulated using two models: damage simulated as an increase in the ship displacement mass and as a flooded tank within the midship area. Calculations are based on the linear potential flow theory and added resistance was determined by the wave drift force as the second order wave load. The quadratic transfer function QTF which describes low-frequency second order wave loads was approximated by its zeroth term only, i.e. by the drift load at incoming wave frequencies. The mean added resistance values of the intact and the damaged ship were compared. The calculation results show that added resistance in waves of the damaged ship with a flooded tank differs slightly from the added resistance in waves of the damaged ship with an increased displacement.

Keywords: *potential flow; wave load; added resistance in waves; quadratic transfer function; drift force*

1. Introduction

In order to evaluate the safety of ship navigation and the possibility of carrying out the designed purpose of the ship, it is necessary to know its seakeeping characteristics. In order to ensure good seakeeping characteristics, the ship motion amplitudes and accelerations must be within acceptable limits and the added resistance in waves should not have a significant effect on the decrease in ship speed and the increase in power and fuel consumption [1]. Sailing in waves also affects the ship maneuverability. Due to the ship added resistance in waves, the power and fuel consumption of the ship can increase up to 85% compared to sailing in calm waters [2]. When a ship oscillates in waves, energy is transmitted to the surrounding water due to damping of oscillatory motions. Added resistance is caused by the drift force due to the

interference of incoming waves and the waves due to heave and pitch motions, the hydrodynamic damping of heave and pitch motions in calm water (radiation) and diffraction due to the interaction of the incoming waves and the ship. Drift force is the most important component of added resistance. The diffraction induced added resistance is dominant for relatively high wave frequencies, while the viscous damping can be neglected [3]. Added resistance can be determined with certain accuracy experimentally and by using different analytical and numerical methods. The accuracy of added resistance calculation depends on the accuracy of the ship motion prediction [4].

It is possible to estimate the ship added resistance by using empirical equations. Many theoretical calculation methods have been developed, but not all of these methods provide satisfactory results or their application is relatively limited by the type of the ship, speed, direction of waves and similar factors. The methods for determination of added resistance mainly use the strip method which allows the calculation of the ship motions and loads induced by waves with acceptable accuracy for low Froude numbers and slender bodies. According to some studies, the strip method provides results with satisfactory accuracy if the effect of radiation is dominant, but due to the diffraction of waves on the bow part of a ship the method requires a certain correction in the added resistance calculations [5]. In the seakeeping analysis of three-dimensional bodies it is common practice to calculate radiation/diffraction model by some other methods like boundary element methods or any other method based on the Green function (offshore structures, small L/B ratio or similar).

In this paper, a comparison of the experimental results [6] with numerically calculated results of the container ship added resistance in regular waves is presented in order to evaluate the suitability of numerical methods when solving this complex hydrodynamic problem. Added resistance was calculated by evaluation of the drift forces as the mean value of the second-order wave loading at different frequencies of incoming waves through the quadratic transfer function QTF. The ship added resistance for two different sea states was also calculated. In order to evaluate the impact of fluid motions inside the tank of the damaged ship, a flooded tank inside the ship hull was generated and the obtained calculation results of the damaged ship were compared with the added resistance of the intact ship. For comparison purposes, the added resistance of the ship with the added displacement mass that equals the mass of the fluid inside the tank was also calculated.

The analysis was performed using hydrodynamic software [7].

2. Added resistance in waves

While moving forward in waves, a ship generates waves associated with forward speed through still water and waves associated with its vertical relative motion response to waves [8]. The total resistance of the ship in regular waves consists of resistance in still water that is constant at a given constant speed and oscillating resistance due to motions of the ship depending on encounter wave frequency ω_e . The time averaged part of the resistance increase due to motions relative to waves is called the ship added resistance.

When a ship sails in waves, its forward speed decreases due to the increasing resistance. Therefore, to sustain the forward speed, engine power and fuel consumption are increasing. The ship speed loss is also caused by several other factors such as the resistance due to wind acting on the hull and superstructure, resistance due to diffraction of waves, due to reduction of the propulsion efficiency as well as due to ship maneuverability, etc. [1]. Generally speaking, a ship can experience a 15-30% resistance increase in a seaway mainly caused by added resistance in waves [3].

There are two basic approaches for the estimation of time averaged added resistance: a radiated wave energy method and an integrated pressure method [8]. The radiated energy method is based on the strip theory and determines radiated wave energy during one period of ship oscillation as follows:

$$P = \int_0^{T_e} \int_L b'_{33} v_z^{*2} dx_b dt \quad (1)$$

where b'_{33} is the hydrodynamic damping coefficient of the vertical motion of the cross section, v_z^{*2} is the vertical average velocity of the water particles relative to the cross section, T_e is the period of vertical oscillation of the cross section.

Based on this energy delivered to the surrounding water it is possible to determine the mean added resistance as follows:

$$\frac{R_{AW}}{\zeta_a^2} = \frac{-k \cos \beta}{2\omega_e} \int_L \left(N'_{33} - v \frac{dM'_{33}}{dx_b} \right) \left(\frac{v_{z_a}^*}{\zeta_a} \right)^2 dx_b \quad (2)$$

where k is the wave number, β is the wave direction, ω_e is the encounter frequency, N'_{33} is the two-dimensional potential damping coefficient for heave, v is the ship speed, M'_{33} is the two-dimensional potential mass coefficient for heave.

The method provides good results in head to beam waves, but it is not reliable in the case of the following sea. At low encounter frequencies when the ship transfer function converges to one, the sectional added mass is high and potential sectional damping is almost zero. In that case high motion peaks and extreme added resistance can be expected [8].

The calculation of the ship added resistance using the integrated pressure method is based on the longitudinal integration of oscillating pressure on the wetted surface area of the hull. Since the ship local coordinate system oscillates together with the hull, it rotates with the angle of pitch motions. Due to the relative motion and variation of the wetted surface area, the integrated vertical hydrodynamic load causes the second order contribution to the mean added resistance.

The contribution to the total mean added resistance at the given encounter frequency ω_e is finally given by:

$$\frac{R_{AW}}{\zeta_a^2} = \frac{1}{2} \rho g \int_L \left(\frac{s_a}{\zeta_a} \right)^2 \frac{dy_w}{dx_b} dx_b + \frac{1}{2} \rho \nabla \omega_e^2 \frac{z_a}{\zeta_a} \frac{\theta_a}{\zeta_a} \cos(\varepsilon_{z\zeta} - \varepsilon_{\theta\zeta}) \quad (3)$$

where s_a is the vertical relative motion, $s = \zeta - z_x$, ∇ is the volume of displacement, z_a is the vertical motion, θ_a is the pitch angle, $\varepsilon_{z\zeta}$ is the phase shift for heave, $\varepsilon_{\theta\zeta}$ is the phase shift for pitch.

In order to simulate the fluid flow around a ship when sailing in waves as simply as possible with satisfactory suitability, the linear potential flow theory is used. It is based on the simplest form of two-dimensional progressive surface wave of small amplitude. Fluid flow in potential theory is irrotational, homogeneous, inviscid and incompressible, and the velocity field is described by a continuous scalar function that has a finite value in every point of the fluid and defines the velocity vector. The linear potential flow theory is based on the Laplace equation (continuity of homogeneous incompressible fluid), the Euler equation that defines the balance of forces on the material particle and the Bernoulli equation, i.e. the law of conservation of energy per unit mass.

The hydrodynamic software [7] used to perform necessary calculations is based on the boundary integral equations and linear potential flow theory. In order to take into account the ship forward speed, encounter frequency approximation is implemented in the software [7] based on the Green function. The translating-pulsating source method based on the Green function is used in hydrodynamic calculations for cases where forward speed is simulated. Since the method is very time consuming and numerically demanding, when the forward speed is small in amplitude, an approximate free surface boundary condition can be used:

$$-\omega_e \varphi + g \frac{\partial \varphi}{\partial z} = 0 \quad \text{on } z = 0 \quad (4)$$

where φ is the fluid potential.

Since the low to moderate ship speeds corresponding to $Fr < 0.3$ are considered in this paper, it was possible to base the calculations on the approximated boundary condition given in equation (4).

3. Quadratic transfer function QTF and approximation of QTF

The drift force due to incoming wave is caused by hydrodynamic loads and the second order interaction between motion and wave field of the first order. Given the potential of flow velocity of incoming wave in deep water, and based on the Bernoulli equation, the mean pressure of the second order can be determined. By integration of the mean pressure along the hull it is possible to determine the forces and moments of the second order that act on the ship hull. Added resistance is determined by the drift forces as the second order load due to incoming waves, through quadratic transfer function QTF. During the ship-wave interaction, part of the energy that incoming waves possess is transferred to the ship in accordance with the momentum theorem. The drift force is more significant for ships with a higher block coefficient. Considering the fact that only head waves were taken into account in this paper, the drift force acts in the negative direction of the longitudinal axis of the ship coordinate system. Numerical approximation of motions and loads due to drift forces requires knowing the second-order wave loads at different frequencies of incoming waves. Low-frequency load can be described by the quadratic transfer function QTF of incoming waves and diffraction and radiation of wave fields. To determine the QTF function it is therefore necessary to solve the problem of the second-order wave loads.

Low-frequency second-order wave loads occur at frequencies equal to the difference of two wave frequencies in all possible combinations at a certain sea state. They are proportional to the products of wave amplitudes (QTF) and consist of two parts: one part depending on the square products of the wave field of the first order, and the second part of incoming wave and diffraction potentials of the second order, which can be determined based on the Froude-Krylov force of the second order and Haskind integral on the body surface as follows [9]:

$$F(\omega_i, \omega_j) = F_q(\omega_i, \omega_j) + F_p(\omega_i, \omega_j) \quad (5)$$

where $F_q(\omega_i, \omega_j)$ is the first-order wave load, $F_p(\omega_i, \omega_j)$ is the second-order wave load, ω_i , ω_j are the incoming wave frequencies.

While the first part can be determined by wave diffraction and radiation solution of the first order, the second part of the second-order wave load slowly converges and includes gradients of velocity potential [9].

The Haskind integral allows the elimination of the unknown diffraction potential by replacing it with radiation potential [1]:

$$\iint_S \phi_7 \frac{\partial \phi_k}{\partial n} dS = -\iint_S \phi_k \frac{\partial \phi_0}{\partial n} dS, k = 1 \dots 6 \quad (6)$$

where ϕ_7 is the diffraction potential, ϕ_k is the radiation potential, ϕ_0 is the incoming wave potential, S_0 is the wetted surface.

Since the QTF is assumed to be a regular function of the incoming wave frequencies ω_i and ω_j and taking into account the relation $\Delta\omega = \omega_i - \omega_j$, the QTF is developed as Taylor series expansion:

$$F(\omega_i, \omega_j) = F_0(\omega_i) + F_1(\omega_i)\Delta\omega + F_2(\omega_i)(\Delta\omega)^2/2 + \dots \quad (7)$$

The zeroth-order term $F_0(\omega_i)$ represents the load due to integration of pressure along the hull wetted surface, i.e. the drift force. The often used approximation proposed by Newman when determining that term can significantly underestimate the second-order wave loads and also can provide wrong phase shift since the approximation is a real function while the QTF is a complex function. The zeroth-order term depends on the incoming wave frequency as the mean value of two frequencies $(\omega_1 + \omega_2)/2$, the first-order term is linearly proportional to the difference of two frequencies $\omega_1 - \omega_2$ and the second-order term is proportional to the square of difference between the two frequencies.

The term $F_1(\omega_i)$ is composed of four parts:

$$F_1(\omega_i) = F_q^1 + F_{p1}^1 + F_{p2}^1 + F_{p3}^1 \quad (8)$$

where F_q^1 is the contribution of the first-order wave load, F_{p1}^1 is the contribution of the second-order incoming wave load and diffraction waves, F_{p2}^1 is the second-order correction of the boundary condition on the hull, F_{p3}^1 is the effect of forcing pressure over the free surface (second-order correction of the boundary condition on the free surface).

The term $F_2(\omega_i)$ does not have analytical expression but it can be defined as the difference of the total QTF function of the bichromatic wave field and the zeroth and the first-order term previously mentioned:

$$F_2(\omega_i) = \frac{2 \left[F\left(\omega_i + \frac{\Delta\omega}{2}, \omega_i - \frac{\Delta\omega}{2}\right) - F_0(\omega_i) - F_1(\omega_i)\Delta\omega \right]}{(\Delta\omega)^2} \quad (9)$$

4. Added resistance in regular waves of the S-175 container ship

To compare the experimentally obtained results of the ship added resistance with those calculated numerically, the S-175 container ship model test data were used [6]. Tests were conducted on a stationary model and at three different speeds. Regular head waves (180°) of 30 mm wave height and different ratios of wavelength and the length of the model λ/L were generated. The principal dimensions of the full scale S-175 container ship and its model are given in Table 1 and the body plan is shown in Figure 1. The hull mesh created using hydrodynamic software [7] is shown in Figure 2.

In order to numerically calculate the added resistance, hydrodynamic software was used [7]. The software is based on the potential flow theory and provides the solutions of the first- and second-order motions and loads. The hull characteristics and parameters of the waves are necessary as an input. It allows calculation of diffraction and radiation components of the ship

added resistance, the interaction between the ship and the waves and the interaction between several bodies, etc. Since the software is based on the potential flow theory, a fictitious force in dynamic motion equations is added in order to take into account the energy dissipation and damping to avoid the infinite response at resonant frequencies.

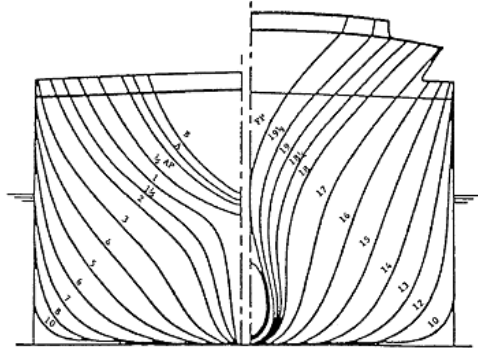


Figure 1 Body plan of the S-175 container ship [6]

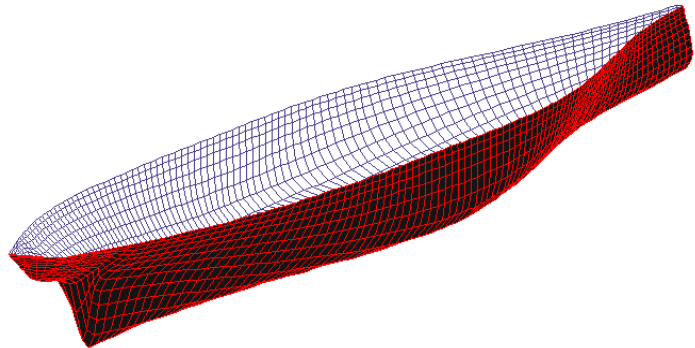


Figure 2 Panel model of the S-175 container ship [7]

Table 1 Principal dimensions of the S-175 container ship

	Full scale	Model
Length L (m)	175.00	3.00
Breadth B (m)	25.40	0.435
Draft of fore peak T_F (m)	7.00	0.120
Draft of midship T (m)	9.50	0.163
Draft of aft peak T_A (m)	12.02	0.206
Displacement volume ∇ (m ³)	24154.13	0.122
Block coefficient C_B	0.572	0.572
Position of LCG x_G (m)	-	-0.141
Radius of gyration k_{yy}/L	-	0.239

The second-order low-frequency wave loads using software package [7] can be solved by three formulations:

1. The **near field formulation** needs the evaluation of the first-order wave field around the hull and along the waterline, as well as the first-order motions caused by that field. Drift load is determined by direct numerical integration of pressure along the defined hull form wetted surface.
2. Another formulation based on the momentum theorem for the horizontal drift forces involves the first-order wave field in the far field and is often called **far field formulation** and is preferable in practice thanks to its better convergence of drift forces and stability. However, the far field formulation cannot provide the accurate vertical drift loads and the low-frequency loads QTF which can be both of great importance in shallow water.
3. The **middle field formulation** combines the advantages of both near field and far field formulation providing sufficiently accurate numerical results and possibility of calculating all components of drift loads and low-frequency loads QTF as the near field formulation. The method is based on the analysis in a finite volume limited by

the hull and a control surface surrounding the hull. Pressure integration over the hull surface is evaluated with a semi-analytical method using the far field potentials like the previous formulation.

The low-frequency wave loading $F(t)$ is evaluated in time domain after the determination of the quadratic transfer function QTF and complex amplitudes and it is defined by a double summation [9]:

$$F(t) = \Re \left\{ \sum_{i=1}^N \sum_{j=1}^N F'_{ij} a_i a_j^* \exp[-i(\omega_i - \omega_j)t] \right\} \quad (10)$$

where F'_{ij} is the drift force in waves, a_i is the complex amplitude of incoming wave, a_j^* is the complex conjugate amplitude of incoming wave, ω_i, ω_j are the incoming wave frequencies.

Within this study only the constant part of the quadratic transfer function QTF is taken into account, i.e. the diagonal elements of QTF matrix which represent the drift forces in waves as constant values at certain frequencies.

The added resistance force in waves is defined as:

$$F(t) = F'_{jj} a_j^2 \quad (11)$$

where a_j is the amplitude of head incoming waves.

Finally, the added resistance coefficient in waves is given by the following relation:

$$C_{xw} = \frac{F(t)}{\rho g a_j^2 B^2 / L} \quad (12)$$

where ρ is the density of water, g is the gravity acceleration, B is the model breadth, L is the model length.

In this paper previously described numerical procedure is used to calculate constant part of the second-order wave load which represents added resistance due to waves.

The calculations are conducted for four Froude numbers and four velocities of the S-175 container ship model shown in Table 2. Based on the model test data, regular head waves were applied on the model. The frequencies of generated incoming waves were determined on the basis of the range of the ratio $\lambda/L = 0.3-2.0$ [6]. Wave frequencies were calculated using the dispersion equation for a limited depth of water considering that the model was tested in a tank of 3.5 meters depth.

Table 2 Froude numbers and velocities of the S-175 container ship model

Froude number Fr	Velocity v , m/s
0	0
0.087	0.472
0.148	0.803
0.198	1.074

The comparison of the calculation results of added resistance coefficients obtained by using different methods and experimentally obtained coefficients is shown in Figures 3-6 based on the ratio of the wavelength and the length of the model.

The abbreviations of the aforementioned methods of calculating the second-order low-frequency wave loads are the following:

- PRE - direct integration of pressure along the wetted surface (near field formulation),
- MOM - method based on the momentum theorem (far field formulation),
- CSF - method using control surface and far field potentials (middle field formulation).

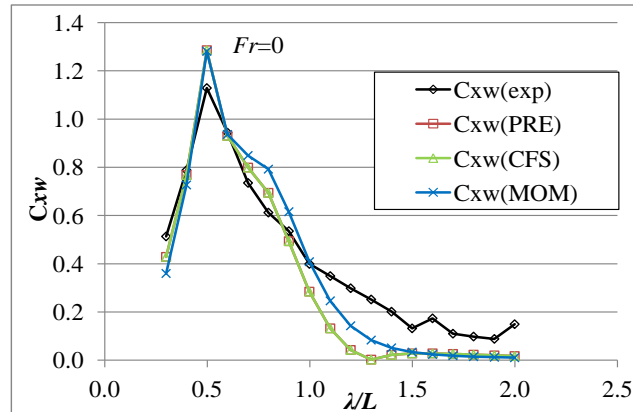


Figure 3 Comparison of added resistance coefficients for Froude number $Fr = 0$

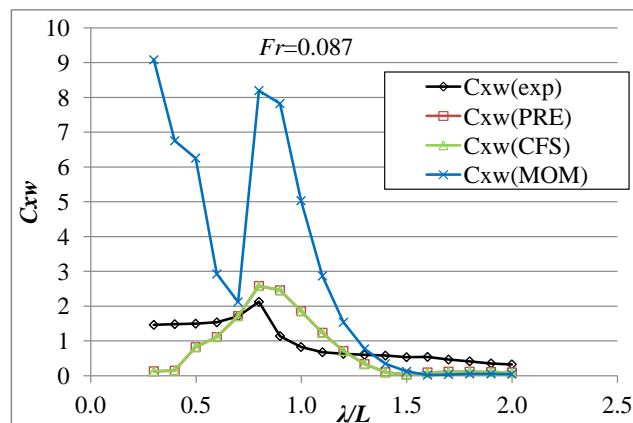


Figure 4 Comparison of added resistance coefficients for Froude number $Fr = 0.087$

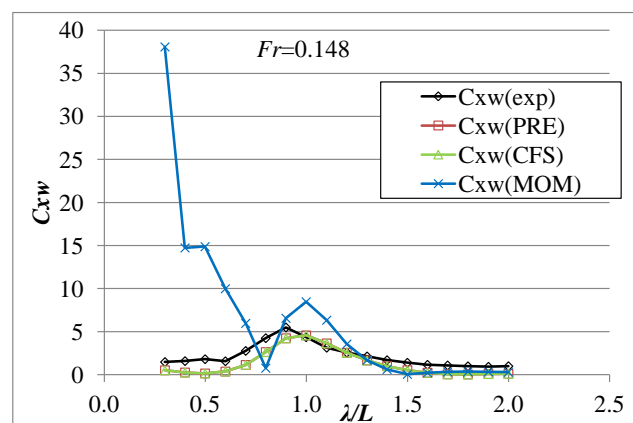


Figure 5 Comparison of added resistance coefficients for Froude number $Fr = 0.148$

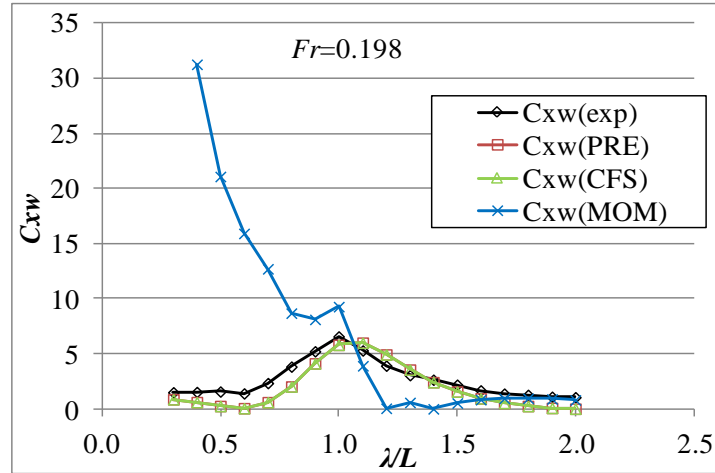


Figure 6 Comparison of added resistance coefficients for Froude number $Fr = 0.198$

As shown in Figures 3-6 there are certain deviations in the numerically calculated added resistance coefficients compared to the experimental data. Since the calculations are based on the potential flow theory and certain simplifications were introduced, it was not possible to fully simulate the real flow and the forces and loads acting on the model during the tests. The method based on the integration of pressure along the hull wetted surface (PRE) and the method that uses a combination of pressure integration and analysis of volume bounded by the ship hull and the control surface (CFS) give identical results and show satisfactory agreement with the experimental data.

The method based on the momentum theorem (MOM) gives approximate results only for a stationary ship, i.e. at Froude number equal to zero. During the progress of the ship at a certain speed the method gives large deviations of the results with regard to other two methods. It is evident that these deviations are increasing as the speed increases. This method does not provide satisfactory results at lower frequencies and in the case of shallow water, which was an assumption in this analysis, since it is a formulation that uses far potentials when simulating flow.

Based on the added resistance coefficients, it is evident that a significant increase of added resistance and total ship resistance will occur at certain frequencies i.e. wavelengths. The frequencies at which the mentioned increase in ship resistance occurs will be higher if the ship sails at a greater speed. Such data are useful when planning a route of the ship depending on sea conditions and the nominal speed of navigation.

Using the experimentally obtained coefficients of added resistance in regular waves [6], the ship added resistance in real conditions was determined based on the Froude similarity of the tested model and the ship. Using the known added resistance coefficients, the added resistance was calculated according to the following expression:

$$R_{AW} = C_{xw} \rho g \zeta_a^2 \frac{B^2}{L} \quad (13)$$

Added resistance in dependence on the ratio of wavelength and length of the model λ / L is shown in Figure 7.

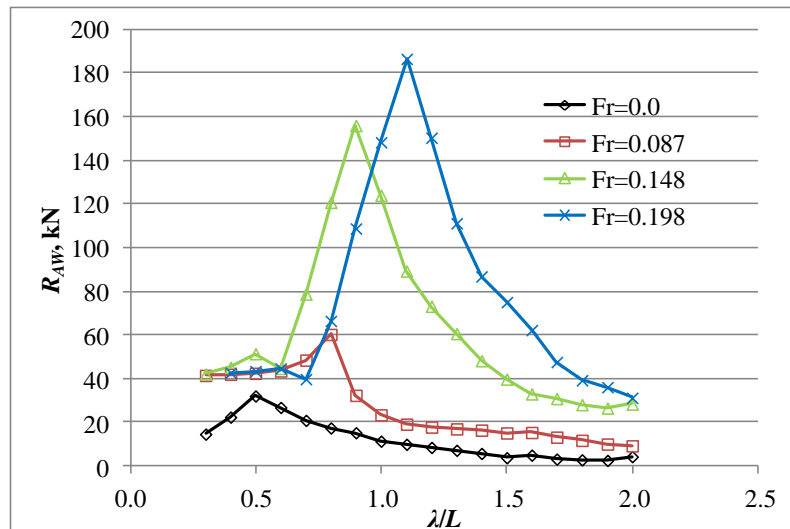


Figure 7 The ship added resistance for different Froude numbers

5. Added resistance of intact and damaged ship at defined sea states

A damaged ship often needs to be removed by towing after a maritime accident. The water that has penetrated inside the hull not only affects and threatens the stability of the ship, but has also an impact on the ship seakeeping characteristics. It is necessary to know what impact has the fluid inside the hull that moves along with the ship on the global motions and loads in order to determine the method and route of towing.

Determination of seakeeping characteristics in this case is based on the coupled motions of ship motions as a rigid body and fluid sloshing inside the tank. Hydrodynamic problem of solving the coupled motions and loads of the ship and internal fluid inside the tanks is also considered under assumptions of linear potential flow theory. Sloshing and seakeeping of the ship are two separately observed hydrodynamic parts. Since a linear potential flow theory is considered, within the tanks there is no damping and the problem of resonant frequencies of fluid motion in the tank and possibly unrealistic responses to these frequencies may occur. It is therefore necessary to correct the hydrodynamic model by using damping factors which may be based on empirical data or experimental tests.

Hydrostatic forces and moments acting on the vessel shall be determined by integrating the pressure along the hull wetted surface. Hydrodynamic loads acting on the ship can be determined based on flow velocity potential of wave diffraction and its components and radiation potential.

In the case of sloshing inside the tank, a similar procedure is used to determine the hydrodynamic and hydrostatic loads. Since according to the liner theory the free surface of a fluid inside the tank is constantly horizontal, a correction of the tank walls vertical coordinates is used in the integration of pressure along the wetted surface of the walls. The calculations are conducted based on the center of the free surface of liquid inside the tank.

In order to simulate the impact of sloshing inside the hull due to damage, a flooded tank is generated inside the mesh of the S-175 container ship as shown in Figure 8. The volume of liquid in the tank is 48.49% of the volume displacement of the undamaged ship on a draft equal to $T = 12.01667$ m, or 32.74% of the volume displacement of the damaged ship that has vertically immersed for $\Delta T = 3.483$ m.

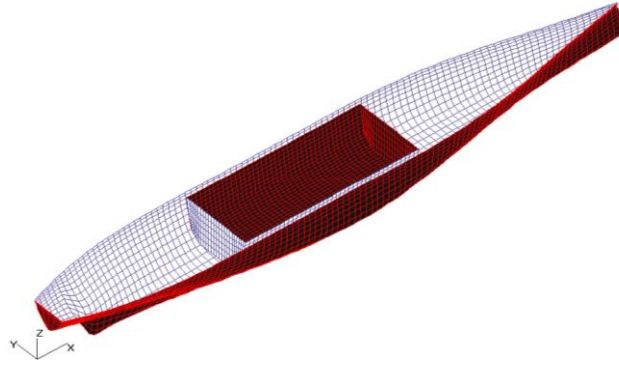


Figure 8 Panel model of the S-175 container ship with internal tank [7]

Fluid inside the tank was first treated as the ship added displacement mass but without affecting the global motions and loads. After that, the seakeeping characteristics of the ship were calculated considering also the sea water inside the tank in order to determine the impact of motion and loads of liquid in the tank on global ship motions at a defined sea state. The calculation is based on the Tabain wave spectrum as a modification of JONSWAP (Joint North Sea Wave Project) spectrum for the Adriatic Sea to assess the safety of towing the ship in the sea conditions characteristic for the Adriatic Sea.

The Tabain wave spectrum is defined as follows:

$$S_{\zeta}(\omega) = 0.862 \frac{0.0135 g^2}{\omega^5} e^{\left[\frac{5.186}{\omega^4 H_{1/3}^2} \right]} 1.63^p \quad (14)$$

where

$$p = e^{\left[\frac{-(\omega - \omega_m)^2}{2\sigma^2 \omega_m^2} \right]} \quad (15)$$

$$\omega_m = 0.32 + \frac{1.8}{H_{1/3} + 0.60} \quad (16)$$

$$\sigma = \begin{cases} 0.08 & \text{for } \omega \leq \omega_m \\ 0.1 & \text{for } \omega > \omega_m \end{cases} \quad (17)$$

According to theoretical predictions the most likely maximum significant wave height for the period of 20 years is 7.20 m and for the period of 100 years it is 8.57 m [10]. For sea states with the aforementioned significant wave heights, the mean value of added resistance of the intact and the damaged S-175 container ship is calculated. The mean added resistance is determined based on a summation or an integration of the contributions over the whole frequency range as follows [8]:

$$\overline{R_{AW}} = 2 \int_0^{\infty} \left| \frac{R_{AW}}{\zeta_a^2}(\omega) \right| S_{\zeta}(\omega) d\omega \quad (18)$$

where $\frac{R_{AW}}{\zeta_a^2}(\omega)$ stands for drift force at certain wave frequency obtained using hydrodynamic software [7].

Relation between the mean added resistance and the regular wave amplitude squared is linear, and the wave spectrum is assumed to be ideal. The spectral function is obtained by multiplying drift force at certain frequency and the corresponding ordinate of the Tabain

spectrum. The first spectral moment is determined by the trapezoidal rule of integration. The mean values of the added resistance at two defined sea states determined by using the equation (18) are based on the absolute values of the drift forces obtained using the hydrodynamic software. For a certain frequency range, these values are shown in dependence on the ship speed in Figures 9 and 10.

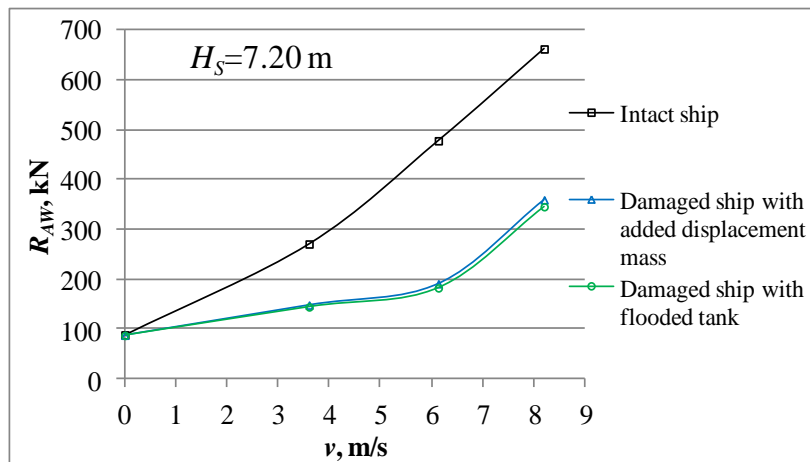


Figure 9 Mean value of added resistance for sea state with significant wave height $H_S = 7.20$ m

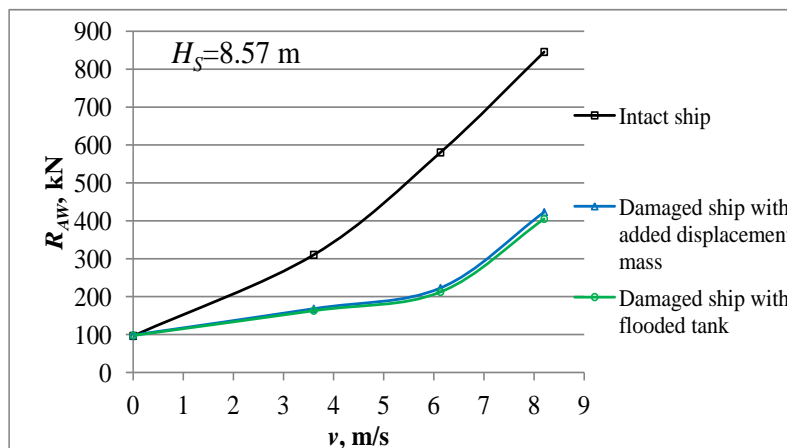


Figure 10 Mean value of added resistance for sea state with significant wave height $H_S = 8.57$ m

Since the second-order wave load or quadratic transfer function QTF was approximated using the Newman formulation with the zeroth-order term only, the ship added resistance is thus underestimated. In order to compare the calculated results with the data from the literature and to substantiate this assumption, the calculation of added resistance coefficients of the S-175 container ship for Froude number $Fr = 0.275$ was conducted. A comparison is made between the calculated results and the data obtained experimentally [11] and numerically using the three-dimensional nonlinear numerical panel method based on the Rankine potential (the Green function) that allows modelling of the nonlinear steady flow in the frequency domain [12]. The results are shown in Figure 11.

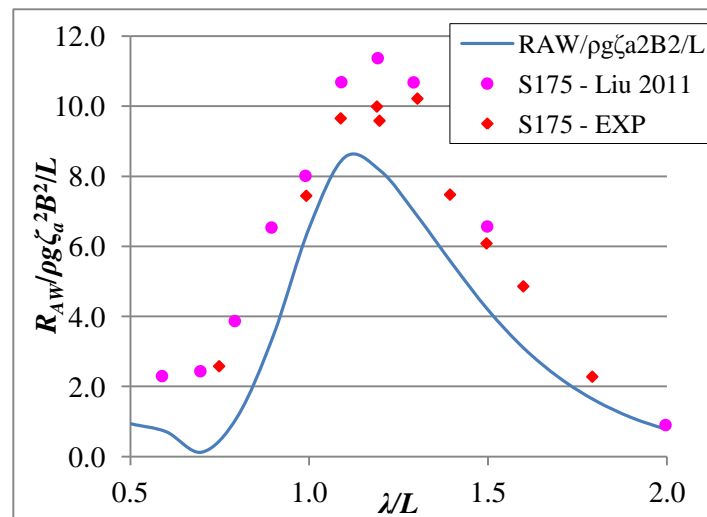


Figure 11 Comparison of obtained results of added resistance coefficients with data from literature [11], [12]

It can be concluded from Figure 11 that the curve form of the calculated values of dimensionless added resistance follows the trend of the experimental and numerical data from the literature. Since in this paper added resistance is based on the potential theory, there are certain deviations in relation to the experimental data. It may also be noted that the extreme value of the resulting dimensionless function of added resistance in dependence on the ratio λ/L occurs at a lower λ/L ratio than in the case of the numerical and experimental data from the literature.

6. Conclusion

Three methods available in the software package [7] were used to calculate the ship added resistance. While the method that uses the direct integration of pressure along the hull wetted surface and the method that analyzes the finite volume around the ship within the control surfaces in conjunction with the integration of the pressure give similar results compared to the experimental data, the method which uses the momentum theorem provides a significant deviation from the experimental results, except in the case of the stationary ship. As the method of momentum theorem uses far potentials, it is almost unusable in the case of shallow water.

The quadratic transfer function QTF that represents low-frequency wave loads is approximated by zeroth-term only or drift force which is constant at a given frequency of incoming wave. This, the so called Newman's approximation simplifies the calculation of added resistance but also significantly underestimates the ship added resistance. Since only the head waves were generated in this paper, it was assumed that Newman's approximation of QTF function would provide satisfactory accuracy of the second-order wave loads. Based on the obtained results of the added resistance of the intact and the damaged ship at defined sea states, it can be concluded that the two calculation models used in this paper, i.e. damage simulated as increased displacement mass and as a flooded tank, give approximately the same results. The intact ship has a significantly larger added resistance than the damaged ship. Since the damaged ship with a larger displacement mass and declining natural frequency has significant response at lower frequencies where the spectral energy at certain sea state is lower, the added resistance will be also smaller than the added resistance of the intact ship.

It is necessary to adjust the boundary conditions and carefully determine the damping factor which will prevent infinite responses especially in the case of sloshing inside the ship

tank. Despite numerous simplifications, the linear potential flow theory gives satisfactory results and greatly simplifies the calculation of the interaction of the ship and incoming waves.

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Ivana Martić
Nastia Degiuli, nastia.degiuli@fsb.hr
Ivan Čatipović
University of Zagreb,
Faculty of Mechanical Engineering and Naval Architecture,
Ivana Lučića 5, 10000 Zagreb, Croatia