Deliverable D4.8

Validation results of numerical tools for the ROPAX ship

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**Document History**

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Abstract
This deliverable presents the results of the numerical analysis of the manoeuvrability of the CALMAC RoPax ferry. First the approach for the determination of a complete set of calm water manoeuvring derivatives (at 5 kn advance speed) with the RANS code NEPTUNO is explained briefly and corresponding literature for deeper insight is cited. After examination of the quality of the numerical computations, the results of the calm water manoeuvring forces for different conditions are shown. Then the mean forces in waves are calculated with the panel code GL Rankine and modelled. Finally some exemplary turning circle tests in calm water and waves are shown and discussed.

This report represents the work of a task added at a late stage of the project and results from the cooperation between TUB and UDE.

Summary Report:

Introduction
The task of this deliverable “Validation results of numerical tools for the ROPAX ship” was defined at a late stage of the project. The investigated ship ‘MV Isle of Lewis’ of Caledonian MacBrayne is an existing seagoing ship operating in the north Atlantic and the only actually built ship investigated in the project.

State of the Art
The prediction of ship manoeuvrability of seagoing ships is usually done for calm water conditions only. To assess IMO conformity for instance, zigzag and turnings circle tests are analysed. Regarding the evaluation of the ships manoeuvrability in waves, at most speed loss in waves is investigated. Ship motions are mostly predicted with potential flow methods.

A widely used method for manoeuvring prediction is based on a set of hydrodynamic coefficients, obtained from PMM or CPMC model tests, which allow calculating the forces and moments acting on the ship in the course of simulated manoeuvres. In recent years, the use of pure CFD techniques to determine these coefficients has increased. However, they mostly do not include yet the effect of waves and the used body force models replacing the propeller mostly rely on simple potential theories.

A complete analysis of manoeuvrability under real environmental conditions is usually not performed for merchant ships during the design process.

Value added to SHOPERA
An extensive calculation of the manoeuvring behaviour of an existing sea going RoPax ship has been performed. Calculations of mean wave forces are compared with experimental data obtained in WP3. Any desired rudder manoeuvre can be quickly predicted using a mathematical model based on hydrodynamic coefficients for approximating the forces and moment acting on the ship.

Achievements
TUB determined a calm water manoeuvring coefficient set for the RoPax at 5 kn advance speed using a RANS method.
UDE calculated the mean forces and moments on the hull due to waves of several lengths and encounter angles.
TUB inserted the mean wave forces in the mathematical model and simulated turning circle tests in waves.
Not achieved

Input from other Deliverables
Deliverable D3.3, Report on Model Tests at MARINTEK

Exploitation of results

This executive summary may be published outside the SHOPERA consortium. **YES/NO**

<table>
<thead>
<tr>
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<th>Approved by</th>
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<tr>
<td>Sebastian Uharek (TUB)</td>
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<td>Andrés Cura Hochbaum (TUB)</td>
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Name of internal reviewer and date of acceptance:

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Apostolos Papanikolaou, 21/10/2016
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1 Introduction and method outline

The following report describes the prediction of the manoeuvring behaviour for the ROPAX vessel 'MV Isle of Lewis' of Caledonian MacBrayne in regular waves. The manoeuvres are simulated by solving the rigid body motion equations in 4DOF, as explained in (Cura Hochbaum A., 2006). The solved system of equations in a hybrid coordinate system which does not follow roll motions is listed below.

\[
m \left[ \ddot{U} - \dot{\psi} V - x_G \dot{\psi}^2 + z_G \left( 2 \dot{\psi} \dot{\varphi} \cos \varphi + \dot{\psi} \sin \varphi \right) \right] = X
\]

\[
m \left[ \ddot{V} + \dot{\psi} \dot{U} + x_G \dot{\psi} + z_G \left( \left( \dot{\psi}^2 + \dot{\varphi}^2 \right) \sin \varphi - \dot{\varphi} \cos \varphi \right) \right] = Y
\]

\[
I_{xx} \dot{\psi} - I_{xz} \psi \cos \varphi + \left( I_{zz} - I_{yy} \right) \dot{\psi}^2 \sin \varphi \cos \varphi - m z_G \cos \varphi \left( \dot{V} + U \psi \right) = K
\]

\[
\left( I_{yy} \sin^2 \varphi + I_{zz} \cos^2 \varphi \right) \dot{\psi} + 2 \left( I_{yy} - I_{zz} \right) \dot{\psi} \dot{\varphi} \sin \varphi \cos \varphi - I_{xx} \left( \ddot{\varphi} \cos \varphi - \dot{\varphi}^2 \sin \varphi \right) +
\]

\[
m x_G \left( \dot{V} + U \psi \right) + m z_G \sin \varphi \left( \dot{U} - V \dot{\psi} \right) = N
\]

The forces on the right hand side are approximated using a mathematical model. The model splits the forces into two independent contributions, one stemming from the wave forces and the other one from the manoeuvring forces in calm water, see equations (2).

\[
X = X_c + X_w
\]

\[
Y = Y_c + Y_w
\]

\[
K = K_c + K_w
\]

\[
N = N_c + N_w
\]

The hydrodynamic coefficients for calm water manoeuvring were determined by TUB by means of virtual captive model tests, performed with the in-house RANS code NEPTUNO. The results are used to determine a set of manoeuvring derivatives of Abkowitz type (Abkowitz, 1964). The used procedure is thoroughly explained in (Cura Hochbaum & Uharek, 2016) and in the SHOPERA Deliverable D2.9 and will therefore only be summarized briefly here. Examples of previous successful application of this procedure to manoeuvring problems in calm water can be found in (Cura Hochbaum A., 2006), (Cura Hochbaum, Vogt, & Gatchell, 2008) and (Cura Hochbaum & Uharek, 2014).

The wave contribution to the right hand side of the motion equation only takes the mean effect of wave forces into account. These mean forces can be calculated using either a RANS code or a potential flow method. In the present case the computations were performed by the UDE using the potential flow code GL Rankine (Söding, von Graefe, Shigunov, & el Moctar, 2012). The mathematical model used to approximate these forces is described in (Uharek & Cura Hochbaum, 2015) and in the SHOPERA Deliverable D2.9.
2 Numerical computations for calm water virtual captive tests

2.1 Numerical code

The numerical simulations needed are performed with an updated version of the RANS code NEPTUNO, see (Cura Hochbaum & Vogt, 2002). The code uses a finite volume method to solve the governing equations on a ship fixed, block-structured grid with non-matching interfaces. Pressure and velocities are coupled by means of the SIMPLE algorithm (Pantakar & Spalding, 1972) and the turbulence is modelled with the standard k-ω model from (Wilcox, 1993). A body force model is used instead of the propeller during the virtual captive tests. Hereby a force distribution in a defined region of the grid does approximate the effect of the (disregarded) propellers on the flow, depending on the current velocities in the propeller plane.

2.2 Main characteristics of the ship and numerical grid

The investigated ship is a ROPAX vessel of Caledonian MacBrayne. The main characteristics are shown in Table 1. Note that the displacement ∑ is given including all appendages. The rudder and propeller information was extracted from the available AutoCAD drawings on the SHOPERA website. The hull geometry was provided by MARINTEK in form of a Rhino file.

<table>
<thead>
<tr>
<th>Ship</th>
<th>Full scale</th>
<th>Model scale</th>
</tr>
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<tbody>
<tr>
<td>scale</td>
<td>1</td>
<td>25</td>
</tr>
<tr>
<td>LPP</td>
<td>90 m</td>
<td>3.6 m</td>
</tr>
<tr>
<td>BWL</td>
<td>17.8 m</td>
<td>0.712 m</td>
</tr>
<tr>
<td>T</td>
<td>4.2 m</td>
<td>0.168 m</td>
</tr>
<tr>
<td>∑</td>
<td>3771 m³</td>
<td>0.241 m³</td>
</tr>
<tr>
<td>CB</td>
<td>0.56</td>
<td>0.56</td>
</tr>
<tr>
<td>U</td>
<td>5 kn</td>
<td>0.514 m/s</td>
</tr>
<tr>
<td>GM</td>
<td>2.396</td>
<td>0.096</td>
</tr>
<tr>
<td>FR</td>
<td>0.087</td>
<td>0.087</td>
</tr>
<tr>
<td>RS</td>
<td>2.31 10³</td>
<td>1.85 10⁶</td>
</tr>
<tr>
<td>Propeller</td>
<td></td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>3 m</td>
<td>m</td>
</tr>
<tr>
<td>x from AP</td>
<td>2.7 m</td>
<td>m</td>
</tr>
<tr>
<td>Y from CP</td>
<td>±3 m</td>
<td>m</td>
</tr>
<tr>
<td>Z from K</td>
<td>1.65 m</td>
<td>m</td>
</tr>
<tr>
<td>Rudder</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Y from CP</td>
<td>±3 m</td>
<td>m</td>
</tr>
</tbody>
</table>

Due to the very low Froude number, the effects of the free surface are considered negligible and therefore all simulations are performed for the double body flow condition.
All appendages of the vessel except for the stabilizer fins were taken into account. For the determination of the rudder angle dependent forces, six rudder angles ($\delta = 0^\circ, 10^\circ, 20^\circ, 25^\circ, 30^\circ, 35^\circ$) were considered. For each rudder angle a separate grid box was built and merged with the other parts using non-matching interfaces, indicated in red in Figure 1.

Figure 1: Fine grid for the RoPax with non-matching interface for the rudder box

The grids were generated with the commercial software GridPro, the total cell number being 3.8 million. In order to check the grid dependence of the solution, the grid was coarsened two times, resulting in cell counts of 1.5 million and 0.5 million cells and selected calculations were repeated.

The RANS simulations were performed using wall functions in the boundary layer of the ship hull. The non-dimensional wall distance $y^*$ was chosen to be 80.

2.3 Computational setup and pre-Checks

Before the computations for the determination of the hydrodynamic coefficients were started, some preliminary checks were performed.

- Figure 2 shows that the $y^*$ values of the cell centres of the first cell layer at the wall lie in an acceptable range throughout the whole grid

- The convergence of the individual computations was also checked, as shown in Figure 4
- The calm water resistance was compared with MARINTEK measurements. The measured resistance was just 2.2 N for the model at 5 kn. The computed calm water resistance on the fine grid is 2 N.

- The pressure distribution on the rudders for the vessel sailing straight ahead is shown in Figure 3. As can be seen the port rudder has a high pressure region (red) at the top, whereas on the starboard rudder it is located at the bottom. This confirms that both propellers are turning inside over the top.
2.4 Test setup

The propellers in the simulations are replaced by pre-calculated body force distributions for a stock propeller. These forces are obtained by RANS calculations for the isolated rotating propeller in uniform inflow and are stored in a database. During the computation, body forces according to the inflow condition are added to the right hand side of the momentum equation in all cells within prescribed propeller regions.

All virtual model tests are performed at the model self-propulsion point (MSPP). To determine the model self-propulsion point, preliminary computations with varying revolution rates have been performed. The result is shown in Figure 5. The revolution rate leading to a balance of (effective) resistance and the sum of both propeller thrusts is 5.6 rps.
In order to determine all necessary hydrodynamic coefficients for calm water, five dynamic tests and 132 static tests, listed in Table 2, were performed on all three grids. The considered rudder angles ($\delta = 0^\circ, \pm 10^\circ, \pm 20^\circ, \pm 25^\circ, \pm 30^\circ, \pm 35^\circ$) were coupled with a static drift motion ($\beta = -20^\circ, -10^\circ, 10^\circ, 20^\circ$), steady yaw motion ($r' = r_L/u = -0.7, -0.35, 0.35, 0.7$) and longitudinal speed variation (-50%, -25%, +25%). Since the ship is equipped with stabilizer fins, the roll motion is assumed to be rather small and not considered in the mathematical model.

The dynamic tests were performed mainly to determine hydrodynamic coefficients depending on accelerations. Due to the very low speed of 5 kn considered, a motion period of 50 seconds was used to avoid memory effects. The computed tests were: one pure surge test with a non-dimensional amplitude of $\hat{u}' = \hat{u}/u_0 = 0.2$, one pure sway test with $\hat{v}' = \hat{v}/u_0 = 0.5$, one pure yaw test with $\hat{r}' = \hat{r}L/u_0 = 0.7$, and two combined sway/yaw tests with $\hat{v}' = -0.4$, $\hat{r}' = 0.7$ and $\hat{v}' = 0.4$, $\hat{r}' = 0.4$. The dynamic tests were computed using 2500 time steps per motion period and 10 SIMPLE iterations per time step.

### 2.5 Results of virtual captive tests in calm water

All simulations are performed on the HPC-Cluster of the Department Dynamics of Maritime Systems of TUB using a single core per computation. The average computation time for the static tests on the coarse grid was 11 hrs, 30 hrs on the medium grid and 72 hrs on the fine grid. Since only a single core is used per computation, all calculations can be performed at once. The computation time for the dynamic tests is 72 hrs on the coarse grid and 216 hrs on the medium grid for three complete motion periods.

<table>
<thead>
<tr>
<th>Rudder angle / drift angle ($\delta = 0^\circ, \pm 10^\circ, \pm 20^\circ, \pm 25^\circ, \pm 30^\circ, \pm 35^\circ$)</th>
<th>11</th>
</tr>
</thead>
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<tr>
<td>Rudder angle / drift angle ($\beta = -20^\circ, -10^\circ, 10^\circ, 20^\circ$)</td>
<td>4 x 11</td>
</tr>
<tr>
<td>Rudder angle / yaw rate ($r' = -0.7, -0.35, 0.35, 0.7$)</td>
<td>4 x 11</td>
</tr>
<tr>
<td>Rudder angle / surge speed (-50%, -25%, +25%)</td>
<td>3 x 11</td>
</tr>
<tr>
<td>Yaw rate / drift angle</td>
<td>4 x 4</td>
</tr>
<tr>
<td><strong>total</strong></td>
<td><strong>148</strong></td>
</tr>
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</table>

Figure 5: Thrust and Resistance for determination of MSPP
2.5.1 Influence of appendages

In order to capture the influence of the appendages, two different grids were built. Figure 6 shows the different vortical structures at a 20° drift angle in front of the propeller plane for both cases. As can be seen the wake field is significantly influenced by the struts and the corresponding vortices. On the windward side, a quite large vortex emerges from the struts and on the leeward side the vortex emerging from the skeg gets overestimated, without capturing the interaction with the appendages. Table 3 shows the differences in the longitudinal and side force and especially as in the yaw moment are not negligible.

![Figure 6: Vortex structures in front of the propeller plane during static drift (β=20°), with and without appendages](image)

Figure 6: Vortex structures in front of the propeller plane during static drift (β=20°), with and without appendages
Table 3: Forces with and without appendages at β=20°

<table>
<thead>
<tr>
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<th>$F_x$</th>
<th>$F_y$</th>
<th>$M_z$</th>
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<tr>
<td>with appendages</td>
<td>-3.837 N</td>
<td>18.747 N</td>
<td>9.169 Nm</td>
</tr>
<tr>
<td>w/o appendages</td>
<td>-3.937 N</td>
<td>18.371 N</td>
<td>9.638 Nm</td>
</tr>
<tr>
<td>difference</td>
<td>-2.5%</td>
<td>2%</td>
<td>-4.9%</td>
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2.5.2 Static tests and grid dependence

The results for all three grids are presented in the figures in this section. All mean forces and moments are made non-dimensional as shown in equations (3). Note that $U$ is the ship velocity in the hybrid coordinate System.

$$F' = \frac{F}{\frac{1}{2} \rho L_p^2 U^2}$$

$$M' = \frac{M}{\frac{1}{2} \rho L_p^3 U^2}$$

Figure 7: Rudder angle dependent forces and moments
Figure 7 shows the longitudinal and side force as well as the heeling and yaw moment for several rudder angles obtained from the static virtual tests. The grey lines are additional computations without body forces. It is clearly visible that the correct treatment of the propeller effect is crucial for the accuracy of the rudder angle dependent forces. The forces show a fair grid convergence, especially in the side force and yaw moment. The stall is not captured correctly on the coarse grid, even though the linear region is predicted well.

The forces show a fair grid convergence, especially in the side force and yaw moment.

Figure 8: Drift angle dependent forces and moments

The Figure 8 shows the dependence of the forces and moments on the drift angle or transversal speed \((v')\). The influence of the grid resolution on the longitudinal force can be neglected, since for the medium and fine grid the absolute value of this force is always less than 0.03. This is very small compared to the longitudinal force that arises due to rudder deflection (0.6).

Figure 9 shows the grid dependence of the forces depending on the yaw rate. Again the medium and fine grid yields very similar results, whereas the coarse grid shows significant differences in the side force and heeling moment.
2.6 Dynamic tests

Since the computation of a dynamic test is very time consuming, the computations on the fine grid have been performed for two motion periods instead of three. Figure 10 shows the traces of the side force and yaw moment for the pure sway test. It can be seen that all three grids yield the same results. In the case of the pure yaw test (see Figure 11), there are some slight differences in the side force between the three grids, as has been observed in the static tests before. The slight fluctuations which can be observed in all traces in Figure 11 occur due to numerical reasons, but do not influence the determined coefficients, because the Fourier analysis of the time trace (which is necessary to determine the hydrodynamic coefficients) filters out this high frequency oscillations.
Figure 10: Pure sway test

Figure 11: Pure yaw test
2.7 Evaluation of calm water manoeuvring coefficients

The hydrodynamic coefficients depending exclusively on motion parameters are determined by Fourier analysis of the dynamic tests. Rudder angle dependent coefficients are determined by multilinear regression of the forces obtained from the static virtual tests shown in Table 2. Figure 12, Figure 13 and Figure 14 show the reconstructions of the forces by means of the determined coefficient set. It should be noted, that all coefficients purely depending on sway and yaw motions as well as the respective coupling terms stem from the dynamic tests (solid lines in the figures) and reproduce the forces determined in the static tests (dots in the figures) quite well.

All traces show the expected behaviour. On the left hand side of Figure 13 for instance, it can be seen that for the case shown in red, where the rudder is set to port (δ positive) and the ship is drifting towards the port side (β positive), the effective angle of attack is increased and therefore stall occurs earlier than for the case where the ship is drifting to the starboard side (blue line).

Figure 12: Reconstruction of pure rudder, drift, yaw and surge speed dependent forces

Figure 14 shows on the left side the effect of the propellers on the rudder dependent global forces. As the ship speed decreases, the propeller load starts to increase. Therefore the flow around the rudder is even more accelerated, increasing the efficiency of the rudder.

The right hand side of Figure 14 shows the reconstruction of static tests with combinations of yaw rate and drift angle with the coefficients from the coupled dynamic tests. It should be noted that none of the computed points in this Figure have been used as input for the determination of the coefficient set.
Figure 13: Reconstruction of rudder angle dependant forces for different drift angles and yaw rates
Figure 14: Reconstruction of rudder angle dependent forces for different forward speeds and combinations of drift angle and yaw rate

2.8 Calm water coefficient set

The complete set of calm water hydrodynamic coefficients for the RoPax ship at 5 kn advance speed is listed in Table 4. Note that the sub index 𝑢 represents 𝛿𝑢. All coefficients have to be multiplied by 0.001.
Table 4: Calm water coefficient set for the RoPax ferry

<table>
<thead>
<tr>
<th></th>
<th>$X_0$</th>
<th>$Y_0$</th>
<th>$K_0$</th>
<th>$N_0$</th>
<th>$X_0'$</th>
<th>$Y_0'$</th>
<th>$K_0'$</th>
<th>$N_0'$</th>
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<tr>
<td>2</td>
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3 Calculation of mean forces and moments in waves

3.1 Introduction

Wave induced second order forces and moments acting on a Ropax ship are required in simulating ship manoeuvres in seaways. The quadratic response amplitude operators of the second order forces and moments were systematically computed for different wavelengths (0.25$L_{pp}$ and 4$L_{pp}$), encounter angles (from 0° to 180° in 15° steps) and ship speeds (0 and 5kts) using a Rankine source method. The numerical results were compared to model test experiments performed by MARINTEK within workpackage 3. General Definitions

3.1.1 Nomenclature

- $L_{pp}$ ship length between perpendiculars [m]
- $B$ ship breadth [m]
- $\lambda_w$ wave length [m]
- $H$ wave height [m]
- $v$ ship velocity [kn]
- $\mu$ wave encountering angle [°]
- $\rho$ water density [$\frac{kg}{m^3}$]
- $g$ gravity acceleration [$\frac{m}{s^2}$]
- $x_A$ surge translation amplitude [m]
- $y_A$ sway translation amplitude [m]
- $z_A$ heave translation amplitude [m]
- $\phi_A$ roll angle amplitude [rad]
- $\theta_A$ pitch angle amplitude [rad]
- $\psi_A$ yaw angle amplitude [rad]
- $\bar{F}_x$ mean x-Force [N]
- $\bar{F}_y$ mean y-Force [N]
- $\bar{F}_z$ mean z-Force [N]
- $\bar{M}_x$ mean x-Moment [Nm]
- $\bar{M}_y$ mean y-Moment [Nm]
- $\bar{M}_z$ mean z-Moment [Nm]
- $\tilde{\omega}$ non dimensional wave length [-]
- $\text{surge}$ non dimensional surge translation [-]
- $\text{sway}$ non dimensional sway translation [-]
- $\text{heave}$ non dimensional heave translation [-]
- $\text{roll}$ non dimensional roll angle [-]
- $\text{pitch}$ non dimensional pitch angle [-]
yaw non dimensional yaw angle [-]
\( C_{F_x} \) non dimensional x-Force [-]
\( C_{F_y} \) non dimensional y-Force [-]
\( C_{F_z} \) non dimensional z-Force [-]
\( C_{M_x} \) non dimensional x-Moment [-]
\( C_{M_y} \) non dimensional y-Moment [-]
\( C_{M_z} \) non dimensional z-Moment [-]

3.1.2 Non dimensional frequency

\[ \tilde{\omega} = \sqrt{\frac{L_{pp}}{\lambda_w}} \]

3.1.3 Motion Amplitudes

\( \text{surge} = \frac{x_A}{H} \)
\( \text{sway} = \frac{y_A}{H} \)
\( \text{heave} = \frac{z_A}{H} \)
\( \text{roll} = \frac{\phi_A}{H} \)
\( \text{pitch} = \frac{\theta_A}{H} \)
\( \text{yaw} = \frac{\psi_A}{H} \)

3.1.4 Non dimensional Forces

\[ C_{F_x} = \frac{\bar{F}_x}{\rho \cdot g \cdot \frac{B^2}{L_{pp}} \cdot \frac{H^2}{2}} \]
\[ C_{F_y} = \frac{\bar{F}_y}{\rho \cdot g \cdot \frac{B^2}{L_{pp}} \cdot \frac{H^2}{2}} \]
\[ C_{F_z} = \frac{\bar{F}_z}{\rho \cdot g \cdot \frac{B^2}{L_{pp}} \cdot \frac{H^2}{2}} \]
3.1.5 System of Coordinates

The used coordinate system is right handed with the x-axis positive towards the bow and the z-axis positive downwards. The origin of the ship bounded coordinate system is located at [Lpp/2, Centreline and T above baseline] as shown in Figure 15. Head waves are denoted by 0°, following waves by 180°, beam waves by 90° from starboard.

\[ C_{M_x} = \frac{\overline{M_x}}{\rho \cdot g \cdot B^2 \cdot \frac{H^2}{2}} \]
\[ C_{M_y} = \frac{\overline{M_y}}{\rho \cdot g \cdot B^2 \cdot \frac{H^2}{2}} \]
\[ C_{M_z} = \frac{\overline{M_z}}{\rho \cdot g \cdot B^2 \cdot \frac{H^2}{2}} \]
3.2 Description of Used Method

3.2.1 Solving Finite Frequency Problem

The hydrodynamic damping $B(\omega_e)$ and hydrodynamic added mass $A(\omega_e)$ for finite wave encounter frequencies, as well as the diffraction force amplitudes $\hat{F}_D(\omega_e)$, are determined using the boundary element method GL Rankine, see (Graefe, 2014). This method takes into account the interaction between the nonlinear stationary flow in calm water (including ship wave, dynamic trim and sinkage) and the periodic flow in waves. Before performing a seakeeping computation, the steady flow problem is solved by a double body simulation. This method improves the solution at low Froude numbers. The steady perturbation potential $\phi^0$ is represented as a superposition of Rankine sources $G(\vec{x}', \vec{\xi}_j) = |\vec{x}' - \vec{\xi}_j|^{-1}$ of strength $4\pi$,

$$
\phi^0 = \phi^0(\vec{x}) = \sum_{j=1}^{n} q_j G(\vec{x}, \vec{\xi}_j).
$$

The body boundary condition and the kinematic boundary condition at the free surface,

$$
(\nabla \phi^0 - \vec{U}) \vec{n} = 0,
$$

are fulfilled using the patch method, (Söding, 1993). Consequently the boundary conditions are fulfilled on the average over each panel and not at collocation points. A nonlinear dynamic boundary condition at the free surface is not required, because of the double body simulation.

Following the so-called Hachmann approach, the total potential $\phi^t$ in waves including forward speed is sought as a superposition of the steady and periodic potentials

$$
\phi^t(\vec{x}, t) = -ux + \phi^0(\vec{x}) + \text{Re}(\hat{\phi}^1 e^{i\omega_et}).
$$

$\vec{x}$ represents a point in the body-fixed coordinate system and $\vec{x}$ the same point in the initial coordinate system. Following this approach, one can derive the dynamic

$$
(\nabla \phi^0 - \vec{U})(\nabla \hat{\phi}^1 + \hat{\alpha} \times \nabla \phi^0 - (\nabla^2 \phi^0)\vec{v}) + i\omega_e(\hat{\phi}^1 - \hat{v}\nabla \phi^0) + g(\hat{\xi}^1 - \hat{v}\nabla \xi^0) = 0
$$

and kinematic

$$
i\omega_e(\hat{\xi}^1 - \hat{v}\nabla \xi^0) + \nabla \xi^0 \nabla \hat{\phi}^1 + (\nabla \phi^0 - \vec{U})\nabla \xi^1 - \frac{\partial}{\partial z} \hat{\phi}^1 + \hat{\alpha}
\left[-\frac{\partial \phi^0}{\partial y}, -\frac{\partial \phi^0}{\partial x}, u \frac{\partial \phi^0}{\partial y}\right]^T + \hat{A} = 0
$$

boundary conditions on the free surface, where

$$
\hat{A} = \frac{-\hat{v}\nabla \phi^0}{|\nabla \phi^0|^2} [\nabla^2 \phi^0 \vec{n}^0 + (\nabla^2 \xi^0) (\nabla \phi^0 - \vec{U})]
$$
\[ \mathbf{n}^o = \left[ \frac{\partial \zeta^o}{\partial x}, \frac{\partial \zeta^o}{\partial y}, -1 \right]^T \]  

(10)

The corresponding body boundary condition is given as

\[ \mathbf{n}(\mathbf{x})(\nabla \hat{\phi}^1 + \mathbf{\hat{a}} \times \mathbf{\hat{U}} - i \omega_e \mathbf{\hat{v}}) = 0 \]  

(11)

Due to the Hachmann approach, error-prone terms involving second derivatives of the steady potential are ‘shifted’ from the body boundary condition to the kinematic boundary condition on the free surface. By experience, this approach leads to more accurate results, in particular at higher Froude numbers. For more details on the frequency domain approach, see (Söding, von Graefe, Shigunov, & el Moctar, 2012).

### 3.2.2 Solving Infinite Frequency Problem

To determine the hydrodynamic added mass and damping at the infinite encounter frequency, a simplified boundary value problem is solved, where the free surface condition is substituted by the high-frequency condition

\[ \hat{\phi}^1 = 0, \]  

(12)

where \( \hat{\phi}^1 \) denotes the complex amplitude of the first-order potential. Usually, this condition is fulfilled by placing image sources with opposite source strengths symmetrically with respect to the plane \( z = 0 \). However, if the condition should be fulfilled on the nonlinear steady free surface \( z = \zeta^o \), it is better to mirror the image source taking into account the nonlinear steady wave (compare also (Söding & Bertram, 2009)). In this way, the boundary condition is not fulfilled exactly, but to a sufficient level of accuracy.

As mentioned before, GL Rankine uses the Hachmann approach to describe the total potential (Hachmann, 1991). Following this approach, the steady flow follows the ship motions. Due to this approach, an additional term due to the periodic motion of the steady perturbation potential is present in the free surface boundary condition, even at the infinite frequency. However, this term is neglected in the simplified free surface condition above, because it cannot be canceled by simple image sources. The resulting approach is exact in the limit of zero forward speed, because no nonlinear ship wave is present, and additional terms due to using Hachmann approach are zero.

At infinite encounter frequency, the wave length approaches zero, as well as the wave length to water depth ratio. Therefore, finite water depth effects including the boundary condition on the bottom are neglected. The body boundary condition is formulated as

\[ \mathbf{n} \nabla \hat{\phi}^1 = \mathbf{n} (-\mathbf{\hat{a}} \times \mathbf{\hat{U}} + i \omega_e \mathbf{\hat{v}}). \]  

(13)

Thus, only the real part of the potential depends on the ship's forward speed; the imaginary part is independent from the forward speed. Additionally to the boundary conditions, \( \hat{\phi}^1 \) has to fulfill Laplace equation (conservation of mass) in the whole fluid domain and a suitable radiation condition. If \( \hat{\phi}^1 \) is described by a superposition of Rankine sources, only the boundary conditions have to be fulfilled by a suitable choice of the source strength. Because the condition on the free surface is fulfilled using image sources, only the boundary
conditions on the body surfaces must be treated numerically using a boundary element approach (see e.g. (Söding, 1993)).

After determining \( \hat{\phi}_1 \), the contributions from each panel to the added mass and damping are determined. In the limit \( \omega_c \to \infty \), the contributions from each panel to the added mass depend only on the imaginary part of \( \hat{\phi}_1 \); therefore, they are independent from the forward speed. However, the contributions from each panel to damping depend on both the real and imaginary parts of the first-order potential \( \hat{\phi}_1 \) and also on the steady base potential including parallel inflow. The contributions from each panel to damping are zero at zero forward speed but non-zero for cases with forward speed. The contributions from each panel to the added mass and damping are summed over the ship hull to obtain the added mass and damping matrices at the infinite frequency. Although the influence of the damping matrix at the infinite frequency is expected to be small it should not be neglected.

3.3 Validation

First, computed and measured first order response amplitude operators of surge, heave, pitch, roll, sway and yaw motions are compared for different encounter angle and ship speeds, see Figure 16 to Figure 21. Further on, the quadratic response operators of added resistance, lateral drift force and yaw moments are computed and compared to experimental data, see Figure 22 to Figure 25. Numerical results agree for a wide range of wave frequencies more or less favorably with model test results, see Figure 16 to Figure 27. Figure 22 and Figure 23 are plotting measured and computed lateral drift forces and yaw moments for head waves and stern waves; as expected the computed values are zero. While, the measured deviate and are not reliable. The deviations between computed and measured second order forces and moments are pronounced for shorter waves. This may due to nonlinearities which are not captured properly by the Rankine source method.
3.3.1 Validation charts of ship motions

Figure 16: Surge motion at $v=0$ kn and $\mu=0^\circ$

Figure 17: Heave motion at $v=0$ kn and $\mu=90^\circ$
Figure 18: pitch motion at v=0kn and µ=60°

Figure 19: roll motion at v=0kn and µ=120°
Figure 20: sway motion at $v=5$ kn and $\mu=30^\circ$

Figure 21: yaw motion at $v=5$ kn and $\mu=120^\circ$
3.3.2 Validation charts of forces

Figure 22: sway force at \( v=0 \) kn and \( \mu=0^\circ \)

Figure 23: yaw Moment at \( v=0 \) kn and \( \mu=0^\circ \)
Figure 24: sway force at $v=0$ kn and $\mu=90^\circ$

Figure 25: yaw moment at $v=0$ kn and $\mu=90^\circ$
Figure 26: surge force at $v=5\text{kn}$ and $\mu=60^\circ$

Figure 27: sway force at $v=5\text{kn}$ and $\mu=30^\circ$
4 Manoeuvring prediction in waves

4.1 Modelling of mean wave forces

The mean wave forces needed for the manoeuvring prediction were provided by the UDE. The used procedure to obtain these forces has been explained in section 3 of this report. The mean wave forces were then used to determine the mean wave force coefficients as described in (Uharek & Cura Hochbaum, 2015). After all the wave dependent coefficients are determined, the forces can be calculated using the following formulas:

\[
F' = \frac{a_0}{2} + \sum_{n=1}^{6} \left[ a_n(\lambda') \cos(n\alpha) + b_n(\lambda') \sin(n\alpha) \right]
\]

with
\[
a_n(\lambda') = \sum_{i=0}^{3} a_{ni}(\lambda')^i \quad \text{and} \quad b_n(\lambda') = \sum_{i=0}^{3} b_{ni}(\lambda')^i
\]

(14)

Figure 28 shows the input data (symbols) used for determining the coefficients \(a_{ni}\) and \(b_{ni}\) (shown in equation (14)) and the traces obtained with the mathematical model for six different wave lengths. As can be seen the quality of the reconstruction is very satisfactory.

![Graph showing input data and traces for mean wave forces](image)

**Figure 28:** Reconstruction of mean wave forces with mathematical model

4.2 Simulation of rudder manoeuvres in regular waves

After all necessary coefficients are determined, the simulation of any desired rudder manoeuvre can be simulated by explicit time integration of the rigid body motion equations, see equation (1).

<table>
<thead>
<tr>
<th>Approach situation</th>
<th>time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head (t_{90})</td>
<td>128 s</td>
</tr>
<tr>
<td>Head (t_{180})</td>
<td>243 s</td>
</tr>
<tr>
<td>Beam (t_{90})</td>
<td>133 s</td>
</tr>
<tr>
<td>Beam (t_{180})</td>
<td>250 s</td>
</tr>
<tr>
<td>following (t_{90})</td>
<td>130 s</td>
</tr>
<tr>
<td>following (t_{180})</td>
<td>252 s</td>
</tr>
</tbody>
</table>

Table 5: Times during turning circle tests
Figure 29 shows on the left side the trajectory of a turning circle test in calm water with the rudder deflected 35° to starboard. The right side shows the same turning circle in waves of length $\lambda' = \frac{\lambda}{L_P} = 0.75$ and amplitude $\zeta' = \frac{\zeta}{L_P} = 0.003$. The initial wave encounter angle is $\alpha = 0^\circ$ (head seas) at $t = 0$. As can be seen the circles do not only get shifted in the direction of the incoming waves, but also show a significant twist referred to this direction. This twist does not depend on the starting condition, as shown in Figure 30. The corresponding times to reach a heading change of 90° and 180° are shown in Table 5.

The left side of Figure 31 shows the traces of a turning circle test in a wave with length $\lambda' = 0.75$ for several wave amplitudes. As can be seen, there is a very strong influence on the shift of the turning circles, whereas the twist is hardly influenced at all. The trajectories on the right hand side are the results of simulations for different wave lengths, while the wave steepness is kept constant ($H/L = 0.02$). It is clearly visible, that the wave length has a strong influence on the twist.

![Figure 29: Turning circle test trajectories and time traces in calm water (left) and in waves with $\lambda'=0.75, \zeta'=0.003$ (right)](image-url)
Figure 30: Turning circle test trajectories and time traces in waves with $\lambda'=0.75$, $\zeta'=0.003$, for two additional staring conditions.

Figure 31: Turning circle test trajectory for varying wave amplitudes (left) and wave lengths (right).
5 Concluding Remarks

An extensive calculation of the manoeuvring behaviour of an existing sea going RoPax ship in calm water and in regular waves has been performed. For this purpose, TUB determined a calm water manoeuvring coefficient set for the RoPax at 5 kn advance speed using a RANS method and UDE calculated the mean forces and moments on the hull due to waves of several lengths and encounter angles. Calculations of mean wave forces are compared with experimental data obtained in WP3.

Finally, TUB inserted the mean wave forces in the mathematical model and simulated turning circle tests in calm water and regular waves of different lengths at several initial wave encounter angles. Any desired rudder manoeuvre can now be quickly predicted using a mathematical model based on hydrodynamic coefficients for approximating the forces and moment acting on the ship.
6 References


