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Xujian Lyu  
Haiwen Tu  
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ON RESISTANCE REDUCTION OF A HULL BY TRIM OPTIMIZATION

UDC 629.5.012.442:629.5.015.2
Original scientific paper

Abstract

The paper aims at conducting trim optimization for a hull to reveal the influence of trim on wave resistance by a potential-based panel method coupled with a response surface method. First, a numerical program for solving the linear free-surface flow problem of a hull moving with a uniform speed in calm water is built by the panel method. The S60 hull model is used to validate the numerical procedure. Next, calculation for hull is performed with two different trims at a wide range of Froude number; resistance test is conducted to validate the numerical method in demonstrating the influence of trim on wave resistance. Finally, a response surface of wave resistance is constructed with respect to variations of trim and Froude number, using the database of wave resistance calculated by the surface method. In this way, a framework is developed to perform trim optimization. The optimum trim point for the present hull shows a significant improvement in both wave resistance and total resistance, compared with that of even keel and the worst trim point. The optimization framework is proved to be effective in energy saving due to resistance reduction.

Key words: Trim optimization; Wave resistance; Panel method; Response surface

1. Introduction

In the face of global climate change, the importance of addressing greenhouse gas emissions attracts worldwide attention gradually. For shipping industry, the problems of greenhouse gas emission and fuel efficiency in operation are becoming more and more important due to the requirement of Environmental Ship Index (ESI) from International Maritime Organization (IMO) [1, 2]. When it comes to green shipping, one of the main concerns about ship hydrodynamics is resistance reduction. As it is well known, employing excellent hull forms or optimizing hull lines with low resistance in ship design stage is an effective measure for this goal [3-6]. However, it usually requires modifications to hull lines, which may cost lots of money and time for ships in operation. Another option to optimize the resistance performance without modifications to hull is increasing the operational efficiency of ships [7]. Traditionally, ships have been optimized for minimum resistance at the designing trim condition, which is the even keel trim condition or trimming somewhat aft for most
ships. Thus, one might think that the optimum trim condition, when wave-making resistance is considered, should not always be the designing one. With typical wave-making resistance taken into account, the paper is therefore to explore the optimum trim condition for a certain hull at given Froude numbers and displacement.

As one of the most widely used tools for wave-making calculation in hydrodynamics, the potential-based panel method deals with the Laplace boundary-value problem with solving integral equation only at boundaries of the flow field [8-11]. Therefore, it is usually more effective in wave resistance calculation than other tools such as viscous flow based Computational Fluid Dynamics (CFD). Ghassemi and Kohansal studied the generation of wave due to moving hydrofoil in steady streams close to free surface, employing a potential based panel method [12]. A modified Rankine panel method was used to improve the prediction of wave pattern and wave-making resistance for ships of full form by Peng et al. [13], and an alternative method was first developed to satisfy the free surface boundary conditions on arbitrary body-fitted lines instead of streamlines. Lv et al. employed a potential-based boundary element method to determine a better trim condition with lower wave resistance of the hull. However, without an optimization model, the trim had not been proved to be optimum for the hull. Iakovatos et al. [14] experimentally studied the impact of trim angles on resistance. In addition, Sherbaz and Duan assessed the influence of trim on ship resistance with CFD as well [15]. More recently, Sun et al. [16] developed a response surface based program with database from experiment and CFD to perform trim optimization. The optimized results had been validated by real ship test as well. However, experiment and CFD require a lot of money or computer resource input to assess the influence of trim on ship resistance.

In this study, an efficient and economical framework is developed, to get resistance reduction based on trim optimization for a certain hull. Other than using a double model based boundary element method for wave resistance calculation [17], a direct potential-based panel method is applied to build a database for wave resistance. The Series 60 hull is utilized as a verification example and model test for a certain hull is conducted to verify the influence of trim on wave resistance as well. Finally, the response surface method, with surface fitting by Lagrange interpolation function instead of polynomial function, is applied to perform trim optimization, and the influence of trim on wave resistance and total resistance is evaluated.

2. Statement of the Boundary Value Problem

Let us consider a ship moving with a constant speed $U$ in calm water. A ship-fixed Cartesian coordinate system $O$-xyz is defined with origin $O$ located at the mid-ship on the calm water surface and the $x$-axis positive towards the ship stern. The $y$-axis positive extends to starboards and the $z$-axis is vertically towards (Fig. 1).

It is assumed that the fluid is incompressible and inviscid, and the flow is irrotational. The total velocity potential $\Phi$ can be expressed as a sum of the oncoming velocity potential $\phi_\infty$ and the perturbation potential $\phi$, namely

$$\Phi = \phi_\infty + \phi = \bar{U} \cdot \bar{x} + \phi$$  \hspace{1cm} (1)
Conservation of the mass applied to the potential flow gives the Laplace equation as a governing equation

$$\nabla^2 \phi = 0$$  (2)

Then motion of the flow can be uniquely defined by imposing the boundary condition on the boundary surface as follows:

1) Hull boundary condition

$$\frac{\partial \phi}{\partial n} = -\vec{U} \cdot \vec{n} \text{ on } S_H$$  (3)

where \(\vec{n}\) is the unit vector normal to the boundary, defined positive when pointing into the fluid region.

2) Linearized free surface boundary condition can be simplified as

$$\frac{\partial \phi}{\partial z} + \frac{U^2}{g} \frac{\partial^2 \phi}{\partial x^2} = 0 \text{ on } S_F$$  (4)

where \(g\) is gravity acceleration.

By applying Green’s theorem, the equation about the perturbation potential can be expressed as

$$\int_{S_H} \int_{S_F} \left( \phi \frac{\partial G}{\partial n} - G \frac{\partial \phi}{\partial n} \right) dS = 0$$  (5)

For the problem of flow around the hull. Equation (4) can be written as

$$4\pi E \phi(p) = \int_{S_H} \phi(q) \frac{\partial G}{\partial n_q} dS - \int_{S_H} G \frac{\partial \phi(q)}{\partial n_q} dS - \int_{S_F} \phi(q) \frac{\partial G}{\partial n_q} dS$$  (6)

where

$$E = \begin{cases} 
\frac{1}{2} & \text{on } S_H \\
1 & \text{on } S_F 
\end{cases}$$  (7)

Green’s function \(G\) is given by

$$G = \frac{1}{R(p,q)} + \frac{1}{R'(p,q)}$$  (8)

where \(R = \sqrt{(x-\xi)^2 + (y-\eta)^2 + (z-\zeta)^2}\) is the distance between field point and the source point, and \(R' = \sqrt{(x-\xi)^2 + (y-\eta)^2 + (z+\zeta)^2}\) is the distance between field point and the image of source point.

The hull surface \(S_H\) and the free surface \(S_F\) are discretized into a number of \(N_H\) and \(N_F\) panels respectively to get an approximate solution of the problem [18]. The perturbation potential \(\phi\) and its normal derivative \(\frac{\partial \phi}{\partial n}\) are assumed constant at each panel and equation about \(\phi\) and \(\frac{\partial \phi}{\partial n}\) can be obtained as
where \( I_{HD} \), \( I_{HS} \) and \( I_{FS} \) are influence coefficients proposed by Morino et al. [19] and the details can be referred to the literature [20].

As the free surface should be satisfied on the true surface, an iterative method is used to obtain the solution of equation (9). To get acceptable results, the iteration number is about 4~7 in this paper.

The wave resistance coefficient

\[
C_w = \frac{R_w}{(1/2)\rho U^2 S_w} = \frac{1}{S_w} \sum_{i=1}^{n_{p/2}} C_p(i)n_{ni}\Delta S_{Wi}
\]  

(10)

where \( S_w \) is the wetted surface area of the hull in calm water, \( \rho \) is the water density, \( \Delta S_{Wi} \) is the area of the panel element, \( n_{ni} \) is \( x \)-component of the unit vector normal to the surface panel element, and \( C_p \) is the pressure coefficient determined by the Bernoulli’s equation.

The wave height can be calculated with the linearized kinematic free surface condition

\[
\zeta(x, y) = -\frac{U}{g} \frac{\partial \phi}{\partial x} \quad \text{on} \quad z = 0
\]  

(11)

3. Sample calculation and validation

The Series 60 hull model (\( C_b = 0.6 \)) is used as an example to check the reliability for the numerical procedure of present panel method [21]. The free surface is discretized (-1 ≤ \( x / L_{pp} \) ≤ 2, -1 ≤ \( y / L_{pp} \) ≤ 1) by 100 × 40 quadrilateral panels and the hull by 1346 panels, 5 × 6 for the stern and 28 × 47 for other parts of the hull respectively. Calculation is conducted for the S60 hull with fixed trim and sinkage and computed wave resistance coefficient is compared with that obtained from the literature by Tarafder & Suzuki [18], as shown in Fig. 2. Though there are some differences between the results of wave resistance from present study and that from literature at some Froude numbers, the overall agreement of the results is quite acceptable. In addition, numerical results on wave profiles along the body surface at \( Fn = 0.22 \) and \( Fn = 0.34 \) are shown in Figs 3 and 4 respectively. The variation trend of wave profiles from present study and literature matches well with each other. The present results show higher accuracy than that from literature when compared with experimental results, as shown in the figures. The comparisons show that the present numerical method is able to catch typical flow characters around the hull, and thus it can be concluded that the numerical procedure is feasible for wave resistance and wave profile prediction. Subsequently, the numerical method is applied for the calculation of a HUST (Huazhong University of Science & Technology) hull [17], of which the main parameters are summarized in table 1. Both the hull surface and the free surface domain (-1 ≤ \( x / L_{pp} \) ≤ 2, -1 ≤ \( y / L_{pp} \) ≤ 1) are discretized by quadrilateral panels, as shown in Figs 5 and 6. The displacement of the hull is kept constant during the calculation.

To provide further validation of the numerical procedure, resistance tests of the HUST hull were performed at the towing tank laboratory of the School of Naval Architecture and Ocean Engineering, Huazhong University of Science and Technology, Wuhan, P. R. China. The principle dimension of the towing tank is 175 × 6 × 4m [16]. According to the ITTC guidelines, the wave resistance coefficient of HUST hull from model test can be written as

\[
C_r = (1+k) \cdot C_f + C_w
\]  

(12)
where $C_t$ is the total resistance coefficient got from model test, $C_f$ is the frictional resistance coefficient determined by the ITTC ’57 correlation line and $k$ is the form factor determined by Prohaska method adopted by ITTC in 1978.

The trim condition of present hull is defined by the aft draft $T_a$ and the fore draft $T_f$, that is

$$Trim = T_f - T_a$$  \hspace{1cm} (13)

where $Trim > 0$ indicates trim by bow and $Trim < 0$ indicates trim by stern, as shown in Fig. 7.
Table 1 Typical panel arrangement for the hull

<table>
<thead>
<tr>
<th>Descriptions</th>
<th>Parameters</th>
<th>Value</th>
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<tr>
<td>Volume of displacement</td>
<td>$\nabla$ (m$^3$)</td>
<td>0.7136</td>
</tr>
<tr>
<td>Wetted surface area</td>
<td>$S_w$ (m$^2$)</td>
<td>4.4961</td>
</tr>
<tr>
<td>Length between perpendiculars</td>
<td>$L_{pp}$ (m)</td>
<td>4.4225</td>
</tr>
<tr>
<td>Beam</td>
<td>$B$ (m)</td>
<td>0.6950</td>
</tr>
<tr>
<td>Draft</td>
<td>$D$ (m)</td>
<td>0.2687</td>
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<tr>
<td>Block coefficient</td>
<td>$C_B$</td>
<td>0.7443</td>
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Fig. 5 Typical panel arrangement for the HUST hull

Fig. 6 Typical panel arrangement for the free surface

Fig. 7 Sketch of trim conditions for the hull
Comparisons of wave resistance for the hull at two different trim conditions between numerical results and experiment results are shown in Fig. 8. The variation trend of calculated and experimental wave resistance with respect to trim conditions agrees quite well at a wide range of Froude number (i.e. \(Fn > 0.17\)), even at the inflection point (i.e. \(Fn \approx 0.192, Fn \approx 0.207\)) where the trim variations leads to few changes in wave resistance. During the experiment, the HUST hull is tested with free sinkage and it’s found that sinkage of the hull is more significant at \(Fn=0.203\) than that at other Froude numbers. Since the wave resistance calculation was conducted with fixed sinkage for the HUST hull, one possible reason for the peak difference between wave resistance curves from calculation and experiment (\(Fn = 0.203\) in Fig. 8) is that, the fixed sinkage during calculation covers up the influence of significant free sinkage to wave resistance. Typical wave profiles for the hull at \(Fn = 0.203\) and 0.217 are shown in Figs 9 and 10 respectively. The variation trend of wave heights over trim from calculation shows a good agreement with that from experiment as well. In addition, calculated wave patterns provide further verification by detailing wave height changes with different trim conditions around the hull, as depicted in Figs 11 and 12. Thus, it is can be concluded that the present numerical procedure is able to properly demonstrate the wave resistance changes caused by trim adjustment for the HUST hull at \(Fn > 0.17\).

4. Trim optimization

To explore the relationship between a few explanatory variables (i.e. trim condition, Froude number) and a response variable (i.e. wave resistance), the response surface methodology is introduced to conduct trim optimization [16, 22]. Instead of using a second-degree polynomial model to approximate the surface, the Lagrange interpolation function is applied to make sure that the effective points from calculation is included in the response surface.

![Fig. 8 Wave resistance of the HUST hull](image)
Fig. 9 Wave profiles along the HUST hull at Fn=0.203

Fig. 10 Wave profiles along the HUST hull at Fn=0.217

Fig. 11 Wave patterns around the HUST hull at Fn=0.203

Fig. 12 Wave patterns around the HUST hull at Fn=0.217
To create a database, the wave resistance of the HUST hull at different Froude numbers and trims is calculated with above numerical method and summarized in Table 2. The response surface of wave resistance with respect to Froude numbers and trims is constructed based on the wave resistance database, as depicted in Fig. 1.

The trim optimization of the HUST hull is conducted by using the response surface, in which a response of wave resistance is influenced by trim and Froude number. In present trim optimization for the HUST hull with the displacement at a certain initial draft (i.e. \(D = 268.7\) mm), the design variables are trim and ship speed based Froude number. The objective is to find the optimal value of trim to minimize the wave resistance of the hull at a given Froude number. The optimization model is given as follows:

\[
C_w = f(Trim, Fn, Dis) 
\]

\[
\min f(Dis, Fn, Trim) 
\begin{cases}
Dis = Dis0 \\
Fn = Fn0 \\
Trim1 < Trim < Trim2
\end{cases} 
\]

where \(f(Trim, Fn, Diss)\) is the response surface, \(Dis0\) is present volume displacement of the hull, \(Fn0\) is the Froude number based on given speed, and \(Trim1\) and \(Trim2\) define the variation range of the trim condition. The overall optimization procedure is demonstrated in Fig. 14. There exists an optimal trim curve (may be made up by several curve portions) on the response surface, and the corresponding wave resistance is the minimum. The intersection point of the optimal trim curve and the Froude number isoline represents the optimal point of the minimum wave resistance at present ship velocity, as shown in Fig. 13 and detailed in Figs 15-17.

To evaluate the trim optimization results, the wave resistance reduction ratio is defined as

\[
\alpha_w = \frac{(C_w^{\text{Optimum}} - C_w)}{C_w} \times 100\% 
\]

and the total resistance reduction ratio is defined as

\[
\alpha_t = \frac{(C_t^{\text{Optimum}} - C_t)}{C_t} \times 100\% 
\]

where \(C_w^{\text{Optimum}}\) and \(C_t^{\text{Optimum}}\) are wave resistance coefficient and total resistance coefficient at the optimum trim condition, respectively. \(C_w\) and \(C_t\) are wave resistance coefficient and total resistance coefficient at the reference trim condition point, respectively. The optimized trim condition is able to provide 9.7%-11.5% of wave resistance reduction and 3.5%-3.8% of total resistance reduction, compared with the even keel condition for the HUST hull. What’s more,

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<th>Tirm (mm)</th>
<th>0.149</th>
<th>0.163</th>
<th>0.176</th>
<th>0.190</th>
<th>0.203</th>
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<td>2.265</td>
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<td>2.981</td>
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Xujian Lyu, Haiwen Tu, De Xie, Jianglong Sun

On resistance reduction of a hull by trim optimization

<table>
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<th>Trim (m°)</th>
<th>0.114</th>
<th>0.246</th>
<th>0.298</th>
<th>0.631</th>
<th>1.447</th>
<th>1.406</th>
<th>1.951</th>
<th>2.848</th>
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<td>2.810</td>
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<td>1.582</td>
<td>1.235</td>
<td>1.853</td>
<td>2.714</td>
<td>2.740</td>
</tr>
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</table>

Fig. 13 Response surface of wave resistance

**Optimization problem setup**
Design variables, Objective functions, Constraints

**Response surface generation**
A function by fitting and interpolation

**Conduction of optimization**
Using the Response surface

**Optimum point**

Fig. 14 Optimization procedure based on response surface
the reduction ratio can be as high as 20.2%-35.8% of wave resistance and 6.8%-11.3% of total resistance while the comparison is taken between the optimized trim condition and the worst trim condition, as shown in Table 3. Obviously, the contribution of trim optimization to resistance reduction for the HUST hull is significant.

Table 3  Resistance reduction ratio at optimum trim condition point

<table>
<thead>
<tr>
<th>Fn</th>
<th>$C_w^{\text{Optimum}} \times 10^3$</th>
<th>$\alpha_w$ related to Even-keel point</th>
<th>$\alpha_w$ related to Worst point</th>
<th>$\alpha_t$ related to Even-keel point</th>
<th>$\alpha_t$ related to Worst point</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.203</td>
<td>1.281</td>
<td>11.5%</td>
<td>20.2%</td>
<td>3.5%</td>
<td>6.8%</td>
</tr>
<tr>
<td>0.217</td>
<td>1.038</td>
<td>26.2%</td>
<td>35.8%</td>
<td>7.2%</td>
<td>11.3%</td>
</tr>
<tr>
<td>0.240</td>
<td>2.397</td>
<td>9.7%</td>
<td>23.7%</td>
<td>3.8%</td>
<td>11.8%</td>
</tr>
</tbody>
</table>
5. Conclusion

Traditionally, hull lines are designed or optimized for minimum resistance at the designing trim condition, which is the even keel trim condition or trimming somewhat aft for most ships. To achieve reduction of resistance in a different way, the paper in which typical wave making of a hull is taken into account therefore aims at exploring a trim for the HUST hull, with minimum wave resistance at given Froude numbers and displacement. After being validated by results from calculation of the S60 hull and resistance test of the HUST hull, a potential-based panel method coupled with a response surface method is used to develop a framework for trim optimization. The optimum trim condition for the hull shows a significant improvement in both wave resistance and total resistance. The optimization framework is effective in energy saving due to resistance reduction, not only for the present HUST hull, but also for general hull models.

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On resistance reduction of a hull by trim optimization

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COMPARISON STUDY OF EXPERIMENTS AND PREDICTIONS OF WAVE KINEMATICS FOR ROGUE WAVE

UDC 629.5.016.8: 629.5.017.2: 629.5.018.1

Original scientific paper

Summary

To investigate the wave kinematics under the rogue wave crest, a series of experiments were performed in 2-D wave tank with the application of PIV technique to measure the velocities under the free surface. Three different prediction methods of linear extrapolation, Wheeler stretching, and modified stretching were applied to estimate water wave kinematics and compared with PIV experimental results under the highest wave crest of irregular wave trains satisfying with rogue wave criteria. Also, the cut-off frequency dependence for three prediction methods was investigated with varying spectral peak frequencies to estimate wave kinematics including velocities and accelerations in horizontal and vertical directions. It was suggested that the cut-off frequency for the reasonable prediction of the wave kinematics under the rogue wave crest could be chosen three times of spectral peak wave frequency for the linear extrapolation and higher frequency than four times of spectral peak wave frequency for Wheeler stretching and modified stretching method.

Key words: rogue wave; wave kinematics; cut-off frequency; linear extrapolation; Wheeler stretching; modified stretching

1. Introduction

For the last several decades, the damages of many ships and offshore structures have been caused of rogue waves which had an exceptional height and abnormal shape. Two large Norwegian bulk ships M/S “Norse Variant” and M/S “Anita” disappeared at the same time at the same location [1]. According to the conclusion of the Court of Inquiry, a very large wave suddenly broke several hatch covers on deck, and the ships were filled with water and sank before any emergency call was given. The Queen Elizabeth II was struck by a rogue wave of 29 m wave height in 1995 in the North Atlantic [2]. The Caledonian Star, sailing in the South Atlantic in 2001, was hit by a rogue wave estimated to be 30 m, approximately, of wave height. The Explorer, on a “semester-at-sea” sailing in the North Pacific, was damaged in January 2005 when the ship, carrying nearly 1000 people including almost 700 college students, was struck by a wave estimated to be 17 m in height. The wall of water smashed into the bridge of the 180 m long ship. These well-built cruise ships suffered a little damage and had a few injuries from
the attack of rogue waves [3]. The Norwegian Dawn, a 3 years old 294 m long cruise ship carrying more than 2200 passengers and heading back to New York from the Bahamas, was pounded by a rogue wave during a storm in April 2005 off the South Carolina coast. The wave reached the 10th deck of the towering ship, and 62 cabins were flooded and some public areas were damaged [4]. Offshore platform “Draupner oil rig” in the North Sea have also been meet with the rogue wave of 26 m in height measured by an onboard laser, while the surrounding waves reached 12 m on January 1, 1995 [5]. MaxWave, a German scientific group, examined 30,000 worldwide satellite photos taken by the European Space Agency (ESA). According to MaxWave, 10 rogue waves, each larger than 25 m in height, were identified around the globe within the three-week research period in 2001 [6]. These rogue waves occur more frequently where there are strong currents, such as the Gulf Stream off the eastern coast of North America.

After Draper [7] suggested using the term “freak waves” and developed a theory for application to a real ocean wave spectrum, the terms for waves having exceptional high wave height varied as extreme waves, giant waves, mountain waves, rogue waves, etc. Recently, many researchers have used “rogue waves” as academic terms. Researchers [8, 9] defined rogue wave as a wave height exceeding twice of the significant wave height of surrounding waves. Olaignon and Iseghem [10] defined a rogue wave when the wave height was larger than two times of significant wave height and the ratio of crest height to significant wave height was larger than 1.25.

The factors of wind-driven waves, currents, ocean bottom topography or inclement weather can play a role in rogue wave development. With the assumption of the linear wave theory, rogue waves can be considered as the superposition of a number of independent monochromatic waves with different frequencies and directions. A rogue wave may appear in the process of geometrical focusing, dispersion enhancement, and wave-current interactions. Kharif et al. [11], and Smith and Swan [12] explained the rogue wave phenomenon with an aspect of dispersion enhancement in a random sea. The combination of the geometrical focusing and dispersion enhancement mechanism to form an extreme wave has also been examined by Wu and Nepf [13]. Peregrine [14] and Smith [15] investigated the rogue waves with the mechanism of wave-current interactions for the Agulhas current in the South Africa where strong currents are often observed. Levrenov [16] described the rogue waves in Agulhas with an aspect on the concentration of wave-energy density.

Three processes mentioned above are investigated analytically and numerically in the framework of weakly nonlinear models like the nonlinear Schrödinger equation, as well as researched in the laboratory. Tulin and Waseda [17] observed highly nonlinear waves including breaking waves in deep water in a large wave tank. Trulsen and Dysthe [18, 19] focused on the numerical simulation of rogue waves with 3rd and 4th order nonlinear Schrödinger equations and provided breather type solutions using the equations suggested by Henderson et al. [20]. Osborne et al. [21, 22] studied the dynamic behaviour of rogue waves with the numerical model and extended to the directional, random oceanic sea states. Lu et al. [23] adopted the wavelet analysis method to analyse the rogue waves to the time-frequency energy distribution. Zhang and Soares [24] investigated the ship responses to rogue waves simulated by the nonlinear Schrödinger equation.

Nonlinear wave-wave interaction has been addressed in association with the rogue wave formation [25, 26]. Grue [27], and Clamond and Grue [28] performed a fully nonlinear numerical simulation of a long time evolution using a two-dimensional localized long wave packet. In a 2-D wave tank, Wu and Yao [29] investigated rogue wave kinematics using a combined mechanism of dispersion enhancement and wave-current interaction. Zou and Kim [30] generated a strongly asymmetric wave in the 2-D wave tank by the time series distortion to the highest wave in irregular wave train. Kim and Kim [31] simulated the Draupner rogue
wave in the 2-D wave tank with measuring the horizontal particle velocity and the horizontal force on a vertical truncated cylinder fixed in the wave. In the towing tank, Bennett et al. [32] provided a technique for experimental modelling of rogue waves using NewWave method.

In this study, PIV (particle image velocimetry) technique was employed to measure a water particle velocity profile under the wave crest, and measured water velocity components was verified by comparing with theoretical results of regular waves. The rogue wave was generated with the distortion method applying to the highest wave in irregular wave train of JONSWAP spectrum in 2-D wave tank. The water velocity profiles under rogue wave crests were measured by PIV technique, and their local and convective accelerations were calculated from consecutive and individual velocity profiles, respectively. Water kinematics of rogue waves were compared with results with three prediction methods of the linear extrapolation, Wheeler stretching [33], and the modified stretching [34], and their cut-off frequency dependences were investigated on the wave kinematics prediction under wave crests of rogue waves.

2. Experimental setup and technique

A series of experiments was conducted to measure the water particle velocities of regular and irregular water waves in 2-D wave tank which is 35 m long, 0.91 m wide, and 1.22 m deep glass-walled flume as shown in Fig. 1. It was equipped with a permeable wave absorbing 1:5.5 sloped beach having 5% reflection. Wave maker had a dry-back, hinged flap type and was driven by a synchronous servo-motor controlled by a computer and hydrostatically balanced using an automatic near constant force and a pneumatic control system. The set-up is shown schematically in Fig. 1, where x is the horizontal coordinate positive in the direction of wave propagation from the wave maker and z is positive upward. Free surface elevations of regular and irregular waves were measured at 800 cm from the wave maker, respectively, and the water depth in the wave tank was maintained at 90 cm.

Regular waves having the wave period of 0.9 s were generated with three different wave heights (Table 1), and four irregular wave trains using JONSWAP spectrum with a peak enhancement factor of 6.5 were generated in 2-D wave tank (Table 2). The distortion method [33] was applied at the highest wave crest in order to increase the maximum height of irregular wave trains. The maximum wave heights of in irregular waves of cases PH3 and PH4 were satisfied with the rogue wave criteria listed in Table 2.

<p>| Table 1 Wave parameters of regular waves (H_C: Wave crest height, H_t: Wave trough height) |
|-----------------------------------------------|---------------|---------------|---------|---------|---------|</p>
<table>
<thead>
<tr>
<th>Case</th>
<th>H (cm)</th>
<th>H_C (cm)</th>
<th>H_t (cm)</th>
<th>T (s)</th>
<th>H/L</th>
<th>k_a</th>
</tr>
</thead>
<tbody>
<tr>
<td>PR1</td>
<td>4.17</td>
<td>2.05</td>
<td>-2.12</td>
<td>0.90</td>
<td>0.033</td>
<td>0.104</td>
</tr>
<tr>
<td>PR2</td>
<td>8.13</td>
<td>4.52</td>
<td>-3.61</td>
<td>0.90</td>
<td>0.064</td>
<td>0.202</td>
</tr>
<tr>
<td>PR3</td>
<td>12.29</td>
<td>7.27</td>
<td>-5.02</td>
<td>0.90</td>
<td>0.097</td>
<td>0.305</td>
</tr>
</tbody>
</table>

<p>| Table 2 Wave parameters of irregular waves |
|---------------------------------------------|---------------|---------------|---------|---------|</p>
<table>
<thead>
<tr>
<th>Case</th>
<th>H_0 (cm)</th>
<th>T_0 (s)</th>
<th>H_{max} (cm)</th>
<th>H_{max} / H_0</th>
<th>H_C / H_0</th>
</tr>
</thead>
<tbody>
<tr>
<td>PH1</td>
<td>6.63</td>
<td>1.25</td>
<td>14.11</td>
<td>2.13</td>
<td>1.20</td>
</tr>
<tr>
<td>PH2</td>
<td>7.00</td>
<td>1.27</td>
<td>15.11</td>
<td>2.16</td>
<td>1.22</td>
</tr>
<tr>
<td>PH3</td>
<td>7.43</td>
<td>1.19</td>
<td>16.09</td>
<td>2.17</td>
<td>1.25</td>
</tr>
<tr>
<td>PH4</td>
<td>7.78</td>
<td>1.18</td>
<td>16.36</td>
<td>2.11</td>
<td>1.29</td>
</tr>
</tbody>
</table>
The PIV technique was employed to obtain the velocity field of water waves. The PIV system and the wavemaker were synchronized by computer A housing a data acquisition board (Fig. 1). Computer B saved wave elevation from the wave gage, and the timing of laser pulses and CCD camera were controlled by computer C housing the Programmable-Timing-Unit-Board. The digital CCD camera mounted with a 105 mm f/1.8 micro focal lens set at f/2.8 ~ 4.0 was used to obtain PIV images. It had 1280 x 1024 pixels, 6.7 μm x 6.7 μm pixel size, 12 bit dynamic range, and 8 Hz framing rate. A pair of images was obtained by the double-frame/single-pulsed method shown in Fig. 2. The time difference (dt) between the 1st frame and the 2nd frame was adjusted to be about 3~5 ms, which was determined by the maximum displacement to be less than a third of the width of the interrogation window size. The fields of view (FOV) sizes were 127 x 159 mm² and 172 x 215 mm² for regular and irregular wave conditions, respectively, corresponding to spatial resolutions of 2.01 mm and 2.72 mm between every velocity vectors in the interrogation area of 32 x 32 pixels with 50% overlap. The adaptive multi-pass algorithm was applied to reduce faulty vectors. Because this method has shifted an interrogation area to the location where particles moved, the stronger cross-correlation can be taken. Once the velocity vectors have been calculated in the interrogation area with 50% overlap, spurious false vectors were eliminated by the median filter [36]. The left-over empty spaces were filled-up with interpolated vectors and smoothed by a simple 3 x 3 smoothing filter to reduce noise.
\[ H_{rms} = \sqrt{\frac{\sum_{i=1}^{N} (H_i - \bar{H})^2}{N}} \]  

(1)

where \( H_i \) represents measured wave height for each wave period, \( \bar{H} \), the average wave height of twelve selected regular wave trains, and \( N \) the number of selected regular waves to be averaged. The error rate of selected regular waves was calculated by Equation (2).

\[ ER_{wave}(\%) = \left( \frac{H_{rms}}{\bar{H}} \right) \times 100 \]  

(2)

The total error of the wave generation and the wave elevation measurement system was estimated below 1% for all regular wave conditions in Table 3.

![Fig. 3 Time series of regular waves for the case PR2, T= 0.9 s, H= 8.13 cm](image)

<table>
<thead>
<tr>
<th>Case</th>
<th>PR1</th>
<th>PR2</th>
<th>PR3</th>
</tr>
</thead>
<tbody>
<tr>
<td>( H_{rms} ) (cm)</td>
<td>0.01</td>
<td>0.02</td>
<td>0.07</td>
</tr>
<tr>
<td>( ER_{wave} ) (%)</td>
<td>0.21</td>
<td>0.23</td>
<td>0.65</td>
</tr>
</tbody>
</table>

To simulate the rogue wave in the 2-D wave tank, a series of irregular wave trains have been generated with JONSWAP spectrum (peak enhancement factor, \( \gamma = 6.5 \)). Two important characteristics of the rogue wave were considered for generating the rogue wave in the 2-D wave tank instead of in the ocean. The first characteristic was the wave height to be larger than twice of the significant wave height, and the second characteristics was the ratio of crest height \( H_c \) to significant wave height \( H_s \) to be greater than 1.25 [10]. The second characteristic of the rogue wave was presenting a strongly asymmetric wave profile which was one of typical patterns of highly nonlinear wave.
The distortion method [35] was applied to generate the rogue wave in 2-D wave tank, which has three steps of amplitude distortion, time distortion, and crest distortion techniques to increase the highest wave height and make a stronger asymmetry wave profile in the irregular wave. Amplitude distortion was intended to increase the crest height and reduce the trough height but kept the amplitude spectrum and changed the phase spectrum only. The time distortion made the duration of the trough to be longer and that of the crest to be shorter but its local wave period remained. Crest distortion was employed to move the location of the highest wave crest forward, and therefore, the front steepness would be increased. Fig. 4 showed the whole time series of four irregular wave trains and the comparison of strong asymmetric wave

\[ \text{Fig. 4 Time series of four irregular waves} \]
profiles. Wave profiles of the highest wave elevation in cases PH3 and PH4 met the rogue wave criteria. Amplitude spectra of cases PH3 and PH4 were presented with respect to the spectral peak frequency \( (\omega_p=5.75 \text{ rad/s}) \) in Fig. 5, which were going to be referred for the investigation of cut-off frequency dependence on the wave kinematics predictions.

\[
U_k = \frac{1}{N} \sum_{l=1}^{N} u_k^{(l)}
\]  

where \( U_k \) was the phase-averaged velocity, \( u_k^{(l)} \) was the \( k \)-component velocity obtained from the \( l \)th instantaneous velocity measurement, and \( N \) the total number of instantaneous velocities at that phase.

The mean velocity profiles were compared with results of the 3rd order Stokes wave theory for regular waves of three wave heights. Horizontal velocity profiles for three regular waves were averaged from 12 instantaneous velocity profiles taken at the same phase of each wave length and were compared with those of the 3rd order Stokes wave theory under the wave crest shown in Fig. 6. Averaged horizontal velocity profiles normalized by wave phase velocity \( (V_p) \) were plotted with the respect to the vertical axis normalized by water depth \( d \). The experimental results showed the exponentially increasing horizontal water velocity up to the wave crest and agreed well with results of the 3rd order Stokes wave theory for all three regular waves.
The RMS horizontal particle velocity was obtained by Equation (4), i.e.

\[ u_{rms} = \sqrt{\frac{\sum_{i=1}^{N} (U - u_i)^2}{N}} \]  

(4)

where \( U \) and \( u_i \) were the phase-averaged velocity and the instantaneous measured velocity, respectively, and \( N \) the number of selected regular waves.

The root mean square (RMS) values for the case PR3 were presented in Fig. 7, which showed the relatively large value near the free surface due to the limitation of PIV technique near the boundary. The error rate of horizontal velocities of selected regular waves was estimated with the ratio of \( u_{rms} \) and the maximum velocity which was within 2% except very near the free surface region.
The total acceleration are made of local and convective accelerations in Equations (5) and (6).

\[
\frac{du}{dt} = \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + w \frac{\partial u}{\partial z} \tag{5}
\]

\[
\frac{dw}{dt} = \frac{\partial w}{\partial t} + u \frac{\partial w}{\partial x} + w \frac{\partial w}{\partial z} \tag{6}
\]

where \(du/dt\) and \(dw/dt\) represent total acceleration in horizontal and vertical directions, respectively. The local and convective acceleration fields were computed by applying the time and spatial central difference scheme to successive velocity profiles (\(\Delta t = 75\) ms) and each velocity profile in \(\Delta x\) and \(\Delta z\) (2.01 mm), respectively. The acceleration was normalized by the gravity acceleration \(g\). The horizontal local and convective accelerations of experimental results had a good agreement with those of the third-order Stokes wave theory through the depth for three regular waves in Fig. 8. But, for the case PR3, the horizontal local and convective accelerations became larger than the theory results getting closer to the wave crest. Note that the nonlinear effect of accelerations became more significant closer to the wave crest as the wave steepness increased.

![Fig. 8 Horizontal local and convective acceleration under the wave crest (case PR3)](image)

3. Prediction methods of irregular wave kinematics

Three prediction methods of the linear extrapolation, Wheeler stretching \([33]\), and the modified stretching \([34]\) were introduced to estimate the wave kinematics of irregular waves.

Using Fast Fourier Transform, a two-dimensional wave elevation can be decomposed into a series of component waves,

\[
\eta(x, t) = \sum_{i=1}^{N} A_i \cos(k_i t - \omega_i t + \phi_i) \tag{7}
\]
where $A_i$, $k_i$, $\omega_i$, $\phi_i$ and $N$ are the wave amplitude, wave number, wave circular frequency, wave phase, and the number of waves, respectively. The wave number and wave circular frequency are related to each other based on the linear dispersion relationship,

$$\omega_i^2 = g \cdot k_i \cdot \tan h(k_i h)$$

where $h$ is the water depth.

According to the linear wave theory, the velocity components can be computed as following,

$$u(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\cosh[k_i(h+z)]}{\cosh k_i h} \cdot \cos(k_i x - \omega_i t + \phi_i)$$

$$w(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\sinh[k_i(h+z)]}{\cosh k_i h} \cdot \sin(k_i x - \omega_i t + \phi_i)$$

The linear extrapolation method to predict the irregular wave kinematics up to the wave crest was that Equations (9) and (10) were modified above MWL with their linear Taylor expansion about MWL:

$$u(x, z, t) = u(x, 0, t) + z \frac{\partial u}{\partial z}(x, 0, t), \quad \text{for } 0 \leq z \leq \eta$$

The linear extrapolation method made the relatively large variation of the wave kinematics prediction over the mean water level (MWL) depending on the cut-off frequency of wave spectrum. Wheeler stretching method [33] was suggested to reduce the dependency on the cut-off frequency of the linear extrapolation method, which was made to map the vertical coordinate $z$ (from the seabed to the instantaneous free surface) onto the effective vertical coordinate $z_e$

$$z_e = \frac{h(z_a - \eta)}{h + \eta}$$

where $z_a$ is the actual vertical coordinate ($-h \leq z_a \leq H_c$).

The horizontal and vertical water particle velocity, Equations (13) and (14) were developed as in the following,

$$u(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\cosh[k_i \cdot (\frac{z_a + h}{1 + \eta/h})]}{\cosh k_i h} \cdot \cos(k_i x - \omega_i t + \phi_i)$$

$$w(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\sinh[k_i \cdot (\frac{z_a + h}{1 + \eta/h})]}{\cosh k_i h} \cdot \cos(k_i x - \omega_i t + \phi_i)$$

The linear extrapolation and Wheeler stretching methods were based on the linear wave theory and could be used for the wave kinematics prediction of a linear wave and a weakly nonlinear wave. Fig. 9 provides a brief concept for determining how the horizontal water
velocity under the wave surface with the linear wave theory, the linear extrapolation, and Wheeler stretching. Velocity prediction of the linear extrapolation equaled to the linear wave theory value up to MWL and was increased with the vertical partial derivative in Equation (11) above MWL. The water particle velocity at the instantaneous free surface predicted by Wheeler stretching equaled to the linear wave theory value at MWL.

![Conceptual sketch of horizontal water particle velocity of approximate methods](image)

**Fig. 9** Conceptual sketch of horizontal water particle velocity of approximate methods

![Asymmetric factors of a rogue wave for the modified stretching method](image)

**Fig. 10** Asymmetric factors of a rogue wave for the modified stretching method

Kim et al. [34] proposed the modified stretching model to take into account the asymmetries of the wave in prediction of the highly nonlinear wave kinematics. The asymmetric factors of the rogue wave are defined as shown in Fig. 10. The modified stretching method was given by

\[ z_a = az_a^3 + bz_a^2 + cz_a + d \text{ for } -h \leq z_a \leq H_c \]  

(15)

with

\[ a = \left[ \frac{(-h + H_c) + k(h + H_c)}{(h + H_c)^3} \right] \]

\[ b = \frac{-2(h^2 - hH_c + H_c^2) - k(h + H_c)(H_c - 2h)}{(h + H_c)^3} \]

\[ c = \frac{H_c(H_c^2 - H_c h + 4h^2) + kh(h + H_c)(h - 2H_c)}{(h + H_c)^3} \]

\[ d = -\frac{h^2 H_c[kh(h + H_c) + 2H_c]}{(h + H_c)^3} \]

\[ k = (2.00 - \lambda)T_f/T_r \]

\[ \lambda = T_f/T_r \]

When \( H_c / H_t \leq 1.0, \lambda = 1.0 \), and when \( H_c / H_t > 1.0, \lambda = 1.95 \), where \( z_e \) and \( z_a \) were the effective vertical coordinate \((-h \leq z_e \leq 0)\) and the actual vertical coordinate \((-h \leq z_a \leq H_c)\), respectively.
The velocity components of the highly nonlinear wave can be obtained through substituting the effective coordinate of Equation (15) into Equation (9) and (10) for a highly nonlinear wave following as:

\[
u(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\cosh[k_i \cdot (h + z_e)]}{\cosh k_i h} \cdot \cos(k_i x - \omega_i t + \phi_i) (16)\]

\[
w(x, z, t) = \sum_{i=1}^{N} A_i \frac{g \cdot k_i}{\omega_i} \cdot \frac{\sinh[k_i \cdot (h + z_e)]}{\cosh k_i h} \cdot \sin(k_i x - \omega_i t + \phi_i) (17)\]

4. Experimental results for rogue wave kinematics

![Velocity vector fields under the highest wave crest in cases PH3 and PH4](image)

Fig. 11 Velocity vector fields under the highest wave crest in cases PH3 and PH4

Four irregular wave trains were generated from the JONSWAP spectrum with the peak enhancement factor \( \gamma \) (6.5) in 2-D wave tank (Table 2). The distortion method was applied to the irregular wave time series, and the highest wave crests of irregular wave trains in cases PH3 and PH4 met with two rogue wave criteria (\(H_H/H_s > 2.0\) and \(H_c/H_s > 1.25\)) in Table 2. Twenty eight instantaneous velocity fields were obtained with the time step of 75 ms including highest wave crests. The velocity profiles under the highest wave crest of cases PH3 and PH4 are shown on PIV snapshot images in Fig. 11. The solid line in the images indicated MWL and the velocity vectors under the free surface were well measured.

In Fig. 12–17, the measured and predicted velocity profiles were compared in normalized values; i.e., the vertical position \( z \) normalized by water depth \( d \) and velocities normalized by the phase velocity \( V_p \) of the local wave period (0.9 s) at the highest wave crest shown in Fig. 4. The measured horizontal velocities were exponentially increased up to approximately 95% of the wave phase velocity. The measured vertical velocities were gradually increased up to 30% of the wave phase velocity over MWL and reduced to zero velocity at the wave crest because the water surface was the highest level at this moment.
Comparison Study of Experiments and Predictions of Wave Kinematics for Rogue Wave

Hae Jin Choi, Seung Jae Lee, Hyo Jae Jo
Gang Nam Lee, Kwang Hyo Jung

The wave kinematics estimated by three prediction methods of the linear extrapolation, Wheeler stretching, and the modified stretching with varying spectral cut-off frequency in order to investigate the cut-off frequency dependency. The linear extrapolation method overestimated horizontal water velocities at MWL with three times of peak wave period ($\omega_p$) for cut-off frequency, but predicted similar magnitude and pattern for vertical velocities up to MWL. Because the linear extrapolation method assumed that the vertical partial derivative of a kinematic variable was constant above MWL as Equation (11), the predicted velocities were linearly increased up to the wave crest above MWL and were overestimated at three times of spectral peak frequency in comparison with the measured velocities over MWL as shown in Fig. 12 and 13.

Fig. 12 Comparison of measured horizontal velocity profiles and predictions of linear extrapolation varying cut-off frequency

Fig. 13 Comparison of measured vertical velocity profiles and predictions of linear extrapolation varying cut-off frequency
Wheeler stretching underestimated water velocities at MWL, because it was mapped the vertical coordinate \( z \) onto the effective vertical coordinate and the same magnitude of water velocity predicted by the linear wave theory at MWL was stretched to the instantaneous free surface of the wave crest. It was clear that Wheeler stretching method reduced the cut-off frequency dependency for the sake of the underestimation of the water velocity magnitude above MWL in Fig. 14 and 15.
The modified stretching was proposed to predict the wave kinematics for highly nonlinear waves by Kim et al. [34], which calculated the horizontal and vertical velocity profiles under wave crests for rogue waves with varying the cut-off frequency and compared with experimental results in Fig. 16 and 17, respectively. It showed less sensitive to be chosen the cut-off frequency than results of the linear extrapolation method, and the better agreement with experimental results over MWL than the prediction of Wheeler stretching. From the investigation of the sensitivity of cut-off frequency in comparison with experimental results, the linear extrapolation method was able to predict the water velocity profile under the highest wave crest in irregular wave train with three time of spectral peak wave frequency for cut-off frequency, and predictions of Wheeler and modified stretching methods had a good agreement with the experimental results with higher cut-off frequency than four or five times of spectral peak wave frequency.
Local accelerations under the wave crest of rogue waves were shown for cases PH3 and PH4 in Fig. 18 ~ 23. The local acceleration was calculated from consecutive velocity profiles ($\Delta t = 75$ ms) taken by PIV system under the highest wave crest of irregular wave trains with the central difference method. Because the central difference method was able to be applied below the velocity vector at the lower free surface among the previous and following PIV images, local accelerations were calculated only below the water level for lower wave crest of them. But, note that the results of local acceleration profiles calculated with the measured velocities were reasonably agreed with the Stokes 3rd order wave theory for steep regular waves in Fig. 8. The horizontal local acceleration could be reached up to $0.8 \, g \sim 1.0 \, g$ and the vertical local acceleration up to $-0.8 \, g \sim -1.0 \, g$ at near wave crest for PH3 and PH4 if it was extended up to the wave crest level.

![Graph](a) Case PH3  
![Graph](b) Case PH4

**Fig. 18** Comparison of measured horizontal local accelerations profiles and predictions of linear extrapolation varying cut-off frequency

![Graph](a) Case PH3  
![Graph](b) Case PH4

**Fig. 19** Comparison of measured vertical local accelerations profiles and predictions of linear extrapolation varying cut-off frequency
Comparison Study of Experiments and Predictions of Wave Kinematics for Rogue Wave

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Fig. 20 Comparison of measured horizontal local accelerations profiles and predictions of Wheeler stretching method varying cut-off frequency

Fig. 21 Comparison of measured vertical local accelerations profiles and predictions of Wheeler stretching method varying cut-off frequency

Linear extrapolation method was over-predicted the local acceleration with three times of spectral peak frequency for cut-off frequency in comparison with those computed from measured velocities as shown in Fig. 18 and 19 because of the constant vertical partial derivative above MWL. Local accelerations predicted by Wheeler stretching method were exponentially increased up to the wave crest with the higher cut-off frequency and had a less sensitivity for increasing the cut-off frequency in Fig. 20 and 21. However, it underestimated the horizontal and vertical local accelerations across the overall water depth.
Fig. 22 Comparison of measured horizontal local accelerations profiles and predictions of Modified stretching method varying cut-off frequency

In Fig. 22 and 23, local accelerations predicted by the modified stretching were compared with experimental results. With varying the cut-off frequency, they were reasonably increased across the water depth and higher increasing rate over MWL. The modified stretching method had a good agreement with horizontal and vertical local accelerations of experimental results at three to four times of spectral peak frequency for the cut-off frequency.

In comparison of measured velocity and local acceleration profiles and predictions of three methods in varying cut-off frequency, it showed similar trends that the modified stretching had less sensitive for the higher cut-off frequency than the linear extrapolation method, and quantitatively a better agreement with experimental results across the water depth rather than Wheeler stretching method.
Fig. 24 Comparison of measured horizontal convective accelerations profiles and predictions of linear extrapolation varying cut-off frequency

Fig. 25 Comparison of measured vertical convective accelerations profiles and predictions of linear extrapolation varying cut-off frequency
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Fig. 26 Comparison of measured horizontal convective accelerations profiles and predictions of Wheeler stretching method varying cut-off frequency

(a) Case PH3

(b) Case PH4

Fig. 27 Comparison of measured vertical convective accelerations profiles and predictions of Wheeler stretching method varying cut-off frequency

(a) Case PH3

(b) Case PH4
In Fig. 24–29, horizontal and vertical convective accelerations for cases PH3 and PH4 calculated with the application of the central difference scheme in the spatial resolution, 2.72 mm, were compared with predicted results of three methods. The convective accelerations were very small and nearly close to zero under MWL and were rapidly increased over the gravity acceleration in both directions. Even though the experimental results were some scattered, those increasing patterns were consistent for all irregular wave conditions. The horizontal accelerations predicted by three methods were well agreed with experimental results below MWL, but showed the opposite sign with experimental results over MWL, although those magnitudes were small. However, the vertical convective accelerations were predicted in the pattern of linearly increasing over MWL by the linear extrapolation method and suddenly increasing near the wave crest with higher cut-off frequency by Wheeler stretching. Modified stretching method estimated the vertical convective acceleration in the similar magnitude and
increasing pattern up to the wave crest with higher cut-off frequency overall water depth. The large discrepancy in convective accelerations, especially in horizontal direction, between experimental results and prediction methods could be caused by the limitation of three method based on the linear wave theory.

5. Conclusions

A series of experiments were conducted in 2-D wave tank to simulate the rogue waves and measure water wave kinematics including velocity and acceleration profiles. PIV technique was applied to measure the velocities under the free surface of regular and irregular waves. To verify the image acquisition and analysis methods of PIV, three different wave heights of regular wave having 0.9 s wave period were tested and those velocities and accelerations were compared with results of Stokes 3rd order theory, which showed a good agreement up to the wave crest. To generate the rogue wave in the 2-D wave tank, three steps of amplitude distortion, time distortion, and crest distortion techniques were applied for irregular wave train using JONSWAP spectrum with a peak enhancement factor of 6.5. Two of four irregular wave trains were satisfied with the two rouge wave criteria ($H/\bar{H} > 2.0$ and $H_c/\bar{H} > 1.25$). Three different prediction methods of linear extrapolation, Wheeler stretching, and modified stretching were applied to estimate water wave kinematics under the highest water elevation of irregular wave trains and to investigate the sensitivity of cut-off frequency with six different spectral peak frequencies.

Horizontal water velocity measured under the highest wave crest was increased up to 95% of the wave phase velocity calculated with the local wave period near the free surface. And, the maximum velocity in the vertical direction was measured at the middle location of the wave crest and MWL with approximately 30% of wave phase velocity. Although the local acceleration was not calculated with measured velocities up to the wave crest of rogue wave, its magnitude could be estimated approximately to the gravitation acceleration in both directions. The horizontal and vertical convective accelerations calculated with measured velocities were very small under MWL and were rapidly increased over the gravity acceleration.

The velocity and acceleration profiles had a significant effect of the cut-off frequency over MWL. Linear extrapolation predicted the water wave kinematics with strong sensitivity of cut-off frequency and overestimated them to be larger than experimental results except the horizontal local acceleration due to the assumption of constant vertical partial derivative of a kinematic variable above MWL. The water wave kinematics predicted by Wheeler stretching converged well with measurements as the cut-off frequency became higher, but underestimated those magnitudes to be smaller than the measurement results across the water depth. The modified stretching was relatively less sensitive for choosing the higher cut-off frequency than the linear extrapolation and better agreement with experimental results above MWL than other prediction methods. From the investigation of the cut-off frequency dependence on the water wave kinematics of water velocity and acceleration under the wave crest for rogue waves, it can be suggested that the cut-off frequency for the prediction of the water wave kinematics should be three times of spectral peak wave frequency for the linear extrapolation and higher frequency than four times of spectral peak wave frequency for Wheeler stretching and the modified stretching.

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Comparison Study of Experiments and Predictions of Wave Kinematics for Rogue Wave


DETAIL DESIGN OF A BALLAST CONTROL ROOM FOR AN UNDERWATER TIDAL ENERGY CONVERTER

UDC 629.5.584: 629.5.043: 629.5.062.2
Professional paper

Summary

The aim of this article is to design, precisely and fully detailed, the Control Room of a Ballast Actuation System of a marine Tidal Energy Converter. The design would respond to the detail design stage of any shipyard for its subsequent manufacture.

During the following lines, the authors have analysed the Ballast Actuation System, not only from a conceptual way, but also from the detailed design point of view, so it has been considered necessary to complete it, with a study of a technological nature, in which industrial (reducing the cost and increasing reliability) solutions have been developed for measuring and driving systems.

To elaborate the 3D detail design of the pump room, a topological design has been introduced for allowing a quicker evaluation of different design alternatives. All the machinery, piping, supports and auxiliary structures have been modelled in order to evaluate the quality of the conceptual and initial design. The primary structure, as well as the location of the equipment, have been carefully studied and analysed.

The P&ID (Piping and Instrumentation Diagrams) and access drawings have been followed in order to model the 3D pipe routing, supports and auxiliary structure. The 3D components have been defined from its main features (element ID, supplier, description, weight and centre of gravity) and its geometry (library or geometric macro model), provided by the supplier. All this work has been carried out with the intention of not only assess different alternatives, but also to generate at any time the information for manufacturing and mounting.

Key words: Tidal Energy Converter, Pump Room, Ballast Systems

1. Introduction

This technical paper presents the most significant results of the study of a Ballast Actuation System (BAS), carried out by a Research and Development (R&D) group of the Marine Engineering Technical School GIT-ERM from the Technical University of Madrid (ETSIN-UPM), consisting of the development of a specific and innovative Ballast Control
System (BCS) enabling to avoid control problems associated with free surface ballast tanks [7] [9] [13].

For this analysis, it has been chosen a device that control depth and trim as the prototype for study. The device is a Tidal Energy Converter (TEC) with the following dimensions 11 x 2 x 12 meters wingspan, with a rotor of 6.6 meters of diameter [6]. Within each float/hull has three tanks of ballast water and a Double Pump Type Ballast Actuation System (DPTBAS).

In operation, it means producing energy, this TEC works with the rotor shaft horizontal, entering the stream in the direction of the axis of the hulls, by the rounded part of them and keeping the operation depth, with a system of anchoring and mooring quite simple. Increasing the speed of the stream, the hydrodynamic forces will grow and in order to maintain the position, it is necessary to increase the net force of thrust, emptying the ballast water tank [4].

When the flow direction of the stream changes in each tidal cycle, it is necessary to turn the device 180°, passing through a vertical position with the rotor upwards [5]. To perform maintenance operations, it is necessary to emerge most of the device out of the water, eliminating the majority of the ballast water, always with the vertical axis perpendicular with the water surface [9].

Each float/hull is a separate drive system, but with a BCD integrated for the whole TEC. The system proposed, developed in this paper, is focused in elements, pipes, supports, auxiliary structures and circuits, but also it is presented an initial proposal of the sensors and control strategy.

2. Manufacturing design of the Ballast Actuation System

As these devices are not manned, and with a difficult access for maintenance, the complexity of manoeuvres in immersion (similar to a submarine) must be carried out in an autonomous way, efficient, accurate and with high reliability systems. After studying different alternatives [8], it has been perceived that a main driver for the selection of the BCS is the great volume of the tanks, plus the use of industrial elements adapted to the marine environment. After a complete study, the following main alternatives have been studied:
- Use of the pressure air for the water movement.
- Use of volumetric pumps.
- Use of centrifugal pumps.

As the water ballast needs to take water from the sea, this water is salty and with different solid particles, the use of spindle screw pumps or lobe pump is not feasible, so among the volumetric pump, flexible impeller pumps were chosen.

Starting from these conditions and of a preliminary specification, different BAS have been developed from a functional schema level. The selection process has been based on systems used in the maritime sector. Each one of these systems was revised by the R&D group of the ETSIN-UPM GIT-ERM, analysing their advantages, disadvantages and proposing modifications to the development of a new alternative. As a consequence of this process, four alternatives have been analysed, and visualizing the results of the different solutions, it was designed a BAS as a solution used by a simple air system management with flexible impeller pumps for the precise control, centrifugal pumps for managing immersion/submersion manoeuvres.

This system, that it will be referred to as a BAS based on Double Pump Type (DPTBAS), is the one to be developed and design during the following paragraphs in this paper.
After the analysis of the different solutions done above, it was concluded that the design principles of this system should be as follow:

- Insertion of man-holes in every tank, which it allows the incorporation of active elements working in its interior, to be possible its maintenance. According to this, air/water separators have been eliminated situating two independent pipes to each corner of the tank. Placing at the end of each pipe level binary sensors, which act as air/water detectors, producing that when the fluid is not the correct one on the pipe, an order is given to close the valve, located in the DPTBAS.

- Pumping hybrid solution, with three flexible impeller pumps for precision control manoeuvres in immersion and two centrifugal pumps for the immersion/submersion manoeuvres, scaled to manage that this types of movements could be achieved in the times indicated on the paper [1].

3. Structure design of the Ballast Actuation System

To develop the best possible design, it is mandatory to estimate all calculus that allow to use the maximum volume inside the DPTBAS.

When the structure of a cylindrical submarine is being designing, the first decision is to decide if the structure would have frames or not, and later on to calculate what would be the penalization about shell plate thickness if there are not frames.

After that it’s important to define the steel thickness of a cylindrical tank, with main dimensions of four meters long and two meters of diameter, with the ability to submerge until a maximum operational depth of 25 meters, where the TEC device is optimized to obtain the maximum power, while the bottom is 50 meters from the surface.

In a previous design, with grade-A steel, with stiffeners (frames) and margin due to corrosion, the shell plating thickness was of 10 (mm).

In the present study is considered a hollow cylinder, closed at the ends by two watertight bulkheads, as appears in figure 1. For this preliminary study, it is considered a cylinder without frames that will be subjected to the pressure outside of the sea.

![Cylindrical tank to be analysed](image)
To calculate the hull pressure, the regulations of the classification society **American Bureau of Shipping (ABS)** have been followed [2]. It’s necessary to calculate the maximum hull pressure according to different depths, based in the overpressure of the cylindrical hull, and of this way, find the shell thickness. This maximum pressure is the critical one for overpressure in the shell multiplied by a security factor, namely γ. The critical pressure can be also considered as the working pressure that exceeds the creep limit of the material, which produce the following **equation (1)**:

$$ P_y(h) = \frac{\sigma_y h}{R(1-F)}, $$

(1)

Where $\sigma_y$ is the yield stress of the material limit, $h$ is the shell thickness, and $R$ is the radius of the cylinder. The value of $F$ depends on material, hull cylinder forms and the frames that in the case of this structure non-existent. As it was mentioned above, the study aims to estimate the value of the thickness of the hull without reinforcements, so be used $F=0$ in the **equation (1)** for a cylindrical hull without frames.

**ABS** legislation compares this pressure $P_y$ with the buckling pressure of **Von Misses** ($P_m$) for a cylindrical geometry, in a way that has the **equation (2)**:

$$ P_m(h) = \frac{2.42E}{(1-\nu^2)^{3/4}} \left[ L/(2R) - 0.45\sqrt{h/(2R)} \right]^{5/2}, $$

(2)

Where $L$ is the length of the cylinder in the case of a structure without frames, and $E$ and $\nu$ are the **Young’s** and **Poisson’s modulus** of the material, respectively.

The maximum work pressure is given by the critical pressure ($P_c$), displayed in the **equation (3)**:

$$ P_c = \frac{\rho g H_{oc}}{\gamma}, $$

(3)

Where $\rho$ and $g$ are the density of water and gravity, respectively.

The maximum operational depth, $H_{oc}$, is estimated that may be around of 25 (m). In the calculations is considered, due to security restrictions, a safety factor of $\gamma = 0.8$, that it would increase the thickness of the shell and reduce the possibility of structure collapse.

To reduce the possibility of collapse according to different instabilities structural, the **ABS** proposes the following relationship, displayed in the **equation (4)**.

$$ P_c = \begin{cases} 
  P_m/2 & \text{if } P_m/P_y \leq 1 \\
  P_y \left[ 1 - P_y/(2P_m) \right] & \text{if } 1 < P_m/P_y < 3 \\
  5P_y/6 & \text{if } P_m/P_y \geq 3 
\end{cases} $$

(4)

Using **equations (1), (2), (3)** and (4), it’s possible to calculate numerically the thickness of the shell, $h$, attending the different depths that it may have, from 0 (m) to 50 (m).
In figure 2 it is showed the thickness that may have the pressure hull depending on the depth in the case of grade-A steel material.

![Fig. 2 Hull thickness vs depth](image)

**Figure 3** outlines the graphic of the thickness which the hull should have according to the **yield strength** and **Young's modulus** for different depths. In order the cylinder would be immersed to a greater depth, it is necessary to have a material with a higher yield strength and Young’s modulus, and of this way to reduce the shell thickness of the hull and the total weight of the cylinder. In particular, if the cylinder is going down on the water to deep depths, it should be used a material with an elastic limit, $\sigma_y$, higher as possible to reduce the thickness of the hull. On the other hand, to reduce the thickness of the hull to higher depths, the material should be with the higher Young’s modulus possible.

![Fig. 3 Colourful graphics of the thickness value, h, which should have the hull based on the creep limit, $\sigma_y$, and Young's modulus, E, for the on-job cylinder depth](image)
Finally, an additional thickness has been considered in order to take in consideration the corrosion. Although a long-life painting process has been considered, it’s important to consider the effect of the corrosion. After consider several references and sources in order to estimate the speed of the corrosion, it has been decided to take values from 0.13 (mm/year) to 0.2 (mm/year). Therefore, it has been estimated a conservatively corrosion suffered by an element of steel completely submerged in the sea.

If zinc anodes are used and also considering periodical reviews during its lifecycle of 20 years with cleaning works on the hull, coat renewals...the corrosion would not be more than one millimetre during all the useful life of the cylinder.

On the basis that the air system is communicated between all the hulls/floats, it’s almost impossible that all the floats would flooding. If an accident occurs when the pressure inside is almost equal to the atmospheric, there is a big buoyancy that avoid the system goes down to the bottom.

If an accident occurs with all the void spaces and the system descends until the bottom, 50 (m), the pressure inside would be 2.3 (bar), which would allow decrease a few millimetres the thickness of the hull. This is really interesting not just for the cost but also for the weight.

As a summary, considering a corrosion of 0.2 (mm/year) for a lifecycle of 20 years and a depth of 20 (m), the thickness of the cylindrical plates of the hull must be 13.6 (mm). And if the submarine descends up to 50 (m), thickness should be about 15 (mm) without considering corrosion effects.

4. Detail 3D model definition of the Double Pump Type Ballast Actuation System

For the definition of the detail model of the DPTBAS, it is necessary to use Computer Aided Design Systems (CAD), which help engineers and designers in various industries, in the design and construction of aircraft, cars, bridges, digital cameras, ... and of course, also in ships and naval artefacts. There are other acronyms that often accompany the acronyms CAD, such as CAM and CAE. In both the first two letters mean the same, Computer Aided, and the third letter could be M for Manufacturing, or E for Engineering.

In the market there is a wide range of CAD systems, being the one used by us the FORAN® System, whose creation goes back to the 60s of the last century [3].

![Fig. 4 Lateral hull shapes modelled with FORAN® System](image-url)
In order to design the DPTBAS, it is necessary to follow the process that follows with any other design of ship or naval artefact, which is governed by a series of stages. These stages are commonly divided, with the particularity of each geographical area, in Conceptual Design, Contractual Design, Basic Design, Detail Design, Manufacture and Maintenance.

One of the first tasks done in terms of design refers to the definition of shapes. Currently, most of the CAD Systems provide advanced tools for defining the surface model of the vessel based on the English acronym Non Uniform B-Splines (NURBS). Figure 4 shows the shapes of one of the helmets.

The next step followed in the design of the DPTBAS was the definition of the layout, which included the design and management of all pump room spaces and the preliminary drawings of pumps, pipes, supports and accesses. Most of parts of this DPTBAS was compromised in the design phase, where the highest level of accuracy was required. During this stage different design alternatives could be studied.

The definition of the DPTBAS structure has been fully defined in a 3D environment. All work has been done on the ship's product model, which encompassed the basic design and detailing process in a single environment, which simplifies the organization and increases the overall performance of the design process. This solution meets the requirements of any shipyard or technical office through capabilities, automatic mechanisms and extensive use of topology and the existence of configurable procedures and macros. For example, in Figure 5, it is possible to observe the level of detail reached in the definition of the structure, where information has been added for manufacturing, such as chamfers, welded joints, accesses...

Fig. 5 Modelling of the structure of the DPTBAS, with information of chamfers for welding process

One of the most important characteristics of the use of these advanced systems of naval design is the integration of all disciplines. This property, combined with the continuity between the phases of basic design and detail, and together with the homogeneous development of all the modules, enables the realization of collaborative engineering in an effective way.

All information regarding standards, configuration parameters and other requirements of the shipyard is centralized in a database. The raw material catalogue, plate and reinforcement
manufacturing methods and the parameters for configuring all shop drawings and outputs have also been defined.

An interactive 2D environment has also been used to define end types, holes, binders, ties, chamfers, and other parametric standards in which geometry is described from a macro of procedures that corresponds to a number of related elementary geometric operations.

The definition of standards has been created from another existing project. As a summary, the following three tasks have been performed in what has been standardized:
- Definition of the configuration parameters
- Creation of raw material
- Parametric standards.

As far as the definition of the main structure of the DPTBAS is concerned, curved and flat plates have been modelled in 3D, covering from the initial project to the detail. For this, two application contexts have been used, one for curved surfaces and another for flat ones.

It is important to emphasize that new concepts have been used for the initial design of the structure that have allowed a quick definition of a single 3D model that could be realized in the preliminary stages of the design. Even with a very preliminary definition of the hull surfaces, the level of detail increased with the definition of parts without assignment to a mounting unit or structural area. Topological relationships of ceilings, seams and curved plates have been used, incorporating advanced functions for a simple definition and modification of plates and holes.

The division of labour has been done using the concepts of surface, section and zone. Advanced splitting, multiple editing and multiple copying of the plates have been used, so attributes have been introduced in the detail design, with some of the capabilities that allow reuse of all information and significantly reduce the repetition of processes.

From the 3D model generated from the DPTBAS, it is possible to obtain workshop information for manufacturing and assembly and offer a more advanced way capable of generating, in an increasingly intuitive way, the information needed to build all the structural parts, obtaining reports and drawings.

The hull information introduced in the CAD system of the DPTBAS, for example, offers several methods for the expansion and bending of plates for the forming shop. You can create drawings and text files with all the relevant information, including numerical tables with the geometry of all the curves, for a manual marking or cutting, and in the database the pieces are stored for later nesting.

The drawings for the manufacture are automatically generated and include details of the end cut with automatic dimensioning, holes and notches, information about curving, attributes and the list of identical and symmetrical parts. It is also possible to define the assembly of curved panels with prefabrication beds. The base plane is automatically calculated and the projection of the panel in the base plane is shown for an interactive precision alignment of the panel on the guides. Configurable reports have been generated that describe all the relevant data for the assembly and marking of the panels.

An important part of the construction of the DPTBAS is the definition of its Building Strategy, therefore, we have established and organized the project according to the manufacturing and assembly processes that take place in the shipyard. The Building Strategy has been defined by organizing a hierarchical tree that describes the structure of the DPTBAS. The complete product model has been organized into a hierarchy of intermediate products and it has been possible to create alternative constructive strategies by providing interactive
functions for the creation of intermediate products, the assignment of parts, the possibility of classifying interim products using configurable attributes for each type and definition of the assembly sequence, so that drawings and lists of intermediate products can be generated automatically.

Thanks to this, we are able to know the lengths and the classification of the type of welding according to different criteria (position and level of interim product), being able to obtain the information on welding through reports and plans.

Plates and profiles can be nested according to the shipyard's standards, production methods and manufacturing methods. It has been possible to calculate the raw material needed to produce the listing of materials and generate information for numerical control machines, drawings and cutting statistics during the detail design phase.

Regarding the DPTBAS outfitting part, the first thing is the definition of the standards. The following tasks were the first to be defined: libraries, representation groups, user attribute templates, pipe standards, classes and components of equipment and piping, acronym components of Heat, Ventilation and Air Conditioning (HVAC) such as libraries of auxiliary structures and supports.

![Fig. 6 Use of FORAN® System for intelligent pipe layout](image)

The DPTBAS has been divided into zones (geographical) and systems (functional). In a 3D model environment the equipment and auxiliary elements have been defined (see figure 6). The 3D model objects have been assigned to representational groups in order to consider different levels of detail and include disassembly, security and operational representation areas. Technological attributes have been added to equipment model connections.

The equipment have been positioned, pipelines have been routed as well as the ventilation ducts and cable trays and insertion of accessories. The creation of auxiliary structures, arrangement of supports and the definition of spools of pipes and conduits have been completed.

Concepts for the initial design of outfitting have been inserted for a quick definition of equipment that can be used in the early stages of design, with direct access to existing
information on the hull structure. It has been worked in a 3D environment, making a definition of the elements of the ship that allows reusing the information in the detail design.

![Image](image_url)

**Fig. 7** Details of the position of the flow meters, where it is observed that the display is positioned optimally for consultation by the operators

All the available information on the hull structure elements, equipment, auxiliary structures, pipes and ducts has been introduced, as can be seen in figure 7. The specifications of the pipes and the diagrams are integrated into the pipeline layout together with information on parametric fittings such as valves, flanges, branches, gaps, couplings, etc. Once the information is generated during the ship design process, it is possible to detect interferences and generate drawings and reports for the clashes.

Smart Pipe Routing tools have the properties of interactive pipeline layout, by defining an associated skeleton line or by arrangement of individual elements; The geometry of a pipeline may refer to the hull, decks, internal structure, equipment, auxiliary structure, ducts, trays, pipes or joints, or any significant point of an object; Automatic checking of pipe bending and identification of inconsistencies in the layout by the automatic consistency check between the Piping and Instrumentation Diagram (P&ID) and the piping arrangement in the 3D model; As well as on-line interference testing.

Another element designed in the DPTBAS has been the auxiliary structures, such as ladders, which have been defined as a combination of a standard parametric plate and profile pieces. Each individual part has been managed like any other structural element for the purposes of producing and generating reports (for example for nesting). Auxiliary structures have been assigned to structure blocks and outfitting zones.

To create pipe supports and equipment foundations, the user defines it as part of the armament but is also associated with the structure blocks (such as plates and profiles). The supports are specific auxiliary structures linked to distributor type elements. A hierarchy is created to establish dependency between supports. Connectivity is formed between the supports and the supported elements, as can be seen in figure 8.
It is important to emphasize that before creating the 3D model, it is necessary to define the diagrams of the ship's systems, which in the case of the DPTBAS have been defined previously with this routing of distributors, pipe fittings with automatic control of connections, intelligent labelling of equipment, fittings and piping. There is an automatic association of graphic attributes depending on the fluid, the lists of materials are generated automatically and calculations are made in terms of pressure drop and flow, i.e. all the information created at this stage (equipment, piping and accessories) is available in the model 3D.

An especially important feature of the DPTBAS model is the automatic generation of isometric piping drawings according to the standards and formats of the shipyard where it is to be manufactured. Two types of isometric can be obtained: those designed for manufacturing in the workshop (known worldwide as spools): with all the information necessary for bending and the isometric ones for the assembly: with spools and accessories and providing the dimensioning relative to references of the DPTBAS.

The program in which the DPTBAS is modelled automatically controls the consistency between the pipes of the 3D model (figure 9) and the corresponding isometric drawings. Numerical control information could be exported to the bending machines of the yard’s workshop. The pipe spools allow dividing the full pipelines into pieces of conformed pipes for manufacture and grouping them in isometric assemblies (manually or fully automatic).
Finally, it is possible to generate lists of materials related to equipment, accessories and pipelines, with the possibility of obtaining the information by services, zones or constructive strategy, such as:

- Lists of equipment, accessories and pipes by zones or services.
- Special symbol reports to / from the diagrams.
- Detailed list of materials.
- Pipe routing sequence.
- Situation of a particular element with respect to parameters previously defined as request of offer, reception of the order and assembly.
- Summary of accessories and pipes by zones or services.

Conclusions

The advantage of 3D piping routing provides the user with a more realistic experience of the final routing result of the DPTBAS.

Thanks to this, the designer can interactively traces the pipe segments by always attending to the surrounding elements, heights to be saved and possible complications that may arise.

It is essential to take into account the manoeuvrability and ergonomics when routing the pipe in the space available, for example in the pump room where space for crew work is essential.
Routing in 3D allows the detection of interference with other objects in the environment such as equipment, auxiliary structures or other distributors, as well as elements of the structure of the ship resulting in the detection of collision and automatic execution of the intern or hole.

The realistic routing achieved in a 3D oriented CAD will allow to execute the calculations of pressure drop and minimum diameters with more reliability, always taking into account the elbows and heights in each routing between equipment, as well as the water hammer and the powers of connected equipment.

Obviously there are aspects that could be discuss further, as for example the ones regarding hull thickness, or the type of material that could be used. However, they could be part of a new research project.

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REFERENCES

HIGHLY PRECISE APPROXIMATION OF FREE SURFACE GREEN FUNCTION AND ITS HIGH ORDER DERIVATIVES BASED ON REFINED SUBDOMAINS

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Summary

The infinite depth free surface Green function (GF) and its high order derivatives for diffraction and radiation of water waves are considered. Especially second order derivatives are essential requirements in high-order panel method. In this paper, concerning the classical representation, composed of a semi-infinite integral involving a Bessel function and a Cauchy singularity, not only the GF and its first order derivatives but also second order derivatives are derived from four kinds of analytical series expansion and refined division of whole calculation domain. The approximations of special functions, particularly the hypergeometric function and the algorithmic applicability with different subdomains are implemented. As a result, the computation accuracy can reach $10^{-9}$ in whole domain compared with conventional methods based on direct numerical integration. Furthermore, numerical efficiency is almost equivalent to that with the classical method.

Key words: Green function; high-order derivatives; refined subdomains; series expansion

1. Introduction

The wave loads on fixed structures and the oscillatory motions of vessels free to response to the waves are common problems to be solved in ocean engineering. For ideal flow-field and based on sufficiently small motion assumption, the free surface GF is of remarkable significance in solving those problems with boundary element method.

As to frequency domain, there are generally different categories of Green function according to whether or not the harmonically time dependent unit source beneath a free surface is translating, and whether the water depth is finite or infinite. It should be noted that one concentrates on only the frequency domain infinite depth GF without translating. The evaluation of free surface GF and its derivatives is a complicated mathematical issue, especially the second order derivatives, which are very necessary in high-order panel method.

Free surface GF was known to us because of the work of John [1]. There are several different style of mathematical representations [2] for GF. Noblesse [3] advocated the two parts of the GF, which were the so-called near-field and far-field representations. On the basis of that, linear table interpolation fast method was proposed by Ponizy et al. [4], which gave a precision of $10^{-3}$. In 2017, Wu et al. [5] further proposed simple approximations to the local flow components of GF and its first order derivatives without discussion about the calculation error.
As to a representation which was in terms of a semi-infinite integral involving a Bessel function and a Cauchy singularity, Newman [6] developed the classical fast combined method with analytical series expansions and multi-dimensional polynomial approximations, which was applied in the notable hydrodynamic analysis code-WAMIT finally. Consequently, Wang [7] outlined a refined method with refined forty eight subdomains and Chebyshev polynomial giving the $10^{-5}$ precision, Zhou et al. [8] implemented a method which minimized the number of the above subdomains to twenty five. Method with Gaussian integral proposed by Yao et al. [9] reached $10^{-6}$. In 2016, new series expansions for different subdomains are proposed by Duan et al. [14], therefore, a method with five subdomains was proposed to evaluate the second order derivatives of GF. However, their studies may be more reasonable if they had considered better divisions of the domain, and the calculation of special functions, which would affect the computational accuracy and efficiency finally. As to the other representations of GF, in 2004, Peter et al. [10] proposed the eigenfunction expansion method in which the truncation terms number should reach not less than 60 in order to obtain $10^{-6}$ precision. In 2011, Elia et al. [11] advocated a semi-analytical method that divided the integral into two terms, an adaptive quadrature was used for the regular term, and the singular term was completed by an approximation function. Clement [12] introduced the pioneering method using classical fourth-order Runge–Kutta method to solve a second-order ordinary differential equation of frequency-domain GF. Similarly, in 2015, Shen et al. [13] proposed another method combing Numerov type method with Power-series method to solve this second-order ordinary differential equation. To the best of the authors’ knowledge, however, most of the above methods can give the accuracy of $10^{-5}$ to $10^{-6}$, and few investigations have been done on the second order derivatives of GF, which are the prerequisite of high-order panel method.

In this paper, one focuses on investigating the classical representation of free surface GF, composed of a semi-infinite integral involving a Bessel function and a Cauchy singularity. One outlines four kinds of representations of series expansion and furthermore acquires the refined subdomains of the whole calculation domain of physical importance. Compared with conventional direct methods based on numerical integration, which are implemented with Romberg integral method to achieve double precision result, the new method for the calculation of GF and its high order derivatives can give the precision of $10^{-9}$ in every single point of calculation domain. Furthermore, the numerical efficiency is almost equivalent to that derived from the classical method [6]. All the functional parts, including all the special function, are coded in Intel visual Fortran 2013 version, which is portable on many machines.

2. Green function and its derivatives

Firstly one considers a pulsing source point $q(\xi, \eta, \zeta)$, an image point $\overline{q}(\xi, \eta, -\zeta)$ of the source point relative to the free surface, and a field point $p(x, y, z)$ as Fig. 1 showing. The source point and the field point both are lying in the negative half-plane. $|Z|$ is the vertical distance between field point and image source point. $r$ is the horizontal distance between these two points, $R_{pq}$ is the distance between them, $R_{pq}$ is the distance between field point and image source point.

Where $|Z| = |z + \zeta|$, $r = \sqrt{(x-\xi)^2 + (y-\eta)^2}$, $R_{pq} = \sqrt{r^2 + (z-\zeta)^2}$, $R_{-} = \sqrt{r^2 + Z^2}$

The following expression [1] is the complex GF of infinite depth water in frequency domain

$$G_o(p, q) = \frac{1}{R_{pq}} + P. V. \int_{0}^{\infty} \frac{k + k_0}{k - k_0} e^{i\zeta} J_0(kr) dk \mp 2i\pi k_0 e^{i\zeta} J_0(k_0 r)$$

(1)
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Where \( k_0 = \omega^2 / g \) is wave number, \( \omega \) is wave frequency, \( g \) is the gravitational acceleration, \( J_0 \) is zero-order Bessel function of the first kind, \( P.V. \int \) means principal value of the integral. \(-2i\pi k_0 e^{ik_0Z} J_0(k_0r)\) is related to time exponential item \( e^{int} \), \( +2i\pi k_0 e^{ik_0Z} J_0(k_0r)\) is related to time exponential item \( e^{-int} \).

![Diagram](image)

**Fig. 1** Layout and notations of field and source point

Substituting the identity \( \frac{1}{R_{pq}} = \frac{1}{\sqrt{r^2 + Z^2}} = \int_0^{\infty} e^{ikZ} J_0(kr) dk \) into Equation (1), yields,

\[
G_o(p, q) = \frac{1}{R_{pq}} + \frac{1}{R_{pq}} + 2\int_0^{\infty} \frac{k_0}{k-k_0} e^{ikZ} J_0(kr)dk + 2i\pi k_0 e^{ikZ} J_0(k_0r)
\]

(2)

Defining \( X = k_0r, Y = -k_0Z, t = k / k_0 \).

Then these two coordinates of \( X \) and \( Y \) may take on all positive values, one quadrant of a two-dimensional plane should be considered. The equation (2) may be written in the form

\[
G_o(p, q) = \frac{1}{R_{pq}} + \frac{1}{R_{pq}} + k_0 F(X, Y) + 2i\pi k_0 e^{-Y} J_0(X)
\]

(3)

Where

\[
F(X, Y) = \int_0^{\infty} \frac{1}{t-1} e^{-Y} J_0(Xt) dt
\]

(4)

The elementary singularity \( 1/R_{pq} + 1/R_{pq} \) and imaginary part \( 2\pi i k_0 e^{-Y} J_0(X) \) can be implemented maturely with numerical or analytic method. So the issue has been translated from the evaluation of \( G_o(p, q) \) and its derivatives to those of \( F(X, Y) \).

Taking further treatment to infinite integration [6, 8] of equation, yields,

\[
F(X, Y) = -2\int_0^{\infty} \left( X^2 + t^2 \right)^{-1/2} e^{-Y} dt - \pi e^{-Y} \left[ H_0(X) + Y_0(X) \right]
\]

(5)

From equation (5), one can derive the following first and second order partial derivatives of function \( F(X, Y) \) with respect to independent variable \( X \) and \( Y \):
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\[
\frac{\partial F(X,Y)}{\partial X} = -2e^{-y} + \pi e^{-y} \left[ H_1(X) + Y_1(X) \right] + 2X \int_0^y \left( X^2 + t^2 \right)^{-3/2} e^{-y} dt
\]

(6)

\[
\frac{\partial F(X,Y)}{\partial Y} = -\frac{2}{R} - F(X,Y)
\]

(7)

\[
\frac{\partial^2 F(X,Y)}{\partial X^2} = \frac{\pi}{2} e^{-y} \left[ \frac{2X}{3\pi} + Y_0(X) - Y_2(X) + H_0(X) - H_2(X) \right] + 2 \int_0^y \left( X^2 + t^2 \right)^{-3/2} (t^2 - 2X^2)e^{-y} dt
\]

(8)

\[
\frac{\partial^2 F(X,Y)}{\partial Y^2} = \frac{2Y}{R^3} + \frac{2}{R} + F(X,Y)
\]

(9)

\[
\frac{\partial^2 F(X,Y)}{\partial X \partial Y} = \frac{2X}{R^3} - \frac{\partial F(X,Y)}{\partial X}
\]

(10)

Where \( R^2 = X^2 + Y^2 \), \( H_n(X) \) is the n-th order Struve function, \( J_n(X) \) is the n-th order Bessel function of the first type, \( Y_n(X) \) is the n-th order Bessel function of the second type.

Here one can find that \( \partial F(X,Y)/\partial Y \) and \( \partial^2 F(X,Y)/\partial Y^2 \), \( \partial^2 F(X,Y)/\partial X \partial Y \) are related to \( F(X,Y) \), \( \partial F(X,Y)/\partial X \) respectively. So if the calculation of GF and its derivatives are based on Equation (5), the calculation emphasis is the evaluation of \( F(X,Y), \partial F(X,Y)/\partial X, \partial^2 F(X,Y)/\partial X \partial Y \).

Besides, the special function of Struve function, Bessel function of first order and Bessel function of second order have the following identity [15] which will be implemented in the latter manipulation.

\[
Y_2(X) = \frac{2Y(X)}{X} - Y_0(X); \quad Y_0^{(1)}(X) = -Y_1(X); \quad Y_1^{(1)}(X) = \frac{1}{2} \left[ Y_0(X) - Y_2(X) \right]
\]

\[
J_2(X) = \frac{2J(X)}{X} - J_0(X); \quad J_0^{(1)}(X) = -J_1(X); \quad J_1^{(1)}(X) = \frac{1}{2} \left[ J_0(X) - J_2(X) \right]
\]

\[
H_2(X) = \frac{2X}{3\pi} + \frac{2H(X)}{X} - H_0(X); \quad H_0^{(1)}(X) = \frac{2}{\pi} - H_1(X)
\]

\[
H_1^{(1)}(X) = \frac{1}{2} \left[ \frac{2X}{3\pi} + H_0(X) - H_2(X) \right]
\]

\( Y_n^{(m)}(X) \) is the m-th order derivatives of Bessel function of the second type. \( H_n^{(m)}(X) \) is the m-th order derivatives of Struve function. \( J_n^{(m)}(X) \) is the m-th order derivatives of Bessel function of the first type.

Furthermore, without loss of generality and ambiguity, here one assumes that the \( F(X,Y) \) represents the GF in the latter discussion.

3. Series Expansion Method

The division of the whole domain of physical importance is the important precondition of numerical evaluation of GF, in this paper which is derived from different kinds of Series Expansion Method (SEM). In this part, considering the different location of \( X \) and \( Y \) in the XY
two-dimensional plane, one primarily outlines four different kinds of SEM. On the basis of these kinds of SEM, the refined boundary of the whole domain is discussed in the latter part.

3.1 SEM1

When \( X \) is relatively small beside \( Y \) axis, \( J_0(Xt) \) can be expanded as the following identity in even powers of \( Xt \):

\[
J_0(Xt) = \sum_{n=0}^{\infty} \frac{(-1)^n}{(n!)^2} \left( \frac{Xt}{2} \right)^{2n}
\]

Substituting the above equation into equation (4), and after successive partial integration, yields,

\[
F(X,Y) = 2 \sum_{n=0}^{\infty} \frac{(-1)^n}{(n!)^2} \left( \frac{X}{2} \right)^{2n} \int_0^\infty \frac{t^n}{t-1} e^{-by} dt
\]

\[
= -2e^{-y}Ei(Y) + 2 \sum_{n=0}^{\infty} \frac{(-1)^n}{(n!)^2} \left( \frac{X}{2} \right)^{2n} \left[ \sum_{m=1}^{\infty} \frac{(m-1)!}{Y^m} - e^{-y}Ei(Y) \right]
\]

Here, \( Ei(Y) \) is the exponential integral function. From equation (11), one can obtain the following partial derivatives of function \( F(X,Y) \) with respect to \( X \) and \( Y \).

\[
\frac{\partial F(X,Y)}{\partial X} = X \sum_{n=1}^{\infty} \frac{n}{(n!)^2} \left[ \sum_{m=1}^{\infty} \frac{(m-1)!}{Y^m} - e^{-y}Ei(Y) \right]
\]

\[
\frac{\partial F(X,Y)}{\partial Y} = 2 \left( e^{-y}Ei(Y) - Y^{-1} \sum_{n=1}^{\infty} \frac{1}{(n!)^2} \left[ \sum_{m=1}^{\infty} \frac{m!}{Y^{m+1}} + Y^{-1} - e^{-y}Ei(Y) \right] \right)
\]

\[
\frac{\partial^2 F(X,Y)}{\partial X^2} = 2 \sum_{n=1}^{\infty} \frac{1}{(n!)^2} \left[ \sum_{m=1}^{\infty} \frac{(m-1)!}{Y^m} - e^{-y}Ei(Y) \right]
\]

\[
\frac{\partial^2 F(X,Y)}{\partial Y^2} = 2 \left( e^{-y}Ei(Y) + Y^{-1} + Y^{-2} \right) + 2 \sum_{n=1}^{\infty} \frac{1}{(n!)^2} \left[ \sum_{m=1}^{\infty} \frac{(m+1)!}{Y^{m+2}} + Y^{-1} + Y^{-2} - e^{-y}Ei(Y) \right]
\]

\[
\frac{\partial^2 F(X,Y)}{\partial X \partial Y} = X \sum_{n=1}^{\infty} \frac{n}{(n!)^2} \left[ \sum_{m=1}^{\infty} \frac{m!}{Y^{m+1}} + Y^{-1} - e^{-y}Ei(Y) \right]
\]

Here, all these equations are consist of a double infinite series with positive powers of \( X \) and negative powers of \( Y \), one can implement equations (12)~(17) to approximate the GF \( F(X,Y) \) and its derivatives.
3.2 SEM2

To some extent, this SEM works when \( X \) and \( Y \) is moderate value, relative to the other 3 SEM. The following identity exists [14]:

\[
\int_{0}^{Y} (X^2 + t^2)^{-3/2} e^{\gamma t} dt = \frac{1}{X} \frac{\partial}{\partial X} \int_{0}^{Y} (X^2 + t^2)^{1/2} e^{\gamma t} dt
\]  

(18)

So the evaluation will be very convenient if one can makes the integral part of equation (5)~(10) to be expressed with \((X^2 + t^2)^{1/2}\) or \((X^2 + t^2)^{-3/2}\). So (6) and (8) can be transformed as follows.

As to the integral part of equation (6), Making twice trigonometric substitutions \((t = X \tan \varphi)\) and partial integration, after some tedious manipulations, yields,

\[
X \int_{0}^{Y} (X^2 + t^2)^{-3/2} e^{-\gamma t} dt = \frac{Y}{XR} - \frac{R}{X} e^{-\gamma Y} + \frac{1}{X} \int_{0}^{Y} (X^2 + t^2)^{1/2} e^{-\gamma t} dt
\]

Substituting the above equation into equation (6), thus,

\[
\frac{\partial F(X,Y)}{\partial X} = \pi e^{-\gamma} \left[ H_1(X) + Y_1(X) \right] + \frac{2Y}{XR} - \frac{2R}{X} + \frac{2}{X} \int_{0}^{Y} (X^2 + t^2)^{1/2} e^{-\gamma t} dt
\]  

(19)

The same careful manipulations can be taken with equation (8), it becomes

\[
2 \int_{0}^{Y} (X^2 + t^2)^{-1/2} (t^2 - 2X^2) e^{\gamma t} dt
\]

\[
= 2 \left( \frac{Y^3}{X^2 R^3} - \frac{2Y}{X^2 R} \right) + \frac{2}{X^2} \left[ \int_{0}^{Y} t X^2 \left( X^2 + t^2 \right)^{-3/2} e^{\gamma t} dt + \int_{0}^{Y} t \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt \right]
\]

\[
= 2 \left( \frac{Y^3}{X^2 R^3} - \frac{2Y}{X^2 R} \right) + \frac{2Y^2}{X^2 R} + \frac{2e^{-\gamma Y}}{X^2} \left[ \int_{0}^{Y} X^2 \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt - \int_{0}^{Y} \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt \right]
\]

Substituting the above equation into equation (8), thus,

\[
\frac{\partial^2 F(X,Y)}{\partial X^2} = \frac{\pi}{2} e^{-\gamma} \left[ \frac{2X}{3\pi} + Y_0(X) - Y_2(X) + H_0(X) - H_2(X) \right]
\]

\[
+ 2 \left( \frac{Y^3}{X^2 R^3} - \frac{2Y}{X^2 R} \right) + \frac{2Y^2}{X^2 R} + \frac{2e^{-\gamma Y}}{X^2} \left[ \int_{0}^{Y} X^2 \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt - \int_{0}^{Y} \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt \right]
\]  

(20)

So one arrives at the required expression about \( F(X,Y) \), \( \partial F(X,Y)/\partial X \), and \( \partial^2 F(X,Y)/\partial X^2 \) containing \((X^2 + t^2)^{1/2}\) or \((X^2 + t^2)^{-1/2}\). In this SEM, kernel of the issue is the evaluation of the equation (5), (19) and (20).

Furthermore, one introduces the following identity:

\[
\left( X^2 + t^2 \right)^{1/2} = \int_{0}^{X} \frac{u}{\sqrt{u^2 + t^2}} du + t
\]

Then after some manipulations, yields,

\[
\int_{0}^{Y} \left( X^2 + t^2 \right)^{1/2} e^{\gamma t} dt = \int_{0}^{Y} e^{\gamma t} dt \int_{0}^{X} \frac{u}{\sqrt{u^2 + t^2}} du + Ye^{\gamma} - e^{\gamma} + 1
\]
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For the convenient evaluation of the above equation, one introduces the coordinate substitution \( u = r \cos \theta, t = r \sin \theta \), the integral range about \( X \) and \( Y \) that is divided into the following four parts [14], is illustrated as fig. 2.

\[
\begin{align*}
\sum_{i_1} & : \begin{cases} r \in (0, X) \\ \theta \in (0, \theta_0) \end{cases} \\
\sum_{i_2} & : \begin{cases} r \in (X, R) \\ \theta \in (\arccos X/r, \theta_0) \end{cases} \\
\sum_{i_3} & : \begin{cases} r \in (0, Y) \\ \theta \in (\theta_0, \pi/2) \end{cases} \\
\sum_{i_4} & : \begin{cases} r \in (Y, R) \\ \theta \in (\theta_0, \arcsin Y/r) \end{cases}
\end{align*}
\]

Where \( \theta_0 = \arcsin Y/X \).

After some manipulations of integration on the four parts, one obtains

\[
e^{-y} \int_0^y (X^2 + t^2)^{1/2} dt = e^{-y} \left( \sum_{i_1} + \sum_{i_2} + \sum_{i_3} + \sum_{i_4} \right) \frac{ue^t}{\sqrt{u^2 + t^2}} dt + e^{-y} \left( Ye^{-y} - e^{-y} + 1 \right) = R - Xe^{-y} - e^{-y} \int_x^R e^{\sqrt{t^2 - x^2}} dr
\]

One makes substitution \( t = r/X \) to the integral part of the above equation, and implements the Taylor expansion to \( e^{y^{-1}} \) in \( t^2 - 1 = 0 \), thus

\[
e^{-y} \int_x^R e^{\sqrt{t^2 - x^2}} dr = e^{-y} \sum_{n=0}^{\infty} \frac{X^{n+1}}{n!} \int_1^{R/X} \left( t^2 - 1 \right)^{n/2} dt
\]

Then utilizing the definition of Gauss hypergeometric function \( \text{F}_1(a,b,c,z) \) [15], one finds

\[
\int_1^{R/X} \left( t^2 - 1 \right)^{n/2} dt = \text{Re} \left\{ \frac{i^{-n}}{2} \left[ \sqrt{\pi} \Gamma \left( \frac{1+n}{2} \right) + \frac{2R}{X^2} \text{F}_1 \left( \frac{1+n}{2}, -\frac{n}{2}, \frac{3R^2}{2X^2} \right) \right] \right\}
\]

Where \( \Gamma(z) \) is the Gamma function. Meanwhile, one introduces the following two identities.

\[
\sum_{n=0}^{\infty} \frac{i^{-n} X^{n+1} \Gamma \left( \frac{1+n}{2} \right)}{\Gamma \left( \frac{n+3}{2} \right) n!} = \sqrt{\pi} X \left( H_{-1} \left( X \right) - J_i \left( X \right) i \right)
\]
\[ H_{-1}(X) = \frac{2}{\pi} - H_1(X) \]  

(25)

Substituting equation (23) (22) (24) (25) into (21), one has

\[ \int_0^\infty \left( X^2 + t^2 \right)^{-\nu/2} e^{-\nu t} dt = R - \frac{\pi X}{2} e^{-\nu} H_1(X) \]

\[ -e^{-\nu} R \sum_{n=0}^\infty \frac{X^n}{n!} \text{Re} \left[ i^{-n} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{X^2}{2} \right) \right] \]  

(26)

Considering equations (18) (26) and the following identity

\[ \frac{\partial}{\partial z} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{z}{2} \right) = \frac{(1-z)^{n/2} - \frac{1}{2} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{z}{2} \right)}{2z} \]

After some careful manipulations, one has

\[ \int_0^\infty \left( X^2 + t^2 \right)^{-\nu/2} e^{-\nu t} dt = R - X e^{-\nu} - e^{-\nu} \int_X^R e^{\nu^2 - x^2} dx = \frac{1}{R} + \frac{\pi}{2} e^{-\nu} H_0(X) \]

\[ -\frac{R}{X^2} e^{-\nu} \sum_{n=0}^\infty \frac{X^n}{n!} \text{Re} \left[ i^{-n} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{X^2}{2} \right) \right] + \frac{Y^2}{RX^2} \]  

(27)

Substituting equation (26) into (19), (27) into (5), (26) (27) into (20) respectively and using the relation of special functions presented earlier, one gets

\[ F(X,Y) = -\pi e^{-Y} Y_0(X) - \frac{2R}{X^2} + \]

\[ \frac{2R}{X^2} e^{-Y} \sum_{n=0}^\infty \frac{X^n}{n!} \text{Re} \left[ i^{-n} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{X^2}{2} \right) \right] \]  

(28)

\[ \frac{\partial F(X,Y)}{\partial X} = \pi e^{-Y} Y_1(X) + \frac{2Y}{XR} - 2e^{-Y} \sum_{n=0}^\infty \frac{X^{n+1}}{n!} \text{Re} \left[ i^{-n} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{X^2}{2} \right) \right] \]  

(29)

\[ \frac{\partial^2 F(X,Y)}{\partial X^2} = \pi e^{-Y} \left[ Y_0(X) - \frac{Y_1(X)}{X} + 2 \left( \frac{Y^3}{X^2 R^3} - \frac{2Y}{X^2 R} \right) + \frac{Y^2}{X^2 R} \right] + \frac{2R}{X^2} e^{-Y} \sum_{n=0}^\infty \frac{X^n}{n!} \text{Re} \left[ i^{-n} F_1 \left( 1, -\frac{n}{2} ; \frac{3}{2}, \frac{X^2}{2} \right) \right] \]  

(30)

Finally, one arrives at the required expressions (28) (29) (30) (7) (9) (10) with which the evaluation can be implemented.

3.3 SEM3

When X is not very small and X/Y is less than 2, one implements the Taylor expansion to \( \left( X^2 + t^2 \right)^{-\nu/2} \ln t^2 = 0 \), thus
\[ \left[ X^2 + t^2 \right]^{-1/2} = \frac{1}{X} + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!!}{2^n n! X^{2n+1}} t^n \]

Substituting the above equation into the integral part of (5), and exchanging the order of integration and summation, yields

\[ \int_0^Y \left( X^2 + t^2 \right)^{-1/2} e^{-Y} dt = \frac{1}{X} \int_0^Y e^{-Y} dt + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!!}{2^n n! X^{2n+1}} \int_0^Y t^n e^{-Y} dv \]

\[ = \frac{1 - e^{-Y}}{X} + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!!}{2^n n! X^{2n+1}} C_n(Y) \]  

(31)

Where \( C_n(Y) = \int_0^Y t^{2n} e^{-Y} dv \)

Successive partial integration of the right integrals of \( C_n(Y) \) yields the recursion formula

\[ C_n(Y) = Y^{2n} - 2nY^{2n-1} + 2n(2n-1)C_{n-1}(Y) \]

Where \( C_0(Y) = 1 - e^{-Y} \).

Substituting equation (31) into (5), one has

\[ F(X,Y) = -\pi e^{-Y} [H_0(X) + Y_0(X)] - \frac{2(1-e^{-Y})}{X} - 2 \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!!}{2^n n! X^{2n+1}} C_n(Y) \]  

(32)

From equation (32), one can achieves the derivatives of function \( F(X,Y) \) with respect to \( X \) and \( Y \), and simplifies those equations with the former special function identity, thus

\[ \frac{\partial F(X,Y)}{\partial X} = \pi e^{-Y} \left( H_1(X) + Y_1(X) - \frac{2}{\pi} \right) + \frac{2(1-e^{-Y})}{X^2} + \]

\[ 2 \sum_{n=1}^{\infty} \frac{(-1)^n (2n+1)(2n-1)!!}{2^n n! X^{2n+2}} C_n(Y) \]  

(33)

\[ \frac{\partial^2 F(X,Y)}{\partial X^2} = \pi e^{-Y} \left[ H_0(X) - \frac{H_1(X)}{X} + Y_0(X) \right] - \frac{4}{X^3} (1-e^{-Y}) \]

\[ -2 \sum_{n=1}^{\infty} \frac{(-1)^n (2n+1)(2n+2)(2n-1)!!}{2^n n! X^{2n+3}} C_n(Y) \]  

(34)

Finally, the evaluation can be implemented with the required expressions (32) (33) (34) (7) (9) (10).

3.4 SEM

When \( X \) and \( Y \) are all not very small, One makes substitutions \( u = t/Y, v = u - 1 \) to the integral part of (5), thus

\[ \int_0^Y \left( X^2 + t^2 \right)^{-1/2} e^{-Y} dt = \int_{-1}^0 \left[ \frac{R^2}{Y^2} + (v^2 + 2v) \right]^{-1/2} e^{vY} dv \]  

(35)
Then implementing the Taylor expansion to \( \left[ \frac{R^2}{Y^2} + (v^2 + 2v) \right]^{\frac{1}{2}} \) in \( v^2 + 2v = 0 \), one has
\[
\left[ \frac{R^2}{Y^2} + (v^2 + 2v) \right]^{\frac{1}{2}} = \frac{Y}{R} + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+1} (v^2 + 2v)^n
\]
Substituting the above equation into (35), and exchanging the order of integration and summation, yields
\[
\int_0^Y (X^2 + r^2)^{\frac{1}{2}} e^{-r} dr = \frac{Y}{R} \int_0^1 e^y dy + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+1} \int_0^1 (v^2 + 2v)^n e^y dv
\]
\[
= \frac{1-e^{-Y}}{R} + \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+1} B_n(Y)
\]
(36)
Where \( B_n(Y) = \int_0^1 (v^2 + 2v)^n e^y dv \)

Successive partial integration of the right integrals of \( B_n(Y) \) yields the recursion formula
\[
B_n(Y) = \frac{(-1)^{n+1}}{Y} e^{-Y} + \frac{2n(2n-1)}{Y^2} B_{n-1}(Y) + \frac{4n(n-1)}{Y^2} B_{n-2}(Y)
\]
Where \( B_0(Y) = \frac{1-e^{-Y}}{Y} ; B_1(Y) = \left( \frac{1-2}{Y^3} \right) e^{-Y} - 2 \left( \frac{1}{Y^2} - \frac{1}{Y^3} \right) \)

Substituting equation (36) into (5), yields
\[
F(X,Y) = -\pi e^{-Y} \left[ H_0(X) + Y_0(X) \right] - \frac{2 \left( 1-e^{-Y} \right)}{R} - \frac{16}{Y^2} \sum_{n=1}^{\infty} \frac{(-1)^n (2n-1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+1} B_n(Y)
\]
(37)

Similar to the subdomain 3, one has
\[
\frac{\partial F(X,Y)}{\partial X} = \pi e^{-Y} \left( H_1(X) + Y_1(X) - \frac{2}{\pi} \right) + \frac{2X \left( 1-e^{-Y} \right)}{R^3} + \frac{2X}{Y^2} \sum_{n=1}^{\infty} \frac{(-1)^n (2n+1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+3} B_n(Y)
\]
(38)
\[
\frac{\partial^2 F(X,Y)}{\partial X^2} = \pi e^{-Y} \left[ H_0(X) - \frac{H_1(X)}{X} + Y_0(X) - \frac{Y_1(X)}{X} \right] + \left( \frac{2}{R^3} - \frac{6X^2}{R^5} \right) \left( 1-e^{-Y} \right) + \frac{2}{Y^2} \sum_{n=1}^{\infty} \frac{(-1)^n (2n+1)!}{2^n n!} \left( \frac{Y}{R} \right)^{2n+3} \left( 1- \frac{X^2 (2n+3)}{R^2} \right) B_n(Y)
\]
(39)

Finally, one obtains the required expressions (37) (38) (39) (7) (9) (10) with which the evaluation can be implemented.
4. Numerical Results and Discussion

From final identities of all the above four kinds of SEM, one concludes that there are two evaluating emphases to be accomplished. One is accurate and efficient approximations of those special functions, which are \( Y_0(X) \), \( Y_1(X) \), \( J_0(X) \), \( J_1(X) \), \( H_0(X) \), \( H_1(X) \), \( Ei(Y) \), \( _2F_1(a,b,c,z) \), the other is the exact division of the whole domain of physical importance.

For convenient comparison, there are four kinds of different methods. The first kind method [14] is called M\(_1\), which will be used to the comparison of computational precision. M\(_2\) stands for the classical fast method [6]. The third method derived from equations (5) (6) (7) (8) (9) (10), called M\(_{\text{pre}}\), is the direct numerical method, which can provide double precision result employing Romberg integral method. The last method from this paper is called M\(_{\text{new}}\). One benchmarks the corresponding numerical errors of M\(_1\) and M\(_{\text{new}}\) against the M\(_{\text{pre}}\) results, with the representations E\(_1\)=[M\(_1\)-M\(_{\text{pre}}\)] and E\(_{\text{new}}\)=[M\(_{\text{new}}\)-M\(_{\text{pre}}\)].

The notations T\(_1\), T\(_2\), T\(_{\text{pre}}\) and T\(_{\text{new}}\) represent the computational time of M\(_1\), M\(_2\) and M\(_{\text{pre}}\), M\(_{\text{new}}\) respectively. Although X and Y may take on all positive values, without loss of generality one assumes that for error analysis, calculation range of X and Y is up to 40, in which the step length of X and Y is 0.2, and that for efficiency analysis, calculation interval of X and Y is (0, 500] with variable step lengths, which are 0.1, 0.15, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8 respectively. The constant range of X-Y plane with changeable step length is corresponding to one certain wetted surface with different number of discrete panel elements. The following numerical calculations were performed on a desktop computer (Microsoft Windows 7, 64 bit, Intel Core i5-5200 2.2 GHz/ 4 core, 4 GB implementation memory).

4.1 A note on the approximations of special functions

Efficient approximations of \( Y_0(X) \), \( Y_1(X) \), \( J_0(X) \), \( J_1(X) \) are derived from rational functions [16], which are theoretically accurate to at least 18 significant decimal digits. The fast calculation method about \( H_0(X) \), \( H_1(X) \) is proposed by Zhang et al. [17], with precision up to \( 10^{-15} \). The rational Chebyshev approximation [18] of \( Ei(Y) \) has maximal relative errors as low as from \( 8 \times 10^{-19} \) to \( 2 \times 10^{-21} \). Relative to the above special functions, numerical computation of the Gauss hypergeometric function \( _2F_1(a,b,c,z) \) is a challenging task [19][20]. Here one introduces a fast and accurate computation method of this function [19].

Furthermore, the computational efforts of all these special functions are also acceptable, outlined in Table 1 (the consumed time is for one time, which is the average computational efforts from \( 10^8 \) times calculation of the special function). It can be found that the calculation of Gauss hypergeometric function is a little time-consuming, compared with those of other special functions.

<table>
<thead>
<tr>
<th>Special function</th>
<th>Consumed time/ s/once</th>
<th>Special function</th>
<th>Consumed time/ s/once</th>
</tr>
</thead>
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<tr>
<td>( _2F_1 )</td>
<td>1.42\times10^{-5}</td>
<td>( Y_0 )</td>
<td>3.00\times10^{-7}</td>
</tr>
<tr>
<td>( Ei )</td>
<td>2.00\times10^{-7}</td>
<td>( Y_1 )</td>
<td>2.00\times10^{-7}</td>
</tr>
<tr>
<td>( J_0 )</td>
<td>1.00\times10^{-7}</td>
<td>( H_0 )</td>
<td>4.00\times10^{-7}</td>
</tr>
<tr>
<td>( J_1 )</td>
<td>1.00\times10^{-7}</td>
<td>( H_1 )</td>
<td>3.00\times10^{-7}</td>
</tr>
</tbody>
</table>

4.2 The refined division of four subdomains

Generally, exact division of the domain is closely related to the calculation precision and efficiency under appropriate SEM. In this paper, the algorithmic applicability with different (X, Y) and constant truncated number of each SEM is outlined on the X-Y two-dimensional plane.
Furthermore, the exact division and different truncated number of SEM is derived from the balance between calculation precision and efficiency.

Firstly, one derives the representative numerical $E_{\text{new}}$ of $F(X,Y)$ with constant truncated numbers ($N_1=15$, $N_2=35$, $N_3=13$, $N_4=15$ respectively) of each SEM.

From Fig 3(1), it can be clearly found that SEM1 is available when $Y/X$ is more than 2.0 in one quadrant of the two-dimensional $X$-$Y$ plane. Fig. 3(2) illustrates that the approximation of the square region (when $X$ is less than 10 and $Y$ is less than 11) can be derived from SEM2. Fig. 3(3) shows that SEM3 is suitable when $X/Y$ is more than 2.0 and $X$ is more than 6.5. From Fig. 3(4) one can conclude that SEM4 can be applied to the whole zone except when $X$ is less than 11.0 and $Y$ is less than 31.0.

However, from the computational efficiency of special functions presented earlier, one believes that efficiency sorting order of these four SEM is: $\text{SEM1} \approx \text{SEM3} > \text{SEM4} > \text{SEM2}$, as to exact division of the whole domain, the above coarse result is far from enough. Consequently based on the general consideration for three aspects, which are the efficiency difference between these SEM, the computational error control of transitional region (especially the exact boundary of the SEM2), the balance between the calculation precision and efficiency of the whole domain, one outlines the refined boundary and truncated number of infinite series for each SEM as follows.
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In Fig. 4 of X and Y coordinate plane, the D_i is shorted for subdomain i (i=1, 2, 3, 4), N_i stands for the truncated number of infinite series for subdomain i (i=1, 2, 3, 4). The boundaries of the whole calculating domain are Y/X=2, Y=15, X=9.5, X=6.5, and X/Y=2. With the desired accuracy as the precondition, for the sake of efficiency, D_1, D_2, D_4 is re-divided into different zones. It must be pointed out that this introduction of the re-division for D_1, D_2 and D_4 with different truncated number did not increase computational efforts in practice compared those derived from selection of evaluation algorithm.

D_1 is re-divided into two zones with different truncated number of infinite series, its following boundary is Y=14 and Y=17, with truncated number N_1=15 and N_1=19 respectively. D_2 is re-divided into four zones with different truncated number of infinite series, its following boundary is Y=6 and x=4.5, Y=8 and X=7.5, and Y=11. The truncated number of these four zones are N_2=26, N_2=30, N_2=36, and N_4=42 respectively. D_4 is re-divided into two zones with boundary X=14, the truncated number of these two zones are N_4=20, N_4=15 respectively.

4.3 Numerical results of Green function and its derivatives

Fig. 5 shows numerical E_1 for F(X,Y) with M_1. One finds that M_1 can obtain the accuracy of 10^{-9} in some area, not covering the whole domain. The precision of some sizeable areas is about 10^{-6}~10^{-8}. The numerical results also shows that there are actually some areas with the precision of 10^{-2}~10^{-5}, to some extent, which is not sufficiently accurate for computational
application, such as for the wave-induced motions, forces, and resistance [21]. However, Fig. 6 gives numerical $E_{\text{new}}$ for $F(X,Y)$. It can be obviously concluded that precision of all the evaluating points with different $X$ and $Y$ from the whole domain can reach at least $10^{-9}$, which is due to the proper SEM, the highly precise approximation of special function and refined subdomain division. The same results can be found in Table 2, which is the comparison of numerical results between $M_1$ and $M_{\text{new}}$.

### Table 2 Comparison of numerical error for $F(X,Y)$ between $M_1$ and $M_{\text{new}}$

<table>
<thead>
<tr>
<th>$X$</th>
<th>$Y$</th>
<th>$E_1$</th>
<th>$E_{\text{new}}$</th>
<th>$X$</th>
<th>$Y$</th>
<th>$E_1$</th>
<th>$E_{\text{new}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.8</td>
<td>0.8</td>
<td>$0.00\times10^0$</td>
<td>$0.00\times10^0$</td>
<td>8.2</td>
<td>13</td>
<td>$3.34\times10^{-10}$</td>
<td>$3.34\times10^{-10}$</td>
</tr>
<tr>
<td>0.8</td>
<td>5</td>
<td>$8.99\times10^{-15}$</td>
<td>$0.00\times10^0$</td>
<td>8.2</td>
<td>15</td>
<td>$3.67\times10^{-10}$</td>
<td>$3.67\times10^{-10}$</td>
</tr>
<tr>
<td>0.8</td>
<td>10</td>
<td>$5.57\times10^{-12}$</td>
<td>$0.00\times10^0$</td>
<td>8.2</td>
<td>18</td>
<td>$1.68\times10^{-7}$</td>
<td>$2.50\times10^{-10}$</td>
</tr>
<tr>
<td>0.8</td>
<td>13</td>
<td>$1.85\times10^{-8}$</td>
<td>$0.00\times10^0$</td>
<td>21</td>
<td>0.8</td>
<td>$1.06\times10^{-12}$</td>
<td>$1.06\times10^{-12}$</td>
</tr>
<tr>
<td>0.8</td>
<td>15</td>
<td>$1.10\times10^{-6}$</td>
<td>$0.00\times10^0$</td>
<td>21</td>
<td>5</td>
<td>$1.60\times10^{-14}$</td>
<td>$1.58\times10^{-14}$</td>
</tr>
<tr>
<td>0.8</td>
<td>18</td>
<td>$1.29\times10^{-4}$</td>
<td>$0.00\times10^0$</td>
<td>21</td>
<td>10</td>
<td>$1.94\times10^{-15}$</td>
<td>$0.00\times10^0$</td>
</tr>
<tr>
<td>8.2</td>
<td>0.8</td>
<td>$1.00\times10^{-13}$</td>
<td>$6.99\times10^{-15}$</td>
<td>21</td>
<td>13</td>
<td>$0.00\times10^0$</td>
<td>$5.09\times10^{-15}$</td>
</tr>
<tr>
<td>8.2</td>
<td>5</td>
<td>$1.17\times10^{-8}$</td>
<td>$1.24\times10^{-12}$</td>
<td>21</td>
<td>15</td>
<td>$0.00\times10^0$</td>
<td>$2.11\times10^{-14}$</td>
</tr>
<tr>
<td>8.2</td>
<td>10</td>
<td>$4.14\times10^{-10}$</td>
<td>$1.01\times10^{-11}$</td>
<td>21</td>
<td>18</td>
<td>$1.62\times10^{-9}$</td>
<td>$8.62\times10^{-14}$</td>
</tr>
</tbody>
</table>

### Fig. 7 Numerical $E_{\text{new}}$ of $\partial F(X,Y)/\partial X$

### Fig. 8 Numerical $E_{\text{new}}$ of $\partial^2 F(X,Y)/\partial X^2$

As a result, Fig. 7 and 8 show that evaluations of the first and second order derivatives of GF can also obtain a precision of $10^{-9}$ at least under $M_{\text{new}}$ presented earlier. Furthermore, one can rationally expect satisfactory calculation results of GF other derivatives due to the relationship of equations (6) (7) (8) (9) (10).

In addition, the computation time is also acceptable. Efficiency of the new method is compared with that of the rest three methods. From Table 3, one finds that $T_1$, $T_2$, $T_{\text{new}}$ have the same order of magnitude, which are much less than $T_{\text{pre}}$ derived from the direct integration. Except from $T_{\text{pre}}$, $T_{\text{new}}$ is more approximate to $T_2$. 

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Table 3 Comparison of the computational efforts between four methods

<table>
<thead>
<tr>
<th>Step Length</th>
<th>$T_1 / s$</th>
<th>$T_2 / s$</th>
<th>$T_{pre} / s$</th>
<th>$T_{new} / s$</th>
</tr>
</thead>
<tbody>
<tr>
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<tr>
<td>0.15</td>
<td>3.86</td>
<td>2.88</td>
<td>259.28</td>
<td>3.79</td>
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<tr>
<td>0.20</td>
<td>2.27</td>
<td>1.66</td>
<td>143.02</td>
<td>2.18</td>
</tr>
<tr>
<td>0.30</td>
<td>0.96</td>
<td>0.76</td>
<td>64.72</td>
<td>0.90</td>
</tr>
<tr>
<td>0.40</td>
<td>0.54</td>
<td>0.42</td>
<td>35.56</td>
<td>0.51</td>
</tr>
<tr>
<td>0.50</td>
<td>0.36</td>
<td>0.26</td>
<td>22.80</td>
<td>0.35</td>
</tr>
<tr>
<td>0.60</td>
<td>0.25</td>
<td>0.20</td>
<td>15.87</td>
<td>0.23</td>
</tr>
<tr>
<td>0.70</td>
<td>0.17</td>
<td>0.14</td>
<td>11.72</td>
<td>0.16</td>
</tr>
<tr>
<td>0.80</td>
<td>0.15</td>
<td>0.11</td>
<td>8.95</td>
<td>0.14</td>
</tr>
</tbody>
</table>

Fig. 9 Change trends of GF and its derivatives when $X$ and $Y$ is up to 40
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Fig. 9 shows that along the X direction, which is also the propagation direction of the surface wave, the variation trend generally meets the cyclical variation law with continued amplitude attenuation, and that along the Y direction, which is the increasing direction of water depth, wave amplitude sharply attenuates to zero. The above law accords with the actual physical phenomenon.

4.4 Application to a floating hemisphere

![Fig. 10 The wetted surface of the hemisphere](image)

![Fig. 11 Different location of X and Y of the assumed situation](image)

**Table 4** Comparison of the hemisphere results

<table>
<thead>
<tr>
<th></th>
<th>X</th>
<th>Y</th>
<th>$\partial F(X,Y)/\partial X_{new}$</th>
<th>$\partial F(X,Y)/\partial X_{pre}$</th>
<th>$\partial^2 F(X,Y)/\partial X_{new}^2$</th>
<th>$\partial^2 F(X,Y)/\partial X_{pre}^2$</th>
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<tr>
<td>D2</td>
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</tbody>
</table>

The above-described numerical method of evaluating GF is tested here through a floating hemisphere with radius of 25 meters (see Fig. 10). The wetted surface of this hemisphere is represented by an ensemble of connected four-sided panels. A panel degenerates to a triangle when the coordinates of two vertices coincide. Total number of panel is 1008. The centroid of each panel is used as field point or source point. Here one investigates GF between a fixed field point located at (-19.71, -0.98, 2.93) and all the points regarded as unit sources. So 1007 times calculations need to be solved except for the situation when the field point coincides with the source point. In addition, considering the range of the subdomains, it is reasonable to assume wave number equals to 0.8. Then the resulting $(X, Y)$ is outlined in Fig. 11.
One chooses four points from $XY$ plane in each subdomains to make the verification. Table 4 depicts these results obtained by $M_{\text{new}}$ and $M_{\text{pre}}$. As to the hemisphere, it can be concluded that evaluations of the first and second order derivatives of GF can obtain a precision of $10^{-9}$ at least from all the listed data with accuracy of 11 decimal digits.

5. Conclusion

This paper explored in detail the classical representation of free surface GF, composed of a semi-infinite integral involving a Bessel function and a Cauchy singularity. For the evaluation of the function itself and its high order derivatives, only four kinds of analytical series expansion was developed, and the involved special functions, especially the hypergeometric function were evaluated with high precision and efficiency. In addition, one put forward the refined subdomains of the whole domain. The numerical results showed that the new method could acquire a high degree of precision and an acceptable and practical efficiency.

REFERENCES


A STUDY ON THE ESTIMATION METHOD OF THE FORM FACTOR FOR A FULL-SCALE SHIP

UDC 629.5.015.2:629.5.016.7:629.5.018.72:519.6
Original scientific paper

Summary

In this study, a prediction method of the form factor for a full-scale ship is suggested to minimize the power prediction error from a small model ship. Numerical simulations were carried out at various Reynolds numbers from a small model to a full-scale ship. The variation of the form factors was investigated from the results of the numerical simulation according to the Reynolds numbers. In addition, the results from the numerical simulations and experimental data of the geosim models were utilized to drive the correlation line and predict the form factor of a full-scale ship. The correlation line was applied to predict the effective power and the delivered power of a full-scale ship. As a result, the developed prediction method confirmed the possibility of predicting the power reliably from experiments using a small model.

Key words: Reynolds number; Form factor; Effective power; Delivered power; Geosim models

1. Introduction

Model tests in towing tanks are indispensable for predicting the resistance or the effective power and delivered power of a full-scale ship. In the model tests, the models are towed at speeds scaled to coincide with the Froude numbers of a full-scale ship at the given speeds. Therefore, the Reynolds numbers of the model ship and full-scale ship are different; hence, the frictional resistance and form factor are different. Van et al. (2011) reported that the residual resistance coefficients decrease as the size of the model ship increases, whereas form factors become larger.

The pressure coefficients recovery for the after part becomes faster as the Reynolds number is increased. As a result, the residual resistance coefficient decreases. And Choi et al. (2011), García-Gómez (2000) and Min and Kang (2010) also reported that the form factors increase with increasing Reynolds number.

The effective wake fraction is also decreased because the axial velocity on the propeller plane increases with increasing Reynolds number. All the results suggest that use of larger
models is advantageous for reliable prediction of the effective power or delivered power of a full-scale ship. In a relative small towing tank, however, the size of the model is limited ($Re=10^6$-$10^7$) and a reliable prediction of the effective power or delivered power of a full-scale ship through model tests are not always easy.

Moreover, the scale ratios of a model ship and full-scale ship increase with decreasing size of a ship. In addition, model tests are performed at low Reynolds numbers because the speed of a ship decreases with a model ship size by the Froude scaling. In other words, reasonable prediction of the effective power and delivered power are difficult due to the scale effects.

A low-speed blunt ship gradually appears by the down speeding and size-up of a ship. Generally, the low-speed blunt model ship is large with an intense bilge vortex for the after part. The bilge vortex is caused by the flow field of the hook shape that is separated from bilge of a ship. The bilge vortex is related to the thickness of the boundary layer.

In the ITTC 1978 method, the form factors are kept constant and show small differences in the Reynolds numbers, inevitably showing a difference between the model and full-scale ship. Min and Kang (2010) proposed a method to remedy these problems using only the results of model tests in the range of $Re=10^6$-$10^7$. They adopted a hyperbolic function defined as the ratio between the form factors of a full-scale ship and the model and the Reynolds numbers. On the other hand, the method did not take the differences in the hull forms into consideration and the predicted powers for high speed ships, such as KCS are prone to substantial error. The procedure reported by Ha et al. (2013) is similar to Min and Kang (2010) method except that they adopted an exponential function. This method, however, was devised to take the differences in hull forms among low, middle and high speed ships into consideration, resulting in better prediction for high speed hull forms than those obtained using Min and Kang’s method. In the present study, KVLCC2, KLNG and KCS were selected as the typical hull forms of low, middle and high-speed vessels, respectively. To estimate the dependency of the form factor on the Reynolds number, the form factors of three hull forms were computed numerically at several Reynolds numbers corresponding to the model and the full-scale ship. The form factors found from a numerical computation were combined with those from the tests of the geosim model and went through regression analysis to construct rational correlation lines for model-ship extrapolation. The lines are applied to the power prediction of the target ships and yield marked improvement.

2. Reference Hull Form

The principal dimensions of the ships, KVLCC2, KLNG and KCS are open to the public and can be found in Ha et al. (2013) or elsewhere. Table 1 shows lists the design speeds of the ships and various scale ratios taken for the model tests and numerical simulations.

<table>
<thead>
<tr>
<th>Ship name</th>
<th>KVLCC2</th>
<th>KLNG</th>
<th>KCS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design speed</td>
<td>15.5Knot</td>
<td>19.5Knot</td>
<td>24.0Knot</td>
</tr>
<tr>
<td>Scale ratios for model tests</td>
<td>1/160, 1/100, 1/58</td>
<td>1/103, 1/69, 1/34</td>
<td>1/125, 1/97, 1/63.2, 1/31.6</td>
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<tr>
<td>Scale ratios for numerical simulations</td>
<td>1/160, 1/100, 1/58, 1/29, 1/10, 1/5, full-scale</td>
<td>1/103, 1/69, 1/34, 1/10, 1/5, full-scale</td>
<td>1/160, 1/97, 1/63.2, 1/31.6, 1/21, 1/10, 1/5, full-scale</td>
</tr>
</tbody>
</table>
3. Power Prediction Method for A Full-Scale Ship

Three-dimensional extrapolation method recommended by ITTC are generally used to predict the effective power of a full-scale ship by a model test. Fig. 1 explains the power prediction of a full-scale ship. This method is based on Froude’s law and the wave-making resistance coefficient and form factor of a full-scale ship, and its model ship are assumed to be the same.

Fig. 1 Power prediction of a full-scale ship by ITTC

Fig. 2 compares the pressure contours for the model ships and full-scale ships on the side profile of the reference hull forms by the numerical simulations. The numerical simulation conditions are explained at validation part of the numerical simulation in this paper. The pressure contours of the reference hull forms are similar for the front parts of the model ships and full-scale ships. On the other hand, the positive pressure contours near the sterns of the reference hull forms grow on the full-scale ship. Consequently, the pressure coefficient of the full-scale ship is smaller than the pressure coefficient of the model ship. In other words, the residual resistance coefficients increase with decreasing Reynolds number. Ha et al. (2012) and Van et al. (2011) clearly showed that the residual resistance coefficients increase with decreasing model ship size according to the geosim model about the reference hull forms tests. Therefore, to predict the effective power of a full-scale ship by a model test, two-dimensional extrapolation methods recommended by ITTC is difficult from the small model ship. The two-dimensional extrapolation method is based on Froude's law, and the residual resistance coefficient of a full-scale ship and its model ship are assumed to be the same. On the other hand, the wave resistance coefficients of a full-scale ship and model ship are similar because the Froude numbers of a full-scale ship and model ship are the same. As shown in Fig.3, the wave profiles of a full-scale ship and model ship of the Series 60 CB=0.6 by the numerical calculation are similar. Therefore, the wave resistance coefficient is the dominant value by the Froude number.
Fig. 2 Comparisons of the pressure contours on the hull surfaces

Fig. 3 Wave profile of the Series 60 $C_a=0.6$ (Raven et al., 2010)

Therefore, the three-dimensional method (ITTC 1978 method) takes the Reynolds number dependency of the resistance components into account in the extrapolating procedure by introducing the invariant form factors, which are generally believed to yield better results, and were used in this study. On the other hand, the form factor is a function of the Reynolds number and the values for a full-scale ship and the model should be different. Therefore, to predict the effective power of a full-scale ship, the ITTC 1978 method needs to improve because the ITTC 1978 method does not consider the difference of the form factor of the full-scale ship and model ship. As shown in Fig. 1, the prediction of the effective wake of a full-scale ship is related to the form factor, so the predicted delivered power of a full-scale ship is also needed. Fig. 4 shows the axial velocity contours on the propeller plane of the reference hull forms according to the Reynolds numbers by the model test. where, The IUTT is model test results in Inha university towing tank and other model tests results is reference data by Yang et al.(2010) and Van et al. (2011). The axial velocity accelerated notably with increasing model size and the hook shape in the axial velocity contour decreased with increasing model size. Hence, the prediction result of the effective wake of a ship and the
prediction result of a nominal wake of a ship are different because the velocity contours on the propeller plane differ according to the model ship size. Therefore, the prediction delivered power of a full-scale ship needs to consider the difference of the form factors of a model ship and full-scale ship.

García-Gómez (2000) and Min and Kang (2010) examined the Reynolds number dependency of the form factor and proposed methods to estimate the form factor at a full-scale from the results of model experiments. Ha et al. (2013) also estimated the total resistance coefficient of the reference hull forms in the model scales using the two methods. The method suggested by García-Gómez (2000), however, yielded larger discrepancies in the predicted total resistance coefficients if the model sizes were altered because a change in the form factor due to the variation of the Reynolds number was not considered sufficiently. The method suggested by Min and Kang (2010) produced a similar prediction of the effective power of a full-scale ship to the method suggested by García-Gómez (2000). The predicted effective powers of the full-scale ship were still different. Because the method suggested by Min and Kang (2010) did not consider the difference according to the hull forms, the model test results by Min and Kang (2010) showed considerable scattering of the form factors. Therefore, the reasonable prediction method for the effective power and delivered power of a full-scale ship is needed; hence, a study of the precise prediction form factor of a full-scale ship is needed.
4. **New Power Prediction Method for A Full-Scale Ship**

4.1 Validation of the Numerical Simulation

In this study, the numerical simulations were performed using commercial code, FLUENT. The governing equations are the continuity equation and Navier-Stokes equations. Navier-Stokes equations were solved by SIMPLE (Semi-implicit Method for Pressure Linked Equation). The second order upwind difference method was used for spatial discretization in convection term. The VOF (Volume of Fluid) method was applied to define the location of the free surface. When careful examinations of the grid generation and decision of the turbulence model are needed, the numerical simulations at high Reynolds numbers are very important for deriving similar numerical simulation results to the real flow phenomena. In the case of a numerical calculation of a full-scale ship, the numerical simulation results are compared with the model ship test results because of the difficult obtainment of sea trial data and measurement data of a full-scale ship. Fig. 5 shows the typical grid system regarding KVLCC2 for the numerical simulation, and Table 2 lists the number of grids and the size of y+.

![Grid system of KVLCC2](image)

**Table 2 Grid information of the numerical simulation**

<table>
<thead>
<tr>
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</thead>
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<tr>
<td>Number of cells</td>
<td>1.6 x 10^5</td>
<td></td>
</tr>
<tr>
<td>y+</td>
<td>900</td>
<td>40</td>
</tr>
</tbody>
</table>

The number of grids of a full-scale ship was obtained from Choi and Kim (2010). In the study by Choi and Kim (2010), the numerical simulations regarding the pipe, flat plate and axis of symmetry were performed to survey the used y+, and the change in the numerical values were studied according to the increase in y+. In addition, Choi and Kim (2010) suggested an equation for calculating y+ above Re=10^7. In this study, the equation suggested
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by Choi and Kim (2010) was used to derive $y+$ for the numerical simulation of a full-scale ship. The numerical simulations for validation regarding the turbulence model in this study were performed on a KVLCC2 1/58 scale model because the KVLCC2 is dominated by the viscous effect and the design speed of the KVLCC2 is the slowest of the reference hull forms. In addition, the numerical simulation results were compared with the model test results by the KRISO (Korea Research Institute of Ships & Ocean Engineering). First of all, the flow fields of the numerical simulation results for decision of the proper turbulence model were compared with the results reported by Kim and Kim (2006) and Yang et al. (2010). In these numerical simulations, the turbulence models used were the Realizable k-ε and Reynolds stress models. The reasonable numerical simulations of the flow field of the after part as well as the free surface were performed because the numerical simulations in this study contained the free surface. On the other hand, the numerical simulation results reported by Kim and Kim (2006) and Yang et al. (2010) were both obtained using the double model.

Therefore, these numerical simulation results did not contain the free surface. The numerical results by the Reynolds stress model showed similar model test results to the KRISO reported by Kim and Kim (2006) and Yang et al. (2010). Fig. 6 shows the numerical simulation results by the Realizable k-ε and Reynolds stress models. In addition, similar to the studies reported by Kim and Kim (2006) and Yang et al. (2010), the numerical simulation results by the Reynolds stress model with the free surface showed similar model test results to the KRISO in this study. It is well-known that the Reynolds stress model has a degree of high order for the Realizable k-ε model. Therefore, it can be found that the size of hook shape by Reynolds stress model was relatively larger than the results by the Realizable k-ε model. Fig.
7 shows the wave contours and wave height on the free surface. The numerical simulation results of the wave contours in this study were slightly high for the model test results of the wave contours by KRISO, but these results were qualitatively similar. In addition, in the wave height, the numerical simulation results of the wave height was quantitatively similar to the model test results by KRISO. Therefore, the numerical simulation schemes in the present study are appropriate in the numerical simulation for the free surface, and the numerical simulation schemes in this study could be similarly predicted using the model test results.

![Wave Contours and Wave Height](image)

**Fig. 7** Comparisons of the free-surface wave contours and wave height

### 4.2 Power Prediction Method for a Full-Scale Ship

In this study, numerical simulations of the ship resistance were performed to examine the influence of the Reynolds number on the form factor. The estimated form factors from the numerical simulations were added to those from the model tests to suggest a rational prediction method for predicting the form factor on a full-scale because the model tests were performed at a low Reynolds number, typically in the range of $Re=10^6$-$10^7$. Therefore, in the model test for the transient region, the model tests could be reasonably performed by inducing the turbulence flows, because laminar and turbulence flows coexist. Generally, studs are used for turbulence stimulations. In this study, however, turbulence stimulators, such as studs and wires were installed near the 19.5 station to ensure reliable model test results. As recommended by ITTC, the form factor is determined using Prohaska’s method, which
assumes the form factors are a linear function a function of $F_n^4$. Fig. 8 compares the estimated form factors, plotted as a function of the model length, by model test and numerical simulation for the three test ships.

![Fig. 8 Comparisons of the form factors by the model test and numerical simulation](image)

Fig. 8 shows that the estimated form factor of the numerical simulations are generally in good agreement with those found from the model tests. In addition, both estimated form factors increase with increasing model length. Fig 9 shows the non-dimensional x-velocity contours at each stations about the 2m class model ships and full-scale ships of the reference hull forms, and these non-dimensional x-velocity contours are the results below the boundary layers. In the non-dimensional x-velocity contours about the model ships of the reference hull forms, the bilge vortex appeared from near the 1 station around KVLCC2 and KLNG of the middle-low speed vessels. But the bilge vortex was barely detectable in KCS of the high speed vessels. In non-dimensional x-velocity contours about the full-scale ships of the reference hull forms, the bilge vortex was barely observable for the model ships. On the other hand, a low speed region of the middle-low speed vessels such as KVLCC2 and KLNG appeared for KCS in the case of the full-scale ships. As shown in Fig. 9, the stern lengths of the KVLCC2 and KLSG area were shorter than the stern length of the KCS, and the water plane shapes at the loaded water line were different, respectively. In other words, the appearance locations of the bilge vortex were similar in the middle-low speed vessels such as KVLCC and KLNG model ships, but the bilge vortex was barely observable around the KCS model. The flow phenomena around the KVLCC2 and KLNG ships were similar but the phenomena around the KCS ship was different. Therefore, the form factors as a function of the Reynolds number of the KVLCC2 and KLNG could be similar, and the rising trend of the form factors could be both similar as the Reynolds number increases. The form factors of the KCS could be relatively lower than the form factors of the middle-low speed vessels, and the increasing trend of the form factors of the KCS could be different for the middle-low speed vessels. The study by Min and Kang (2010) was suggested to estimate the form factor of a full-scale ship, but this study did not consider the differences in the hull forms. On the other hand, the middle-low speed ships and high speed ships were classified according to the differences in the hull forms in this study.
Fig. 9 Comparisons of the x-velocity according to the station

Fig. 10 Variation of the numerically estimated form factors with the Reynolds number

Fig. 10 shows the variation of the form factors estimated by the numerical computations with the Reynolds number. The form factors increase rapidly with increasing Reynolds number in the range of $Re=10^6$~$10^7$. Moreover, the tendency of the estimated form factors for KVLCC2 and KLNG almost coincide with each other but that of KCS is separate from the other. Therefore, the tendency of form factors for middle-low speed ships is quite different with high speed ships. The form factor increases with increasing Reynolds number for the entire simulation range. Furthermore, as the Reynolds number approaches the full-scale value,
the estimated form factor appears to converge to certain values. Accordingly, when predicting the effective power and delivered power of full-scale ship from the model test in Re=10^6~10^7, the variation of the form factor with the Reynolds number should be considered.

Fig. 11 shows the model test and numerical simulation results by this study along with the model test results and the form factors estimation band of the full-scale ships reported by Min and Kang (2010) for validation of the numerical simulation results of the full-scale ships. The numerical simulation results of the full-scale ships by this study were contained in the form factors estimation band of the full-scale ships reported by Min and Kang (2010). Therefore, the numerical simulation results of the full-scale ships by this study could be used to improve the estimation method of a ship form factor. Min and Kang (2010) assumed that the form factor increases with increasing Reynolds number, and this form factor could converge a constant value to above the Re=10^9, and is called the terminal form factor. The form factors converged to a constant value as the Reynolds number increases, shown in Fig. 10 and 11. Min and Kang (2010) performed regression analysis using a relative value, i.e., the ratio of the form factor with respect to the terminal form factor, and this the regression analysis result was called the form factor correction factor ((1+k)_M/(1+k)_∞). The form factor correction factor converged as the Reynolds number increased, and was 1.0 when the Reynolds number was greater than 10^9. Ha et al. (2013) referred to the concepts by Min and Kang (2010).

Fig. 12 compares correlation lines of the form factors by Min and Kang (2010) and Ha et al. (2013). The correlation line of Min and Kang (2010) was applied as a hyperbolic function, and the correlation line of Ha et al. (2013) was applied as an exponential function. Unlike the correlation line of Min and Kang (2010), the correlation line of Ha et al. (2013) can be classified into two lines. On the other hand, the form factor of a full-scale ship is difficult to estimate from all of the correlation lines because the model test data are nonexistent in the region of Re=10^8~10^9 (region of a full-scale ship). In this study, the numerical simulation data in Fig. 10 and 11 were used in regression analysis. Regression analyses for an improvement of the method reported by Ha et al. (2013) were performed according to the assumption and concept by Min and Kang (2013) and the correlation line of form factors by Ha et al. (2013).
The correlation lines of the form factors were suggested with each middle-low speed ships and high speed ships based on the design speed 20 knots. In this study, the model test data by García-Gómez (2000) and Min and Kang (2010) and the numerical simulation and model test data by this study were used in regression analysis. The regression analysis results by Ha et al. (2013) were applied to the exponential function but only the exponential function applied. In contrast, various functions were applied in the present study. The applied typical functions are Eqs. 1~4 in this study. The hyperbolic function of Eq. 1 was referred in a study reported by Min and Kang (2010).

\[
\frac{(1+k)_M}{(1+k)_c} = \tanh x, \quad x = a(\log Re) \quad (1)
\]

\[
\frac{(1+k)_M}{(1+k)_c} = a(\log Re)^3 + b \quad (2)
\]

\[
\frac{(1+k)_M}{(1+k)_c} = a + b \cdot x + c \cdot x^2 \cdots A x^n \quad (3)
\]

\[
\frac{(1+k)_M}{(1+k)_c} = a \cdot e^{b(x+c)} + d \quad (4)
\]

Regression analyses were performed. The RMSE (Root Mean Square Error) was smallest in the case of the exponential function. Fig. 13 shows the correlation lines for the form factors, which was constructed from the results of the model test and numerical simulations. The figures show that the correlation line of the high speed vessels such as KCS behaves quite differently from the line for the middle-low speed vessels such as KVLCC2 and KLNG. Therefore, Eq. 5, which is based on an exponential curve and can predict the form factor of a full-scale ship, was used to estimate the correlation lines of the form factors.
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form factor for a full-scale ship
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\[
\frac{(1+k)_M}{(1+k)_c} = a \cdot e^{b(1+c)} + d \cdot x = \text{Log} \text{Re}
\]

\[a = -1.029, \ b = -0.828, \ c = -3.106, \ d = 0.940 \text{ for middle-low speed ship}
\]
\[a = -10.780, \ b = -0.722, \ c = 0.000, \ d = 1.000 \text{ for high speed ship}
\]

![Middle and low speed ship](image1)

![High speed ship](image2)

**Fig. 13** New correlation lines of the form factors

### 4.3 Comparison of the Predicted Results from Four Methods

The form factors, effective power and delivered power of a full-scale ship was predicted using four different methods, i.e., ITTC 1978 method, Min and Kang (2010), Ha et al., (2013) and the present method. Fig. 14 shows the discrepancy in the result of the predicted effective powers, delivered powers and form factors of full-scale ships using the four different methods. Compared to the form factors of the full-scale using the four different methods, the estimated form factors of the full-scale ships by ITTC’ 1978 method differed according to the difference in the Reynolds number due to the model ship size. The estimated form factors of the full-scale ships by Ha et al. (2013) differed according to the difference in the Reynolds number with the model ship sizes. On the other hand, the estimated form factors of the full-scale ships by Min and Kang (2010) and the present study were similar regardless of the Reynolds number according to the model ship size. On the other hand, the predicted effective power of the full-scale ships by Min and Kang (2010) were most different because the correlation line of Min and Kang (2010) was not classified as difference in the hull forms. The predicted the effective power of the full-scale ships by this study were most similar because the correlation line of this study were classified as the differences in the hull forms used the numerical simulation results of the various Reynolds numbers from the model ship sizes to the full-scale ship. In the method reported by Ha et al. (2013), the estimated effective
powers were more similar than the results estimated by the ITTC' 1978 method and Min and Kang (2010). The effective wake of a full-scale ship is needed to predict the delivered power. The prediction method of the effective wake of a full-scale ship is expressed in Eq. 6, and the form factor of the model ship and full-scale ship were applied to predict the effective wake of a full-scale ship.

\[ W_S = (t + W_R) + (W_M - t - W_R) \left( \frac{(1 + k_S)C_{FS} + W}{(1 + k_M)k_{FM}} \right) \]

where, \( W_R \) means a rudder effect. The same form factors of a model ship and full-scale ship were applied to predict the effective wake of a full-scale ship using the ITTC' 1978 method. On the other hand, the axial velocity contours differed according to the model ship sizes in Fig. 4. So the different form factors of a model ship and full-scale ship would be applied to predict the effective wake of a full-scale ship. In other words, the difference in form factors by the Reynolds number were considered for the prediction of the delivered power. The predicted delivered power of the full-scale ship by the ITTC' 1978 method was significantly different with KCS, and the predicted delivered powers of the full-scale ships by Min and Kang (2010) were the most different. The differences in the hull forms were not classified. In KVLCC2, the results of the predicted delivered power of the full-scale ship by Ha et al. (2013) are more different than that by the ITTC' 1978 method. On the other hand, in KCS, the results of the predicted delivered power of the full-scale ship by Ha et al. (2013) were more similar than that by the ITTC' 1978 method. In this study, the results of the predicted delivered powers of the full-scale ships were most similar because this study used the numerical simulation results of the various Reynolds number from the model ship sizes to the full-scale ship, and this study was classified according to the differences in the hull forms. In other words, the correlation lines of this study were proper for the form factor estimation of a full-scale ship. Consequently, the maximum discrepancies in the predicted effective power and delivered power of the full-scale ships occurring due to the changes in the model sizes appeared to be less than approximately 3%, which is far lower than those estimated using existing methods. In particular, this study was more advantageous for the power prediction of high speed hull forms, such as KCS, than the existing methods. Table 3 summarizes the maximum discrepancy in the results due to the variation of the model size. The figure and table show that, for all three test ships, the present method yielded the best results; the discrepancies in the predicted effective powers were confined.
Fig. 14 Comparisons of the form factors, effective powers and delivered powers of the full-scale ships predicted by four different methods
Table 3 Comparisons of the maximum discrepancy of the form factor, effective power and delivered power of the full-scale ships the among four different methods

<table>
<thead>
<tr>
<th></th>
<th>KVLCC2</th>
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<tbody>
<tr>
<td></td>
<td>k</td>
<td>EHP</td>
<td>DHP</td>
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<td>13.4</td>
</tr>
<tr>
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<td>6.4</td>
<td>3.5</td>
<td>5.2</td>
</tr>
<tr>
<td>Present method</td>
<td>0.7</td>
<td>2.6</td>
<td>1.7</td>
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<td></td>
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<tr>
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<td>1.9</td>
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</table>

5. Conclusions

The form factor of a ship during model tests and the same value can be used when extrapolating the results to full-scale. On the other hand, previous studies showed that the form factors vary with the Reynolds number and many attempts have been made to remedy the problem.

In the present study, a practical method was proposed to estimate the form factor of a full-scale ship taking the difference in hull forms into consideration by numerical simulations. The present method was applied to predict the power three test ships. The maximum discrepancies in the predicted effective power and delivered power of the full-scale ships occurring due to the changes in the model sizes appeared to be less than approximately 3%, which is far lower than those estimated by the existing methods. In particular, the present method is more advantageous for the power prediction of high speed hull forms, such as KCS, than the existing methods. The major reason is that this method reflects the difference in hull forms when the form factor of the full-scale ship is predicted. Another reason is that when the correlation line of the form factors is derived, the exponential function better reflects the ratio of the estimated form factor of a model and full-scale ship than the hyperbolic function suggested by Min and Kang(2010).

6. Acknowledgements

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A Study on the Estimation Method of the form factor for a full-scale ship

Young-Gill Lee, Yoon-Jin Ha
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REFERENCES


USE OF A HYBRID MCDM METHOD TO EVALUATE KEY SOLUTIONS INFLUENCING SERVICE QUALITY AT A PORT LOGISTICS CENTER IN TAIWAN

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Summary

The main purpose of this article is to develop a hybrid multiple criteria decision-making (MCDM) model to evaluate key solutions influencing service quality at a port logistics center in Taiwan. At first, this article proposes the use of a hybrid MCDM model incorporating the Analytic Hierarchy Process (AHP), Decision Making Trial and Evaluation Laboratory (DEMATEL), and Analytic Network Process (ANP) techniques to assess the causal relationships between criterion variables, including dependent relationships and feedback mechanisms. Then, using a hypothetical port logistics center case as an example, this study focuses port logistics center service quality solutions, and applies the hybrid MCDM model constructed in this article in order to explain its functioning and assessment procedures. Finally, via the various operating steps of the hybrid MCDM model, the key logistics center service quality solutions are ranked. Furthermore, some discussions and conclusion are provided in the last of this article.

Key words: multiple criteria decision-making (MCDM); Analytic Hierarchy Process (AHP); Decision Making Trial and Evaluation Laboratory (DEMATEL); Analytic Network Process (ANP); port logistics center

1. Introduction

Taiwan's economic activity chiefly revolves around international trade, and it is a typical foreign trade-dependent island country. In terms of volume and value, approximately 80% and 70% of Taiwan's international import/export cargo is shipped by means of the maritime transport chain [1], with the remaining portion being shipped by air. The relationship between international trade and Taiwan's economic development is consequently very close, and maritime transport plays an extremely important role in Taiwan's international trade and logistics activities.

In the maritime transport chain, ports are transshipment intermediaries between land and sea transport, and are also nodal points of cargo consolidation in the international trade
and the logistics system [2]. During the 1990s, Taiwan's Kaohsiung port made active efforts to boost added service value in the maritime transport market [3, 4], and also use more effective operating methods to ship international cargo to all parts of the globe. As a result, by providing efficient logistics services platforms [5] and an effective marine terminal [6], Kaohsiung port was one of the most important maritime transport logistics centers in the Asia-Pacific region for a period of time [1, 2, 7], and was the world's third largest container port in 1999. However, Taiwan currently faces severe competition from the cargo logistics markets of China, Northeast Asia, and Southeast Asia, and the surging cargo volume and geographical locations of ports in these regions are making them an ever-greater threat to Taiwan's ports. As a consequence, Kaohsiung port had fallen to only the world's 13th largest container port in 2017. But as far as port transportation functions are concerned, we believe that the soft power issue on which Taiwan's international ports must focus their attention is their ability to deliver cargo safely and smoothly to destinations around the world via effective international port logistics centers [4, 8], which will facilitate international trade activity.

With today's increasingly stringent product and service requirements, the provision of products and services has gradually shifted from a focus on production and supply to a focus on customers' needs. And with the appearance of various quality management concepts and an operating environment in which consumers place much greater emphasis on service quality, the service quality at international port logistics centers has become a key factor in customers' port selection decisions [4, 9, 10]. Because service quality is a matter of customers' subjective perceptions and assessments, logistics service quality will have a positive influence on a logistics center's perceived operating performance [11-13], and port logistics centers must therefore place emphasis on service quality. And because perceived service quality can be equated with the gap between a customer's service expectations and that customer's assessment after receiving service [14, 15], whether service quality addresses customers' real needs is an urgent question for service providers. In this context, Liang et al. [9] has already investigated what factors influence a port logistics center's service quality (the question of "what"), and how to boost service quality (the question of "how"). They use the Quality Functional Deployment (QFD) method to investigate how logistics center operators can convert customers' service needs to solutions able to enhance service quality, and thereby seek out concrete, feasible plans for improving service quality. It is hoped that the findings of Liang et al. [9] can provide guidance to port logistics center operators assessing service quality improvement methods.

The port logistics service industry [4, 9, 10] is a customer-oriented consumer market, and the ability to accurately determine customers' needs and quickly satisfy those needs is of immense assistance to companies' ability to expand their market share and boost profit. It is therefore very important for service providers [9, 10] to determine factors affecting service quality (the question of what), and, at the same time, when there is a gap between customers' perceived service quality performance and their expectations, it is also extremely important for companies to constantly propose solutions for raising service quality (the question of how).

Because there may be certain correlations or relationships [16] among criteria influencing service quality (the question of what), and also among solutions for boosting service quality (the question of how) (implying that these criteria are not necessarily mutually independent [17], but are instead interdependent and have a feedback relationship [1, 2, 9, 10]), assessment of the weights and performance values can be best performed employing the multiple criteria decision-making (MCDM) method [18-21]. This study consequently seeks to clarify the causal relationship between factors/solutions through an in-depth investigation and
analysis of the causal relationship between the what and how questions, and then convert the causal relationship between factors/solutions into a clear structural model [22, 23].

The Battelle Memorial Institute of Geneva proposed the Decision Making Trial and Evaluation Laboratory (DEMATEL) technique [24-26] as a means of implementing scientific and human affairs programs between 1972 and 1976. The DEMATEL technique can be effectively used to confirm and help decision-makers to understand the complex causal relationships between various factors that exist in structural model concerning relevant research topics. This technique involves the use of matrices and relevant mathematical theory to calculate the causal relationships and the intensity of the influence between factors. Previous studies have shown that the DEMATEL technique can be applied to many management issues and problems [27], such as marketing and logistics management, service quality management, performance evaluation, supply chain management, and supplier selection, etc.

The effective evaluation of the weights of criteria (clusters) and sub-criteria (elements) is an important issue in MCDM problems [28]. In general, the evaluated criteria and sub-criteria may be interdependent and have feedback relationships, in which case the decision structure cannot be built with a hierarchical structure, and a network structure is appropriate instead. The Analytic Network Process (ANP) technique [16] plays an important role in solving this type of MCDM problem, and is used to obtain the weights of all clusters and elements after considering the interdependence and feedback relationships among criteria, objectives, and elements.

This study therefore proposes the use of a hybrid multiple criteria decision-making model incorporating AHP, DEMATEL, and ANP to assess the degree of correlation between factors and solutions, the strength of their influence, and their relative level of importance. Nevertheless, in all types of enterprise diagnostics, it is necessary to ask what is the most important, why the problem occurred (the question of what) or how to resolve it (the question of how). This study tends to believe that the latter is more important. Because enterprises ultimately need solutions, the elimination of various factors that have a negative influence on enterprise performance can provide customers a greater degree of satisfaction. Based on this reasoning, this study focuses on port logistics center service quality solutions, and uses the hybrid MCDM model developed in this paper to illuminate the assessment process employed in this study.

This paper consists of four sections. Apart from the introduction in the current section, the second section introduces the research methods and theoretical basis used in this paper, and constructs a hybrid MCDM model, the third section uses a case study to explain the model's functioning. The final section presents discussions and conclusion.

2. Methodology

This section describes the construction of a hybrid MCDM model incorporating the AHP, DEMATEL, and ANP methods that can be applied effectively, and is scientific and systematic. The proposed hybrid MCDM method based on AHP, DEMATEL, and ANP concepts can be summarized as follows:

Step 1: Find related clusters and elements of the decision-making problem.
Step 2: Use the DEMATEL technique to construct a network framework.
Step 2.1: Calculate the initial direct-relation matrix.
Step 2.2: Calculate the normalized direct-relation matrix and derive the total direct/indirect matrix.
Step 2.3: Set a threshold value and obtain the threshold total direct/indirect matrix.

Step 2.4: Find the normalized threshold total direct/indirect matrix.

Step 2.5: Find the influence coefficients of cluster $C_i$ on cluster $C_j$ and build the network framework.

Step 3: Use the AHP technique to find the influence priority vector of a given set of elements in a cluster on any element in the same cluster or a different cluster.

Step 3.1: Establish pair-wise comparison matrices for all clusters and all elements versus related clusters.

Step 3.2: Consistency testing.

Step 3.3: Calculate the weights of $n_j$ elements in cluster $C_j$.

Step 4: Build an unweighted supermatrix.

Step 5: Obtain a weighted supermatrix.

Step 6: Find the limiting supermatrix.

To be easy to understand the proposed model, the flow chart of the hybrid MCDM method in this article can be shown as Fig. 1.

![Flow chart of the hybrid MCDM method](image_url)
2.1 Finding related clusters and elements in the decision problem

Based on the research topic, we first located related clusters and the elements in those clusters that could be used to resolve the problem. The clusters and elements were obtained from the academic literature and from consultation with managers and experts concerning the research topic. Here we supposed that the network consists \( n \) clusters (i.e., \( C_1, C_2, \ldots, C_n \)) and \( n_i \) elements (i.e., \( e_{11}, e_{12}, \ldots, e_{in} \)) versus related cluster \( C_j \).

2.2 Use of the DEMATEL technique to construct the network framework

The network framework can be constructed using the relationships and coefficients between clusters found by via the DEMATEL technique. The steps employed in the DEMATEL technique are explained as follows:

2.2.1 Calculation of the initial direct-relation matrix

Assume that \( p \) experts (i.e., \( E_1, E_2, \ldots, E_p \)) are asked to indicate the degree of direct influence of each cluster \( C_i \) on each cluster \( C_j \). In this study, we employed a scale in which 0, 1, 2, 3, and 4 represented the range from ‘no influence’ to ‘very high influence.’

Let \( b_{ij}^k \), \( k = 1, 2, \ldots, p \), \( \forall i, j = 1, 2, \ldots, n \), be the degree to which \( C_i \) affects \( C_j \) given by respondent \( E_k \). The separated initial direct-relation matrix \( B^k \) given by expert \( E_k \) is therefore

\[
B^k = (b_{ij}^k)_{n \times n},
\]

where \( b_{ij}^k = 0 \), \( \forall i = j \).

The initial direct-relation matrix \( B \) can then be constructed as follows:

\[
B = (b_{ij})_{n \times n},
\]

where \( b_{ij} = 0 \), \( \forall i = j \); and \( b_{ij} = \sum_{k=1}^{p} b_{ij}^k / p \), \( \forall i \neq j \).

2.2.2 Calculation of the normalized direct-relation matrix and derivation of the total direct/indirect matrix

Letting \( \lambda = \max \{ \max_i (\sum_j b_{ij}), \max_j (\sum_i b_{ij}) \} \), the normalized direct-relation matrix \( X \) can then be obtained:

\[
X = (1 / \lambda) \times B
\]

When \( \lim_{t \to \infty} \sum_{i} X^t = (0)_{n \times n} \), the total direct/indirect matrix \( T \) can now be obtained.

\[
T = X(1 - X)^{-1}
\]

2.2.3 Setting a threshold value and obtaining the threshold total direct/indirect matrix

In order to reduce the complexity of the causal relationships between clusters, decision-makers can set a threshold value for the influence level. When an influence value in the total direct/indirect matrix \( T \) is higher than the threshold value, it can be chosen and converted to a
threshold total direct/indirect matrix. There are many methods that can be used to set the threshold value [29, 30]. In this paper, the scree test [31], derived by plotting the threshold values against the number of influences in their order of extraction, is used to determine the number of influences based on the threshold value considered. When a threshold value has been determined, the threshold total direct/indirect matrix can be identified.

Assume that total direct/indirect matrix $T = (t_{ij})_{n \times n}$. Let $\beta$ be the threshold value. The threshold total direct/indirect matrix $T^\beta$ can now be obtained:

$$T^\beta = (t_{ij}^\beta)_{n \times n}$$ (5)

where $t_{ij}^\beta = \begin{cases} t_{ij}, & \text{if } t_{ij} > \beta \\ 0, & \text{if } t_{ij} \leq \beta \end{cases}$

2.2.4 Finding the normalized threshold total direct/indirect matrix

The normalized threshold total direct/indirect matrix $T^N$ can be obtained as follows:

$$T^N = (t_{ij}^N)_{n \times n}$$ (6)

where $t_{ij}^N = t_{ij}^\beta / S_i$, $S_i = \sum_{j=1}^{n} t_{ij}^\beta$.

2.2.5 Finding the influence coefficients of cluster $C_i$ on cluster $C_j$ and building the network framework

The influence coefficients $f_{ij}$ of cluster $C_i$ on cluster $C_j$ can now be obtained based on the results shown in Section 2.2.4. That is, $f_{ij} = t_{ij}^N$.

This completes the construction of a network framework derived from the research topic on the basis of influencing relationships and the coefficients between clusters.

2.3 Use of the AHP technique to find influence priority vectors

The influences of a given set of elements in a cluster on any one element in the network can be determined using a priority vector derived from pair-wise comparison matrices using AHP technique. The steps employed in the AHP technique [1, 17] are summarized as follows:

2.3.1 Establishment of pair-wise comparison matrices for all clusters and all elements versus related clusters

In this study, the AHP questionnaires were designed based on the network framework. The linguistic values shown in Table 1 were used to evaluate the relative influence (importance) of a given set of elements in a cluster on any element in the same cluster or a different cluster.

Assume that there are $p$ experts ($i.e., E_1, E_2, \ldots, E_p$) in a committee. These experts are responsible for assessing the relative importance of a given $n_j$ elements ($i.e., e_{ij1}, e_{ij2}, \ldots, e_{ijn_j}$) in cluster $C_j$ on any element in the same cluster or a different cluster using the basic scale shown in Table 1.

Letting $a_{ns}^k$ and $k = 1, 2, \ldots, p, \ \forall t, s = 1, 2, \ldots, n_j$ is the relative importance of element $e_s$ to $e_t$ in cluster $C_j$ given by expert $E_k$ after considering the influence on any element in
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the same cluster or a different cluster. The pair-wise comparison matrix $A^k$ of the relative importance of elements $e_j$ and $e_s$ given by expert $E_k$ can be constructed as follows:

$$A^k = [a^k_{ts}]_{n_j \times n_j},$$

where $a^k_{ts} = 1, \; \forall t = s$; and $a^k_{ts} = 1/a^k_{st}, \; \forall t \neq s$.

Table 1 Basic scale used in the AHP technique

<table>
<thead>
<tr>
<th>Intensity of importance</th>
<th>Definition</th>
<th>Explanation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Equal importance</td>
<td>Two activities contribute equally to the objective</td>
</tr>
<tr>
<td>3</td>
<td>Weak importance of one over another</td>
<td>Experience and judgment slightly favor one activity over another</td>
</tr>
<tr>
<td>5</td>
<td>Essential or strong importance</td>
<td>Experience and judgment strongly favor one activity over another</td>
</tr>
<tr>
<td>7</td>
<td>Very strong or demonstrated importance</td>
<td>An activity is favored very strongly over another; its dominance can be demonstrated in practice</td>
</tr>
<tr>
<td>9</td>
<td>Absolute importance</td>
<td>The evidence favoring one activity over another is of the highest possible order of affirmation</td>
</tr>
<tr>
<td>2, 4, 6, 8</td>
<td>Intermediate values between adjacent scale values</td>
<td>When compromise is needed</td>
</tr>
</tbody>
</table>

Source: [17]

2.3.2 Consistency testing

Consistency testing is an important element of the AHP technique [17], and can be performed using the consistency ratio (C.R.), which is defined as:

$$C.R. = \frac{C.I.}{R.I.}.$$ (8)

where $C.I.$ and $R.I.$ denote the consistency index and random index. And

$$C.I. = \frac{\lambda_{max}^k - n_j}{n_j - 1}$$ (9)

where $n_j$ is the number of elements compared, and $\lambda_{max}^k$ is the eigenvalue of the pair-wise comparison matrix $A^k = [a^k_{ts}]_{n_j \times n_j}$.

Here $\lambda_{max}^k$ is calculated by the following steps:

(1) Calculate the weight $w^k_i$ of element $e_i$.

$$w^k_i = \left(\sum_{j=1}^{n_j} (a^k_{ti} / \sum_{r=1}^{n_j} a^k_{tr})) / n_j, \; \; t = 1, 2, ..., n_j; \; k = 1, 2, ..., p.$$ (10)

(2) Calculate the eigenvalue $\lambda_{max}^k$ of pair-wise comparison matrix $A^k = [a^k_{ts}]_{n_j \times n_j}$.

$$\lambda_{max}^k = (1/n_j)(\sum_{i=1}^{n_j} \sum_{j=1}^{n_j} a^k_{ts} w^k_i / w^k_i))$$ (11)
where \( w_i^k \) is the weight of element \( e_{ij} \).

The \( R.I. \) value can be obtained from Table 2. When the \( C.R. \) value is less than or equal to 0.1, the consistency test is successful [17].

Table 2  Random index

<table>
<thead>
<tr>
<th>n</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>R.I.</td>
<td>0.00</td>
<td>0.00</td>
<td>0.58</td>
<td>0.90</td>
<td>1.12</td>
<td>1.24</td>
<td>1.32</td>
</tr>
</tbody>
</table>

Source: [17]

2.3.3  Calculate the weights of \( n_j \) elements in cluster \( C_j \)

Let there be \( q \leq p \) experts whose evaluation results pass the consistency test. Let \( m_{r_i}^j \), \( r = 1, 2, \ldots, q; \forall t, s \in 1, 2, \ldots, n_j \), be the relative importance of element \( e_{ij} \) to \( e_{js} \) given by expert \( E_t \). The pair-wise comparison matrix \( M \) of the relative importance of all elements given by all \( q \) experts can now be constructed as follows:

\[
M = [m_{r_i}^j]_{n_j \times n_j},
\]

where \( m_{r_i}^j = (\prod_{r=1}^{q} m_{r_i}^j)^{1/q} \), if \( t < s \); \( m_{r_i}^i = 1, \forall t = s \); and \( m_{r_i}^j = 1/m_{r_i}^j, \forall t \neq s \).

Let \( w = (w_1, w_2, \ldots, w_s, \ldots, w_n) \) be the eigenvector of the pair-wise comparison matrix \( M = [m_{r_i}^j]_{n_j \times n_j} \). The influence weight \( w_i \) of element \( e_{ij} \) can now be obtained by

\[
w_i = \left( \sum_{s=1}^{n_j} \left( m_{r_i}^j \sum_{t=1}^{n_j} m_{r_i}^t \right) / n_j \right), \quad t = 1, 2, \ldots, n_j.
\]

2.4  Building an unweighted supermatrix

The general form of an unweighted supermatrix can be represented as follows:

\[
W = (W_{ij})_{n \times n},
\]

where \( W_{ij} \) is called a "block" of the unweighted supermatrix, and is a matrix of the form:

\[
W_{ij} = (w_{u_i}^v)_{n_i \times n_j}, \quad i, j = 1, 2, \ldots, n; \quad u = 1, 2, \ldots, n_i; \quad v = 1, 2, \ldots, n_j.
\]

The \( j^{th} \) column \( (w_{1j}^j, w_{2j}^j, \ldots, w_{n_j}^j)^T \) of \( W_{ij} \) is the eigenvector of the influence (importance) of \( n_j \) elements \( (e_{i1}, e_{i2}, \ldots, e_{in_j}) \) in cluster \( C_i \) on the \( j^{th} \) element \( e_{ij} \) in cluster \( C_j \). If the \( i^{th} \) cluster has no influence on the \( j^{th} \) cluster, then set \( W_{ij} = (0)_{n_i \times n_j} \).

2.5  Obtaining the weighted supermatrix

The weighted supermatrix \( W^w \) can be calculated:

\[
W^w = (W^w_{ij})_{n \times n},
\]

where
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\[ W_{ij}^w = (f_j \times w_{ij})_{n \times n}, \quad i, j = 1, 2, ..., n; \quad u = 1, 2, ..., n_j; \quad v = 1, 2, ..., n_i; \]  

and \( f_j \) is the influence coefficient of cluster \( C_j \) on cluster \( C_i \).

2.6 Finding the limiting supermatrix

The weighted supermatrix can be stabilized by multiplying the weighted supermatrix by itself until the values of rows in the supermatrix converge on the same value for each column of the supermatrix. The resulting stabilized supermatrix is the limiting supermatrix. The priority of all elements can be determined based on the limiting supermatrix.

3. Application of the hybrid MCDM model: An illustrative example

This section uses the hybrid MCDM model constructed in this paper to assess the degree of correlation between port logistics center service quality solutions, the strength of their influence, and their relative level of importance. This section employs a hypothetical case as an example to examine this paper's hybrid MCDM model, and includes a description of the problem and the various analytical procedures involved.

Step 1: Identify related clusters and elements

In order to assess whether the quality of its services was sufficient to meet customers’ true needs, a logistics center at Kaohsiung port in Taiwan sought to locate factors affecting the logistics center's service quality. The results of this assessment ultimately revealed that the quality of some services did not meet requirements. In order to convert customer service needs to concrete, feasible solutions for improving service quality, the Kaohsiung port logistics center operator performed analysis focusing on the three major aspects of convenience, professionalism, and service personnel, and drafted nine concrete, feasible solutions [9, 10, 32-38] with the content shown in Table 3. In addition, to ensure that subsequent analysis could be clearly expressed, these three major aspects were represented as three clusters, and the nine feasible solutions represented as nine elements.

Table 3 List of solutions influencing service quality for Kaohsiung port logistics center

<table>
<thead>
<tr>
<th>Aspects</th>
<th>Solutions</th>
<th>Descriptions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Convenience</td>
<td>Comprehensive information system ( (e_{11}) )</td>
<td>Has a 24-hour query system, provides answers to customer queries concerning cargo status at any time.</td>
</tr>
<tr>
<td></td>
<td>Single-window service ( (e_{12}) )</td>
<td>Single-window service, ensures that customers can obtain all services provided by the logistics center at a single counter.</td>
</tr>
<tr>
<td></td>
<td>Simplified administrative operating procedures ( (e_{13}) )</td>
<td>Simplification of logistics center's internal administrative procedures, accelerating implementation of operating procedures and reduction of customer waiting time.</td>
</tr>
<tr>
<td>Service personnel</td>
<td>Service attitude ( (e_{21}) )</td>
<td>The improvement of service attitude among personnel will enhance customers' satisfaction with service quality.</td>
</tr>
<tr>
<td></td>
<td>Professional skills ( (e_{22}) )</td>
<td>Enhancement of the professional skills of service personnel will enhance customers' satisfaction with service quality.</td>
</tr>
<tr>
<td></td>
<td>Responsiveness ( (e_{23}) )</td>
<td>Enhancement of the responsiveness of service personnel will enhance customers' satisfaction with service quality.</td>
</tr>
</tbody>
</table>
Table 3 List of solutions influencing service quality for Kaohsiung port logistics center (Continued)

<table>
<thead>
<tr>
<th>Aspects</th>
<th>Solutions</th>
<th>Descriptions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Professionalism $(C_3)$</td>
<td>Enhanced warehouse safety management $(e_{31})$</td>
<td>The logistics center bears responsibility for safety of cargo while in warehouse and consequently provides enhanced safety management, reducing theft, damage, and missing goods, and boosting the logistics center's service quality.</td>
</tr>
<tr>
<td>Safety of cargo during transport $(e_{32})$</td>
<td>The logistics center bears responsibility for maintaining cargo safety during transport.</td>
<td></td>
</tr>
<tr>
<td>Cargo tracking and management ability $(e_{33})$</td>
<td>The logistics center must be able to know the status of cargo at all times while in warehouse and during transport.</td>
<td></td>
</tr>
</tbody>
</table>

Note: The code names of each aspect and solution are shown in parentheses.

Step 2: Use of the DEMATEL technique to construct a network framework

**Step 2.1.** Five experts (three senior managers of international logistics companies with at least 20 years working experience, and two professors of international logistics field with 15 years teaching and practical experience) were asked to indicate the degree of direct influence of each cluster $C_i$ on each cluster $C_j$. The values 0, 1, 2, 3, and 4 employed in the scale in this example represent the range from 'no influence' to 'very high influence.'

Using equation (1), the separated initial direct-relation matrices given by the five experts were as follows:

$$B^1 = \begin{pmatrix} 0 & 4 & 3 \\ 3 & 0 & 2 \\ 4 & 3 & 0 \end{pmatrix}, \quad B^2 = \begin{pmatrix} 0 & 4 & 2 \\ 1 & 0 & 3 \\ 2 & 4 & 0 \end{pmatrix}, \quad B^3 = \begin{pmatrix} 0 & 4 & 2 \\ 2 & 0 & 3 \\ 2 & 4 & 0 \end{pmatrix}, \quad B^4 = \begin{pmatrix} 0 & 3 & 2 \\ 3 & 0 & 3 \\ 4 & 4 & 0 \end{pmatrix}, \quad B^5 = \begin{pmatrix} 0 & 3 & 3 \\ 3 & 0 & 4 \\ 4 & 4 & 0 \end{pmatrix}.$$  

Using equation (2), the initial direct-relation matrix $B$ was be constructed as

$$B = \begin{pmatrix} 0 & 3.6 & 2.4 \\ 2.4 & 0 & 3.0 \\ 3.2 & 3.8 & 0 \end{pmatrix}.$$  

**Step 2.2.** We took $\lambda = \max\{6, 5.4, 7, 5.6, 7.4, 5.4\} = 7.4$. The normalized direct-relation matrix $X$ could then be obtained using equation (3):

$$X = (1/7.4) \times \begin{pmatrix} 0 & 3.6 & 2.4 \\ 2.4 & 0 & 3.0 \\ 3.2 & 3.8 & 0 \end{pmatrix} = \begin{pmatrix} 0 & 0.486 & 0.324 \\ 0.324 & 0 & 0.405 \\ 0.432 & 0.514 & 0 \end{pmatrix}.$$  

Using equation (4), the total direct/indirect matrix $T$ was

$$T = X(I - X)^{-1} = \begin{pmatrix} 0 & 0.486 & 0.324 \\ 0.324 & 0 & 0.405 \\ 0.432 & 0.514 & 0 \end{pmatrix} \times \begin{pmatrix} 2.228 & 1.836 & 1.465 \\ 1.404 & 2.420 & 1.435 \\ 1.684 & 2.037 & 2.371 \end{pmatrix} = \begin{pmatrix} 1.228 & 1.836 & 1.465 \\ 1.404 & 1.420 & 1.435 \\ 1.684 & 2.037 & 1.371 \end{pmatrix}.$$  

**Step 2.3.** According to the scree test [31], the relationships between threshold value and the number of influence in matrix $T$ are depicted as Table 4, taking 1.41 as the threshold value which is obviously appropriate. That is, the threshold value $\beta = 1.41$.
Then, by using equation (5), the threshold total direct/indirect matrix $T^\beta$ was derived as:

$$T^\beta = \begin{pmatrix} 0 & 1.836 & 1.465 \\ 0 & 1.420 & 1.435 \\ 1.684 & 2.037 & 0 \end{pmatrix}. $$

**Step 2.4.** In the threshold total direct/indirect matrix shown above, the three row sums are:

$$S_1 = 0 + 1.836 + 1.465 = 3.301,$$
$$S_2 = 0 + 1.420 + 1.435 = 2.855,$$
$$S_3 = 1.684 + 2.037 = 3.721.$$

Furthermore, using equation (6), the normalized threshold total direct/indirect matrix $T^N$ was obtained as:

$$T^N = \begin{pmatrix} 0 & 0.556 & 0.444 \\ 0 & 0.497 & 0.503 \\ 0.453 & 0.547 & 0 \end{pmatrix}. $$

**Step 2.5.** Based on the normalized threshold total direct/indirect matrix $T^N$, the influence coefficients $f_{ij}$ of cluster $C_i$ on cluster $C_j$ were obtained as:

$f_{11} = 0$, $f_{12} = 0.556$, $f_{13} = 0.444; $
$f_{21} = 0$, $f_{22} = 0.497$, $f_{23} = 0.503; $
$f_{31} = 0.453$, $f_{32} = 0.547$, $f_{33} = 0.$

Step 3: Use of the AHP technique to evaluate relative importance

**Step 3.1.** Three experts (two senior managers of international logistics companies with at least 20 years working experience, and one professor of international logistics field with 15 years teaching and practical experience) were asked to evaluate the relative importance (influence) of ‘service attitude ($e_{21}$),’ ‘professional skills ($e_{22}$),’ and ‘responsiveness ($e_{23}$),’ in cluster $C_2$ on element ‘comprehensive information system ($e_{11}$),’ in cluster $C_1$. Then, by equation (7), the three pair-wise comparison matrices ($A^1, A^2, A^3$) based on the evaluation results were obtained as

$$A^1 = \begin{pmatrix} 1 & 1/5 & 1/3 \\ 5 & 1 & 2 \\ 3 & 1/2 & 1 \end{pmatrix}, \quad A^2 = \begin{pmatrix} 1 & 1/7 & 1/2 \\ 7 & 1 & 3 \\ 2 & 1/3 & 1 \end{pmatrix}, \quad A^3 = \begin{pmatrix} 1 & 1/8 & 1/3 \\ 8 & 1 & 2 \\ 3 & 1/2 & 1 \end{pmatrix}. $$

**Step 3.2.** By using equation (10), the importance weights $w^k_t$ ($k = 1, 2, 3; \ t = 1, 2, 3$) of elements $e_{2t}$ ($t = 1, 2, 3$) relative to element $e_{11}$ were:
Using equation (11), the eigenvalues $\lambda_k$ (k = 1, 2, 3) of pair-wise comparison matrices $A_k$ (k = 1, 2, 3) were:

$$\lambda_1 = 3, \quad \lambda_2 = 3.002, \quad \lambda_3 = 3.011.$$

Thus, by using equations (8) and (9), the consistency indices and consistency ratios of the pair-wise comparison matrices $A_k$ (k = 1, 2, 3) were:

$$C.I. = \frac{\lambda_k - n}{n - 1} \times \frac{3.000 - 3.000}{2} = 0, \quad C.R. = \frac{C.I.}{R.I.} = \frac{0}{0.58} = 0 < 0.1;$$

$$C.I. = \frac{\lambda_k - n}{n - 1} \times \frac{3.002 - 3.000}{2} = 0.001, \quad C.R. = \frac{C.I.}{R.I.} = \frac{0.001}{0.58} = 0.002 < 0.1;$$

$$C.I. = \frac{\lambda_k - n}{n - 1} \times \frac{3.011 - 3.000}{2} = 0.006, \quad C.R. = \frac{C.I.}{R.I.} = \frac{0.006}{0.58} = 0.100 < 0.1;$$

According to the consistency test results shown above, the consistency test was successful.

**Step 3.3.** Thus, using equation (12), the pair-wise comparison matrix $M$ of the relative importance of all elements given by three experts could be constructed as:

$$M = \begin{pmatrix}
1 & 0.153 & 0.381 \\
6.542 & 1 & 2.289 \\
2.621 & 0.437 & 1
\end{pmatrix}.$$

By using equation (13), the influence weights $w_t$ (t = 1, 2, 3) of elements $e_t$, (t = 1, 2, 3) relative to element $e_1$ were

$$w_1 = 0.100, \quad w_2 = 0.632, \quad w_3 = 0.268.$$

By using similar procedures, the relative importance (influence) of a given set of elements in a cluster on any element in the same cluster or a different cluster could be found.

Step 4: Construction of an unweighted supermatrix

In accordance with Step 3, the first column of block $W_{2i}$ of the unweighted supermatrix was $(0.100 \ 0.632 \ 0.268)^T$.

By using equation (15), and employing similar procedures, all blocks could be obtained:

$$W_1 = (0)_{3 \times 3}, \quad W_{12} = (0)_{3 \times 3}, \quad W_{3} = (0)_{3 \times 3},$$
W_{13} = \begin{pmatrix} 0.486 & 0.436 & 0.410 \\ 0.213 & 0.252 & 0.306 \\ 0.301 & 0.312 & 0.284 \end{pmatrix}, \quad W_{21} = \begin{pmatrix} 0.100 & 0.286 & 0.196 \\ 0.632 & 0.458 & 0.502 \\ 0.268 & 0.256 & 0.302 \end{pmatrix}, \quad W_{22} = \begin{pmatrix} 0.202 & 0.296 & 0.319 \\ 0.452 & 0.408 & 0.396 \\ 0.346 & 0.296 & 0.285 \end{pmatrix}, \quad W_{23} = \begin{pmatrix} 0.203 & 0.214 & 0.266 \\ 0.340 & 0.378 & 0.254 \\ 0.338 & 0.301 & 0.330 \end{pmatrix}, \quad W_{31} = \begin{pmatrix} 0.485 & 0.451 & 0.488 \\ 0.258 & 0.187 & 0.320 \\ 0.402 & 0.435 & 0.426 \end{pmatrix}, \quad W_{32} = \begin{pmatrix} 0.378 & 0.428 & 0.402 \\ 0.284 & 0.271 & 0.268 \\ 0.301 & 0.312 & 0.284 \end{pmatrix}.

Using equation (14), the unweighted supermatrix could be represented as:

\[
W = \begin{pmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0.486 & 0.436 & 0.410 \\
0 & 0 & 0 & 0 & 0 & 0 & 0.213 & 0.252 & 0.306 \\
0 & 0 & 0 & 0 & 0 & 0 & 0.301 & 0.312 & 0.284 \\
0.100 & 0.286 & 0.196 & 0.202 & 0.296 & 0.319 & 0.203 & 0.214 & 0.266 \\
0.632 & 0.458 & 0.502 & 0.452 & 0.408 & 0.396 & 0.485 & 0.451 & 0.488 \\
0.268 & 0.256 & 0.302 & 0.346 & 0.296 & 0.285 & 0.312 & 0.335 & 0.246 \\
0.340 & 0.378 & 0.254 & 0.338 & 0.301 & 0.330 & 0 & 0 & 0 \\
0.258 & 0.187 & 0.320 & 0.284 & 0.271 & 0.268 & 0 & 0 & 0 \\
0.402 & 0.435 & 0.426 & 0.378 & 0.428 & 0.402 & 0 & 0 & 0 \\
\end{pmatrix}.
\]

Step 5: Calculation of the weighted supermatrix

Using equation (17), all weighted blocks were:

\[ W'_{11} = (0)_{3 \times 3}, \quad W'_{12} = (0)_{3 \times 3}, \quad W'_{33} = (0)_{3 \times 3}, \]

\[ W'_{13} = f_{31} \times W_{13} = 0.453 \times \begin{pmatrix} 0.486 & 0.436 & 0.410 \\ 0.213 & 0.252 & 0.306 \\ 0.301 & 0.312 & 0.284 \end{pmatrix} = \begin{pmatrix} 0.220 & 0.198 & 0.186 \\ 0.097 & 0.114 & 0.138 \\ 0.136 & 0.141 & 0.129 \end{pmatrix}, \]

\[ W'_{21} = f_{12} \times W_{21} = 0.556 \times \begin{pmatrix} 0.100 & 0.286 & 0.196 \\ 0.632 & 0.458 & 0.502 \\ 0.268 & 0.256 & 0.302 \end{pmatrix} = \begin{pmatrix} 0.056 & 0.159 & 0.109 \\ 0.351 & 0.255 & 0.279 \\ 0.149 & 0.142 & 0.168 \end{pmatrix}, \]

\[ W'_{22} = f_{22} \times W_{22} = 0.497 \times \begin{pmatrix} 0.202 & 0.296 & 0.319 \\ 0.452 & 0.408 & 0.396 \\ 0.346 & 0.296 & 0.285 \end{pmatrix} = \begin{pmatrix} 0.100 & 0.147 & 0.158 \\ 0.225 & 0.203 & 0.197 \\ 0.172 & 0.147 & 0.142 \end{pmatrix}, \]

\[ W'_{23} = f_{32} \times W_{23} = 0.547 \times \begin{pmatrix} 0.203 & 0.214 & 0.266 \\ 0.485 & 0.451 & 0.488 \\ 0.312 & 0.335 & 0.246 \end{pmatrix} = \begin{pmatrix} 0.111 & 0.117 & 0.145 \\ 0.265 & 0.247 & 0.267 \\ 0.171 & 0.183 & 0.135 \end{pmatrix}, \]

\[ W'_{31} = f_{13} \times W_{31} = 0.444 \times \begin{pmatrix} 0.340 & 0.378 & 0.254 \\ 0.258 & 0.187 & 0.320 \\ 0.402 & 0.435 & 0.426 \end{pmatrix} = \begin{pmatrix} 0.151 & 0.168 & 0.113 \\ 0.115 & 0.083 & 0.142 \\ 0.178 & 0.193 & 0.180 \end{pmatrix}, \]
Weights \( W_{32}^w = f_{32} \times W_{32} = 0.503 \times \begin{pmatrix} 0.338 & 0.301 & 0.330 \\ 0.284 & 0.271 & 0.268 \\ 0.378 & 0.428 & 0.402 \end{pmatrix} = \begin{pmatrix} 0.170 & 0.152 & 0.166 \\ 0.143 & 0.136 & 0.135 \\ 0.190 & 0.215 & 0.202 \end{pmatrix}. \)

Using equation (16), the weighted supermatrix could then be represented as follows:

\[
W^w = \begin{pmatrix}
0.351 & 0.255 & 0.279 & 0.225 & 0.203 & 0.197 & 0.265 & 0.247 & 0.267 \\
0.149 & 0.142 & 0.168 & 0.172 & 0.147 & 0.142 & 0.171 & 0.183 & 0.135 \\
0.151 & 0.168 & 0.113 & 0.170 & 0.152 & 0.166 & 0 & 0 & 0 \\
0.115 & 0.083 & 0.142 & 0.143 & 0.136 & 0.135 & 0 & 0 & 0 \\
0.178 & 0.193 & 0.189 & 0.190 & 0.215 & 0.202 & 0 & 0 & 0
\end{pmatrix}
\]

Step 6: Obtaining the limiting supermatrix

Lastly, the limiting supermatrix could be represented as follows:

\[
W^l = \begin{pmatrix}
0.066 & 0.066 & 0.066 & 0.066 & 0.066 & 0.066 & 0.066 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 \\
0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 & 0.039 \\
0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 & 0.044 \\
0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 & 0.129 \\
0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 & 0.239 \\
0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 & 0.154 \\
0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 & 0.105 \\
0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 & 0.089 \\
0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135 & 0.135
\end{pmatrix}
\]

According to the limiting supermatrix, the weights of all elements were found to be as shown in Table 5.

<table>
<thead>
<tr>
<th>Elements ( e_i )</th>
<th>( e_{11} )</th>
<th>( e_{12} )</th>
<th>( e_{13} )</th>
<th>( e_{21} )</th>
<th>( e_{22} )</th>
<th>( e_{23} )</th>
<th>( e_{31} )</th>
<th>( e_{32} )</th>
<th>( e_{33} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weights ( w_i )</td>
<td>0.066</td>
<td>0.039</td>
<td>0.044</td>
<td>0.129</td>
<td>0.239</td>
<td>0.154</td>
<td>0.105</td>
<td>0.089</td>
<td>0.135</td>
</tr>
<tr>
<td>Ranking</td>
<td>7</td>
<td>9</td>
<td>8</td>
<td>4</td>
<td>1</td>
<td>2</td>
<td>5</td>
<td>6</td>
<td>3</td>
</tr>
</tbody>
</table>

According to Table 5, the three most important service quality solutions for this logistics center at Kaohsiung port are therefore ‘professional skills \( e_{22} \),’ ‘responsiveness \( e_{33} \),’ and ‘cargo tracking and management ability \( e_{33} \).’ Furthermore, based on the weight distribution, the importance weights of the three aspects - convenience, service personnel, and professionalism - are 0.149, 0.522, and 0.329. Hence, ‘service personnel,’ which is ranked 1st, is the most important cluster influencing service quality at this logistics center, followed by ‘professionalism’ in second place, and ‘convenience’ is ranked last.
4. Discussion and conclusion

Many papers in the literature investigate factors affecting service quality at international port logistics centers and solutions for the improvement of service quality, and both are important issues in the assessment of service quality at port logistics centers. Nevertheless, in assessing factors affecting service quality and assessing solutions for the improvement of service quality, each of these influencing factors/solutions can be seen as sets of variables containing several criteria. The DEMATEL technique can be used in conjunction with an expert survey to clarify the causal relationships between individual criterion variables, and can both transform the causal relationships between individual criterion variables into a clear structural model [22], and also account for the interdependence among the criterion variables [23]. It is clear that the conventional AHP technique [17] cannot effectively handle situations in which criteria or sub-criteria have dependent relationships or feedback effects, which is because the AHP technique can only deal with criteria that are mutually independent. For its part, the ANP technique proposed by Saaty [16] is able to adequately deal with criteria that have dependent relationships or feedback effects, which is because ANP consists of AHP plus a feedback mechanism and uses a supermatrix to derive the degree of influence of mutual interdependence. As a consequence, this paper proposes the use of a hybrid MCDM model incorporating AHP, DEMATEL, and ANP to assess the causal relationships between criterion variables, including dependent relationships and feedback mechanisms. Lastly, this study focuses port logistics center service quality solutions, and applies the hybrid MCDM model constructed in this paper in order to explain its functioning and assessment procedures.

First, this paper constructs a hybrid MCDM model incorporating AHP, DEMATEL, and ANP. To ensure that this model was easy to apply, scientific, and systematic, a detailed explanation of application procedures is provided for each of the six operating steps. Second, using a hypothetical port logistics center case as an example, this paper applies the model to the assessment of port logistics center service quality solutions. Finally, via the various operating steps, the key logistics center service quality solutions are ranked, allowing logistics center operators to implement them in the resulting order of priority.

The advantages of the hybrid MCDM model combining the AHP, DEMATEL, and ANP methods can be summarized as follows:
(1) The ANP technique can be used to deal with feedback and dependence among evaluation criteria (including both clusters and elements within clusters).
(2) The DEMATEL technique can be used to gauge the degree of mutual dependence between clusters.
(3) The model ensures that the decision-making process is relatively realistic and effective.

Although the proposed hybrid MCDM model was used in this study to identify the weights of solutions relative to service quality at a port logistics center, it can also be applied to other evaluation/selection problems, such as the evaluation of risk factors, supplier selection, shipping alliance evaluation, port competitiveness evaluation, and many related management decision-making or strategy evaluation/selection problems. Furthermore, in practical assessment work, because the selection of solutions and selection of attributes or criteria tend to be vague and qualitative in nature, and it can be difficult to express assessment results as precise values, this assessment model can also be combined with fuzzy set theory [39] and linguistic variables [40] in order to deal with imprecise decision-making problems.
REFERENCES


Use of a hybrid MCDM method to evaluate key solutions influencing service quality at a port logistics center in Taiwan

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URANS SIMULATION OF A PARTIALLY SUBMERGED PROPELLER OPERATING UNDER THE BOLLARD CONDITION

UDC 629.5.016: 629.5.035:519.6
Original scientific paper

Summary

The performance of a partially submerged propeller under the bollard condition was assessed using numerical simulations based on URANS. The simulations were performed with varying propeller rotating speed and submergence depth. The propeller rotating speed was varied from 2 rps to 8 rps with the interval of 2 rps at the submergence depths of \( h/R = 0.0, 0.5, \) and 1.0. Here, \( h \) is the submergence depth from the free surface to the propeller shaft center and \( R \) is the radius of the propeller. The thrust and torque losses were compared with the thrust and torque in the deep water condition. The thrust and torque decreased rapidly with increasing propeller rotating speed. The thrust and torque ratios were compared with the empirical formula showing generally good agreement. The hydrodynamic characteristics around the partially submerged propeller were investigated using numerical simulations.

Key words: Propeller; Free surface; Partially submerged propeller; Thrust loss; Air ventilation; Bollard condition

1. Introduction

Shipyards perform tests to check the operability of the main propulsion system and auxiliaries for a moored vessel in a quay before a sea trial. A propeller during the test would be partially submerged due to the limited water depth of the quay. An accurate estimation of the thrust and torque for the partially submerged propeller is needed to ensure the safety of mooring lines holding the ship during the test [1]. In addition, thrusters installed for dynamic positioning can work under heavy sea conditions. The vertical motions of a vessel or offshore structure and the waves bring the thrusters closer to the free surface, making them more susceptible to ventilation [2]. A surface-piecing propeller is one of the most efficient propulsion systems for high-speed vessels. They can use a larger propeller size because it is not limited by the minimum blade tip clearance from the hull or the maximum vessel draft. Moreover, they can avoid cavitation damage because the propeller operates under ventilated conditions by drawing air from the free surface [3].
The common characteristics of the above three propellers are working with air ventilation under partially submerged condition. On the other hand, the propeller working in the quay and the thruster for dynamic positioning are different from the surface-piercing propeller in the inflow velocity of a fluid because they work in a fixed-position. The propeller operation with a zero advance speed is called the bollard condition. Thrust and torque are very large under the bollard condition. Large torque and sudden variations of the load conditions can be caused by intermittent ventilations, which occur on the partially submerged propeller experiencing continuous cycles of water-exit and re-entry [4].

The partially submerged propeller experiences thrust loss due to the air ventilation phenomenon. The thrust loss can be caused by the loss of propeller disk area, Wagner effect, thrust loss due to wave-making by the propeller, and air ventilation [5]. Kempf [6] first studied the air ventilation effect on the propellers. Since then, there have been many studies of the air ventilation of propellers [7~12]. Koushan [13] focused on the thrust loss due to air ventilation as well as the effects of air ventilation on the dynamics of the blade thrust and torque about the propeller shaft. The experiment was performed with a ducted propeller and an open propeller with a heave motion. Koushan et al. [2] examined the effects of waves and the propeller loading of an open pushing thruster though an experiment. They showed that the effects of the wave height are significant, particularly for the sub critical region (advance coefficients larger than 0.4), where higher waves lead to larger thrust loss. Califano & Steen [14] proposed two main ventilation mechanisms depending on the propeller submergence, loading, and advance coefficient. One is the free-surface vortex at the deeper submergence, and the other is the tip vortex at moderate submergence. These two mechanisms can exist separately or simultaneously. Park et al. [1] took the model experiment with two partially submerged propellers to derive the empirical formula to predict the thrust and torque losses according the submergence depth and ventilation number.

Numerical methods based on potential theory were developed to simulate the air cavity sucked down from the free surface at the early stages [3, 12]. On the other hand, with the development of computer systems, research using URANS (Unsteady Reynolds averaged Navier-Stokes) simulations is becoming increasingly popular. Califano & Steen [4] simulated a fully submerged propeller \((h/R = 1.4)\) working at a high loading \((J_A = 0.1)\) using a URANS code. Kozlowska et al. [15] compared the URANS simulation results with the experimental data performed using a range of advance coefficients \((J_A = 0.0~0.6)\) under the fully submerged condition \((h/R = 1.5)\). On the other hand, Park et al. [1] showed the possibility of URANS simulation for a partially submerged propeller working under the bollard condition.

In this study, numerical simulations using URANS were carried out to investigate the flow characteristics with air ventilation according to the submergence depth and propeller rotating speed. The propeller rotating speed and submergence depth \((h/R)\) were respectively 2 \(rps\) to 8 \(rps\) at 2 \(rps\) intervals and 0.0, 0.5, and 1.0. Here, \(h\) is the submergence depth from the free surface to the propeller shaft center and \(R\) is the radius of the propeller. The time series of the thrust and torque of one blade during a single revolution were investigated to understand the effects of air ventilation.

2. Mathematical and numerical models

2.1 Governing equations

The governing equations for the numerical simulation are the continuity equation and the incompressible URANS equations. The integral forms of the equations are as follows:
\[
\frac{d}{dt} \int_{\Omega} \rho d\Omega + \int_{S} \rho u_i n_i dS = 0
\]
\[
\frac{d}{dt} \int_{\Omega} \rho u_i d\Omega + \int_{S} \rho u_i u_j n_j dS = \int_{S} (\tau_{ij} n_j - p n_i) dS + \int_{\Omega} \rho b_i d\Omega
\]
where \( \rho \) and \( p \) are density and pressure, respectively; \( u_i \) is the velocity tensor and \( b_i \) is the tensor of body forces; and \( \tau_{ij} \) is the effective stress of the viscosity and turbulence, defined as
\[
\tau_{ij} = \mu_e \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right)
\]

2.2 Numerical methods

Commercial CFD software STAR-CCM+ 11.04 was used for the simulations in this study. The STAR-CCM+ is based on the finite volume method. The convection and diffusion terms were discretized using a 2nd order upwind scheme and a central difference scheme, respectively. For pressure-velocity coupling, the SIMPLE (Semi-Implicit Method for Pressure-Linked Equation) algorithm was implemented. The SST (Shear Stress Transport) \( k-\omega \) model was applied to a turbulence model. The VOF (Volume of Fluid) model based on HRIC (High Resolution Interface Capturing) scheme was implemented to capture the behavior of the free surface due to propeller operation.

3. Numerical simulation set-up

3.1 Model propeller

The model propeller used in this study is KP505, which was designed by the Korea Research Institute of Ships and Ocean Engineering (KRISO) for the KRISO container ship called KCS. The diameter of the full-scale propeller was 7.9 m and the number of blades was 5. The diameter of the model propeller was 250.0 mm from a scale ratio of 31.6. Table 1 and Figure 1 present the principal particulars and drawing of the model propeller, respectively.

Table 1 Principal particulars of the model propeller (KP505)

| Diameter (mm) | 250.0 |
| Scale ratio   | 31.6  |
| No. of blades | 5     |
| P/D (mean)    | 0.95  |
| Ae/Ao         | 0.800 |
| Hub ratio     | 0.180 |
| Section       | NACA66 |

3.2 Grid system

Figure 2 shows the computational domain and grid system for the numerical simulation. The trimmer mesh scheme using unstructured grids was applied to generate the grids around the model propeller. The grid system consists of a propeller block and background block. The propeller block surrounding the propeller blade and hub is a sliding mesh and rotates along a sliding mesh interface to consider the relative rotating motion of the propeller to the free surface. In addition, the boundary layer on the propeller blade surface was constructed using a prism layer so that the dimensionless wall distance was less than 1 (\( y^+ < 1 \)).
The grids around the tip and root of the model propeller have an additional layer to capture the tip and root vortices. Refinement grids were applied to capture the behavior of the free surface. The number of grids for the propeller block and background are 1.58M and 5.30M, respectively. On the other hand, the reinforced grids for the free surface were eliminated for the propeller open water (POW) simulations in the deep water. The number of background grid without the refinement grid for the free surface are 4.2M. The ratio of the submergence depth of the model propeller (\( h/R \)) is defined as the ratio between the depth (\( h \)) from the free surface to the propeller shaft and the radius of the model propeller (\( R \)), which are depicted in Fig. 2.

The propeller blade angle begins from the top position toward the propeller rotation direction, as illustrated in Figure 3. When a key blade is located at the top, the propeller blade angle is defined as 0°.
4. Results and discussion

4.1 Validation of numerical simulation

The numerical simulation method applied in the study was validated by a comparison with the propeller open water (POW) characteristics measured in a model experiment. The experiment was performed in KRISO using a model propeller. The diameter of the model propeller and the rotating speed was \( D = 250.0 \, mm \) and \( n = 14 \, rps \), respectively. The speed of carriage towing the POW test equipment was varied according to the advance coefficient \( (J_A = V_A/nD) \). The numerical simulations were performed under the same conditions as the model test.

The thrust \((K_T)\) and torque \((K_Q)\) coefficients and open water efficiency \((\eta_O)\) using the SST \(k - \omega\) model and \( k - \varepsilon\) model for the turbulence model are compared in Figure 4. The thrust and torque coefficients generally show good agreement with the experiment, even though the slope of the thrust and torque coefficients are slightly different from the experiment. Nevertheless, they showed very good agreement around the low advance coefficients. Because there was no significant difference between two turbulence models, the SST \(k - \omega\) model was applied to the simulations for the partially submerged propeller. Table 2 shows the errors between the CFD simulation using the SST \(k - \omega\) model and experiment.

![Fig. 4 Comparison of the propeller open water characteristics in the deep water between CFD and experiment](image)

**Table 2** Comparison of the thrust and torque coefficients in the deep water between CFD and experiment

<table>
<thead>
<tr>
<th>(J_A)</th>
<th>CFD (SST (k - \omega)) (a)</th>
<th>KRISO (b)</th>
<th>Error% (a/b-1)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(K_T)</td>
<td>(10K_Q)</td>
<td>(K_T)</td>
</tr>
<tr>
<td>0.1</td>
<td>0.480</td>
<td>0.684</td>
<td>0.476</td>
</tr>
<tr>
<td>0.3</td>
<td>0.385</td>
<td>0.569</td>
<td>0.381</td>
</tr>
<tr>
<td>0.5</td>
<td>0.278</td>
<td>0.440</td>
<td>0.276</td>
</tr>
<tr>
<td>0.7</td>
<td>0.168</td>
<td>0.307</td>
<td>0.177</td>
</tr>
<tr>
<td>0.8</td>
<td>0.114</td>
<td>0.237</td>
<td>0.128</td>
</tr>
<tr>
<td>0.9</td>
<td>0.057</td>
<td>0.158</td>
<td>0.076</td>
</tr>
</tbody>
</table>
4.2 Simulation conditions

In the model test, the kinematic similarity is satisfied from the same advance coefficient using a full-scale propeller. To satisfy the dynamic similarity, the Froude number, Reynolds number, Weber number, and ventilation number should be identified.

The ventilation number is usually defined using the relationship between pressure and inertia, such as the cavitation number.

\[ \sigma_v = \frac{2gh_{tip}}{V_R^2} \]

In the model test for the bollard condition, the inflow velocity is zero, which means the advance coefficient is zero \( (J_A = 0) \). Therefore, the ventilation number is expressed below under the bollard condition due to the zero inflow velocity.

\[ \sigma_v = \frac{2gh_{tip}}{(\pi n D)^2} \]

where \( h_{tip} \) is defined as \( h + R \) to compare with previous research (Park et al. [1]). The ventilation number decreases with increasing propeller rotating speed and decreasing submergence depth. The similarity for the ventilation number is satisfied automatically when the similarities for the geometry and Froude number are satisfied.

Park et al. [1] performed model tests with 5-blade and 6-blade propellers at five propeller rotating speeds (2, 4, 6, 8, 10 rps) and seven submergence depths \( (h/R = 0.0, 0.5, 1.0, 1.5, 2.0, 2.5, \text{ and } 3.0). \) The numerical simulations in this study were performed by varying the propeller rotating speed (2, 4, 6, 8 rps) at three submergence depths \( (h/R = 0.0, 0.5, \text{ and } 1.0). \) The maximum propeller rotating speed was determined from the Froude similarity due to the interaction with the free surface, while the propeller rotating speed in the deep water is usually determined to perform the model test in a high Reynolds number. As a result, the propeller rotating speed of 8 rps in the model scale corresponds to 85.4 rpm in the full scale. It is quite reasonable rotating speed when the maximum propeller rotating speed of large commercial vessels is around 100 rpm. Table 3 lists the ventilation number for each simulation condition. While the model tests in Park et al. [1] were carried out under the bollard condition of zero inflow velocity \( (J_A = 0) \), the numerical simulations were conducted with a very slow inflow velocity \( (J_A = 0.01) \) to improve the numerical stability.

Table 3 Ventilation number of the simulation conditions

<table>
<thead>
<tr>
<th>rps</th>
<th>h/R</th>
<th>0.0</th>
<th>0.5</th>
<th>1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>0.994</td>
<td>1.491</td>
<td>1.988</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>0.248</td>
<td>0.373</td>
<td>0.497</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>0.110</td>
<td>0.166</td>
<td>0.221</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>0.062</td>
<td>0.093</td>
<td>0.124</td>
<td></td>
</tr>
</tbody>
</table>

The Weber number for the effect of surface tension is defined as

\[ W_e = nD \sqrt[3]{\frac{\rho D}{s}} \]

where \( s \) is the surface tension of water. Shiba (1953) suggested a criterion \( (W_e > 180) \) to neglect the influence of surface tension. On the other hand, the Weber number at 8 rps is approximately 118, which is smaller than the criterion. The effect of the surface tension on the simulation result was investigated, as shown in Figure 5. The convergence of simulation with
the surface tension was very poor as compared to the simulation without the surface tension, even though very smaller relaxation factors was applied to the simulation with the surface tension. Therefore, the effect of surface tension was not implemented in the other simulations.

Before the simulations for all cases, the size of time step was investigated with the time steps of 1 degree and 2 degrees for the case of 4 rps and $h/R = 0.5$, as shown in Figure 6. They show very good agreement even though the variations of thrust and torque of 1 degree are a little more fluctuating. Therefore, the simulations for the other conditions were performed with the time step corresponding to 2 degrees because it is enough to investigate the tendency of thrust and torque losses due to the air ventilation. When the variation of thrust ratio during nine revolutions after the very first three revolutions are compared at 4 rps and $h/R = 0.5$, as shown in Figure 7, the deviation is not significant. The thrust ratio was based on the thrust under the deep water condition in the same propeller rotating speed. Therefore, the variations of thrust and torque during a single revolution in the same time period are compared for the other conditions.

A sharpening factor ($\kappa$) can be adjusted in STAR-CCM+, which is the factor to reduce the level of diffusion in the simulation. When the sharpening factor is 0, there is no reduction in numerical diffusion. When the sharpening factor is 1, on the other hand, there is no numerical diffusion with a very sharp interface between the two phases (STAR-CCM+ User Guide). To determine the optimal sharpening factor, three sharpening factors were tested with 4 rps at $h/R = 0.5$. Figure 8 shows the iso-surfaces of the free surface according to the number of sharpening factor. The blue iso-surface indicates the surface where the volume fraction of water is 0.5. The interface between water and air at the sharpening factor of 0.0 is relatively smoother than the other higher sharpening factors. When the variations of the thrust and torque of one
blade during a single revolution are compared, as shown in Figure 7, the fluctuations of the thrust and torque increase in the higher sharpening factors. Therefore, a sharpening factor of 0.5 was selected to compromise the shape of interface and the variations of thrust and torque, and then it was applied to all simulations in this study.

![Figure 7](image1)

**Fig. 7** Variation of the thrust ratio according to the revolution

![Fig. 8](image2)

**(a) \(\kappa = 0.0\)  (b) \(\kappa = 0.5\)  (c) \(\kappa = 0.9\)**

**Fig. 8** Oblique view of the iso-surface of the free surface according to the sharpening factor (\(\kappa\))

![Fig. 9](image3)

**Fig. 9** Comparison of the thrust and torque ratios according to the sharpening factor

### 4.3 Simulation results

Figure 10 compares the instantaneous iso-surfaces of the free surface under each simulation condition. The iso-surface under the free surface is the air cavity sucked down to the water, and the iso-surface above the free surface is the water droplet splashed into air. The angle of the blade located on the top positon is zero. The next blade is located at 72° because the number of blades of this propeller is 5.
When the propeller rotating speed is 2 rps, the air bubbles sucked down to the water follow the tip vortex of the blade regardless of the submergence depth. On the other hand, because the relative position of the blade to the free surface and the hydrostatic pressure change according to the submergence depth, the amount of air bubbles ventilated due to the tip vortex decreases with increasing submergence depth. In particular, when \( h/R \) is 1.0, the tip vortex is not observed clearly because there is no water-exit and re-entry of the blade.

At the same submergence depth, air ventilation and wave-making increase with increasing propeller rotating speed. The amount of water droplets splashed by the blade exit and re-entry increase at a higher propeller rotating speed. The water droplet was observed more in \( h/R = 0.5 \) than \( h/R = 0.0 \) at a higher propeller rotating speed. On the other hand, the amount of the water droplet decreases in \( h/R = 1.0 \) because the blades do not pierce the free surface directly. Finally, the air cavity covers the entire blades under the free surface in 6 rps at \( h/R = 0.0 \) and 8 rps at \( h/R = 0.5 \), while the air cavity does not cover the entire blades at any propeller rotating speed at \( h/R = 1.0 \).

Figure 11 compares the thrust and torque ratios according to the submergence depth with those under the deep water condition in the same propeller rotating speed. The curves tend to be oscillating particularly at higher propeller rotating speeds because the curves show just one
revolution event, even though the behavior of the air cavity varies at each revolution due to the instability of the air ventilation, as shown in Figure 7.

![Diagram](image)

(a) 2.0 rps  
(b) 4.0 rps  
(c) 6.0 rps  
(d) 8.0 rps

*Fig. 11 Variation of the thrust and torque ratios according to the submergence depth*

At $h/R = 0.0$ and 2 rps, the thrust was zero from 330° to 60°, which is the range where the blade is out of the water, and the thrust and torque increase rapidly due to the blade re-entry. The maximum ratios of thrust and torque are greater than one, which means that the thrust and torque is bigger than those under the deep water condition. This phenomenon was observed in the experiments conducted by Califano and Steen [14] and Kozlowska et al. [15]. With increasing submergence depth, the range of thrust recovery extends and the slope of the thrust increment at the blade entry region becomes gentle. The thrust loss is larger than the torque loss at the blade entry region, while the torque loss is larger than the thrust loss at the blade exit region at all submergence depths. On the other hand, the thrust loss is generally larger than the torque loss and the difference between thrust and torque losses is not significant at the other propeller rotating speeds except for 2 rps.

The maximum thrust and torque ratios were approximately one at all submergence depths at 4 rps. Air cavity sucked down the free surface covers the entire blade areas under the free surface at $h/R = 0.0$ in 4 rps, as shown in Figure 10. As a result, the maximum thrust ratio at those conditions was less than one. On the other hand, the maximum thrust and torque ratios of $h/R = 0.0$ and $h/R = 0.5$ at 6 rps were smaller than that due to ventilation. In addition, the maximum values at $h/R = 1.0$ were less than one at 8 rps. As shown in Figure 10, the air bubbles ventilated at $h/R = 1.0$ at 8 rps covered outer radii region of the blade positioned at the bottom.
The thrust and torque ratios shown in Figure 11 are averaged during a single revolution and are summarized in Table 4. The thrust and torque ratios at \( h/R = 0.0 \) were larger than the propeller disk area under the free surface at 2 rps, and they reached less than 20% at 8 rps. Furthermore, the thrust ratio at \( h/R = 1.0 \) was smaller than 60% at 8 rps. As explained in Figure 11, the thrust loss was larger than the torque loss in all cases, which means a decrease in propeller efficiency. The propeller efficiency decreased with increasing propeller rotating speed and decreasing submergence depth.

### Table 4 Thrust and torque losses according to the submergence depth and propeller rotating speed

<table>
<thead>
<tr>
<th>( h/R )</th>
<th>rps</th>
<th>( T/T_0 )</th>
<th>( Q/Q_0 )</th>
<th>( \eta/\eta_0 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>2</td>
<td>0.578</td>
<td>0.588</td>
<td>0.982</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>0.360</td>
<td>0.400</td>
<td>0.899</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>0.202</td>
<td>0.239</td>
<td>0.847</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>0.146</td>
<td>0.176</td>
<td>0.827</td>
</tr>
<tr>
<td>0.5</td>
<td>2</td>
<td>0.789</td>
<td>0.801</td>
<td>0.984</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>0.549</td>
<td>0.599</td>
<td>0.916</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>0.407</td>
<td>0.468</td>
<td>0.869</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>0.259</td>
<td>0.314</td>
<td>0.824</td>
</tr>
<tr>
<td>1.0</td>
<td>2</td>
<td>0.944</td>
<td>0.950</td>
<td>0.993</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>0.822</td>
<td>0.849</td>
<td>0.968</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>0.710</td>
<td>0.751</td>
<td>0.945</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>0.568</td>
<td>0.608</td>
<td>0.934</td>
</tr>
</tbody>
</table>

Figure 12 compares the pressure distributions of the blades at \( h/R = 0.0 \) and 2 rps with those under the deep water condition. The area of negative pressure on the suction side was larger due to the air cavity on the blade surface after the blade entered the water. The areas of the positive pressure on the pressure were also larger than those under the deep water condition for the same reason. This can explain why the maximum thrust and torque ratios at \( h/R = 0.0 \) and 2 rps can be greater than one, as shown in Figure 11. At \( h/R = 0.5 \) and \( h/R = 1.0 \), the areas of the pressure and suction side on the fully submerged blade were also larger than those under the deep water condition. In addition, at \( h/R = 1.0 \), the pressure of the blade at the top was affected by ventilation, even though it did not penetrate the free surface.

Figure 13 shows the variation of the thrust and torque ratios according to the propeller rotating speed at each submergence depth to compare the effect of the propeller rotating speed more clearly. The distribution of the thrust and torque ratios showed a similar tendency at the same submergence depth; even the magnitude differs according to the propeller rotating speed. The change in thrust ratio according to the propeller rotating speed depends on the submergence depth. The reduction rate of the thrust ratio decreases with increasing propeller rotating speed at \( h/R = 0.0 \), whereas the reduction rate of the thrust ratio is almost constant with the variation of propeller rotating speed at \( h/R = 0.5 \). The reduction rate of the thrust ratio at 2–6 rps of \( h/R = 1.0 \) is not as distinct as that at the other submergence depths.

The thrust and torque ratios summarized in Table 4 were plotted with the regression curves of Park et al. [1]. The regression was performed based on the data measured from model
tests with 5-blade and 6-blade propellers. The regression formula considers the effects of the ventilation number and submergence depth.

\[
\beta_0 = 1 - \frac{\cos^{-1}\left(\frac{h}{R}\right)}{\pi} + \frac{h}{\pi R} \sqrt{1 - \left(\frac{h}{R}\right)^2}, \quad (0.0 \leq \frac{h}{R} \leq 1.0)
\]

\[
\beta_Q = \beta_0 \left[1 - \frac{1}{28.6 \left(1 - \frac{0.18 h}{R} + \frac{0.18 h}{R}\right)} + 1\right]
\]

---

**Deep water**

- \(h/R = 0.0\)
- \(h/R = 0.5\)
- \(h/R = 1.0\)

Fig. 12 Pressure contours of back (left) and face (right) sides according the propeller rotating speed at 2.0 rps
The effects of the wetted-disk area ($\beta_D$) is determined by the submergence depth, which was suggested by Fleischer \[9\]. $\beta_Q$ is the torque ratio, which includes the effects of wave making and air ventilation according to the submergence depth as well as the effect of the loss of propeller disc area. The effects of wave making and air ventilation tend to decrease due to the lower propeller loading in a smaller submergence depth. The thrust ratio is defined as $\beta_T = \beta_Q^{1/m}$ using the torque ratio. Here, $m$ is a constant for the relationship between the thrust ratio and torque ratio. Minsaas \[11\] suggested a value between 0.8 and 0.85 for the constant, $m$, whereas Park et al. \[1\] obtained 0.9 for $m$ from their experiment. The constant, $m$, from the thrust and torque ratios in Table 4 is between 0.83 to 0.97. As a result, the average value is approximately 0.9, and it coincides with the value obtained from Park et al. \[1\].

Figure 14 compares the thrust and torque ratios obtained from the numerical simulation with the experiment data and regression curves reported by Park et al. \[1\]. The experimental data show an approximately 10% difference between the 5-blade and 6-blade propellers. The thrust and torque ratios of the 5-blade propeller is bigger than those of the 6-blade propeller because the loading of the 5-blade propeller is smaller than that of the 6-blade propeller. The results from the numerical simulation generally show a good tendency with the regression curves. The thrust and torque ratios at $h/R = 1.0$ show very good agreement with the experimental results of a 5-blade propeller even at a lower ventilation number. On the other hand, under the partially submerged condition, the numerical simulation tends to under-predict at lower ventilation numbers and over-predict at higher ventilation numbers.
Simulation results of $h/R = 0.0$ and $h/R = 0.5$ were closer to the experimental results of the 6-blade propeller rather than the 5-blade propeller at the lower ventilation number.

**Fig. 14** Comparison of the thrust and torque losses according to the ventilation number and submergence depth
(The propeller rotating speeds in EFD and CFD are $n = 2, 4, 6, 8, 10 \text{ rps}$)

5. **Conclusions**

The effects of air ventilation on the propeller performance according to the variations of submergence depth and propeller rotating speed in bollard condition were investigated using URANS simulations. The propeller rotating speed was varied from $2 \text{ rps}$ to $8 \text{ rps}$ with intervals of $2 \text{ rps}$ at submergence depths of $h/R = 0.0$, $0.5$, and $1.0$.

Initially, the numerical method applied to this study was validated by a comparison with the experimental data of a POW test performed at KRISO. The thrust and torque coefficients generally showed good agreement with the experimental data, particularly the low advance coefficients.

The numerical simulation with the partially submerged propeller shows realistically the physical phenomena by the air ventilation, such as air cavity and water splash due to the waterexit and re-entry of the propeller blade. The air ventilation increases with increasing propeller rotating speed and decreasing submergence depth.

When the propeller rotating speed is at a lower propeller rotating speed, such as $2 \text{ rps}$, the maximum thrust and torque of a blade during a single revolution are larger than the average values of the deep water condition. The change in pressure around the air cavity generated by the tip vortex increases the thrust and torque under the free surface.

The average thrust and torque losses during a single revolution increase with increasing air ventilation. In addition, the thrust loss is larger than the torque loss in all cases with decreasing propeller efficiency. The propeller efficiency decreases with increasing propeller rotating speed and decreasing submergence depth.

The thrust and torque losses were compared with the experimental data and empirical formula reported by Park et al. [1], showing good agreement at different submergence depths and ventilation numbers. Nevertheless, further study will be needed to improve the accuracy of the numerical simulations using the turbulence model such as detached eddy simulation (DES) or large eddy simulation (LES) and extend the empirical formula to various propeller loading conditions and submergence depths.
ACKNOWLEDGEMENTS

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REFERENCES


A REVIEW OF SHIP MOORING SYSTEMS

UDC 629.5.028.72: 629.5.028.722
Review paper

Summary

The physical principle that governs how ships are moored to a port has changed little over the years. Nevertheless, in recent decades, there have been developments in maritime transport towards increased vessel dimensions and operations in specialist terminals. These trends mean that offshore ports and mooring systems have to face more challenging conditions in terms of the waves, wind and drift current. At the same time, pier side port loading and unloading systems place demands on the mooring system, which must immobilise ships better. In this situation, the mooring system’s own equipment, such as lines, deck fittings and mooring winches, must also evolve to work alongside new port devices. It is also necessary to point out that changes in mooring will take place in subsequent years. These innovations in attaching the ship to the pier will be highlighted here as they mark a significant change in mooring and pier components.

Key words: Shipping; Ports; Ship; Mooring; Mooring lines; Mooring winches

1. Introduction, components and rules

Whenever a vessel ends its navigation, it goes to a terminal and stops its propulsion, remaining subjected to the action of drifts and winds. At this moment, to be kept safe, it must remain immobilised, fixing its position by means of the elements that make up the mooring system. In this work, the term mooring refers to the system that secures a ship to the terminal. Nevertheless, other solutions are possible, such as mooring to a buoy through single point mooring (SPM), multi-buoy mooring (MBM), floating production storage and offloading vessels (FPSO). Alternatively, ship to ship transfer (SST) may fall under the broad category of mooring, and therefore require specialised fittings or equipment in addition to those for mooring the vessel to the terminal. In any of these cases, an efficient mooring system (Liu et al., 2006; Hsu, 2012; Hsu, 2015) is essential for the safety of the ship, terminal and environment.
Traditionally the ship mooring system (SSM) has relied on an arrangement of mooring lines that attach the vessel to shore. Also used are on-board fittings, including chocks-fairlead, pedestal rollers and bitt - bollards (Fig. 1). A third consideration is the deck machinery that operates the lines: mooring winches. Over time, novel systems of mooring (NSM) have come into play, by applying alternative physical principles to join the ship to the quay. These provide a glimpse of how current systems will evolve in the future (Villa, 2015).

![Fig. 1 - Arrangement for mooring manoeuvres with a small electrical windlass to work as a mooring device and a double bitt with a cross-reference function. Source: Carral Design Engineering Solutions.](image1)

All of the mooring system’s components are determined by vessel type and size, as are other aspects of the project. They are also influenced by regulations applied to each case and by the rules from the classification society chosen by the ship-owner. Nevertheless, Classification Societies (CS) vary significantly in how they deal with the operation and design of mooring system’s components (Carral et al., 2015c).

The International Association of Classification Societies (IACS) in part harmonise CS requirements for mooring, anchoring and towing ships in its “Requirements concerning mooring, anchoring and towing” (IACS, 2007a). However, this document only specifies the number of equipment units. The association has not produced a document to harmonise design requirements for the equipment that operates the mooring lines: the mooring winches (Carral et al., 2015c). IACS URA2 - “Shipboard fittings and supporting hull structures associated with towing and mooring on conventional vessels” (IACS, 2007b) has unified CS standards for designing and building mooring fittings.

![Fig. 2 - Arrangement for mooring manoeuvres that employ constant tension winches with hydraulic operation and cross-referencing. Source: Carral Design Solutions.](image2)
It is up to flag states, governing states of the ports, ship-owners and loaders to create the regulation framework that makes it possible for the ship to operate safely. Inside this framework for ship safety, the states will act in two ways. Firstly, their own regulations, such as ISO or UNE standards, come into play. Moreover, international agreements like the OMI or OIT are also adopted (Carral et al., 2015a). For their part, ship-owners and loaders add a further dimension as they fulfil their own requirements. Among the bodies that establish their own guidelines are the Oil Companies International Marine Forum (OCIMF) and International Gas Terminal Tanker and Operators (SIGTTO). A further influence comes from the rules made by the classification society that ship-owners have selected for their ships. The implementing regulations for mooring systems that come from all these regulatory bodies have been provided in a study (Carral et al., 2015c).

The ISO represents regulatory bodies in more than 156 countries, including many from Europe. It has developed its implementing standards to the case under study through the ISO - TC 8 Technical Committee on Shipbuilding and Marine Structures. ISO 3730 (2012) “Shipbuilding - Mooring Winches” for designing and testing the mooring winches cites rules related to winch components and other mooring system elements. All of these are important references in this study.

Currently other, related implementing regulations on deck fittings exist: “Welded Steel bollards and to ropes and cables” (ISO 3913, 1980); “Steel wire ropes” (ISO 2408, 2004); “Fibre ropes” (ISO 1141, 2012); and EN 14687, 14686, 14685, 14684. It is necessary to emphasise that, at the time of publication, there are no national or international standards that specify the minimum strengths for high modulus, synthetic lines.

The Oil Companies International Marine Forum (OCIMF), whose main mission is to promote marine safety by means of responsible development for tankers and terminals, has established in its “Mooring Equipment Guidelines - MEG3”, what it takes for mooring systems to be safe (OCIMF, 2008).

2. Good mooring principles

A ship’s mooring system will always have to resist forces produced by wind, currents and surges from passing vessels (Remery, 1974; Lee, 2015) as well as the effects of waves and swells (Bowers, 1975; Papanikolaou, 1985). At the same time, under all kinds of mooring, it is also necessary to take into account factors like vessel type and size, the characteristics and disposition of its mooring system and terminal and, finally, the physical conditions of the port (Schelfn and Østergaard, 1995).

In the past, the only way to carry out accurate estimates on ship movement and the loads acting on mooring ropes was by performing costly tests with scale models. In recent decades, numerical methods based on simplifications (Table 1) have become available due to increased calculation power. Moreover, it has become possible to develop mathematical models suitable for calculating moored ship motions (Van Oortmerssen, 1976; Seidi et al., 1981; Roberts, 1981; Schellin et al., 1982; Fylling and Andersson, 1988)

Any method for defining design requisites has to envisage how the mooring system is arranged. In other words, it has to take into account the elasticity of the mooring lines and how the moored ship is subject to the action of the wind, currents and forces of the waves. In this way, the designer can choose and position mooring equipment and fittings on board and along the quay. Static calculation methods are used on mooring systems, as proposed by Natarajan and Ganapathy (1995), OCIMF (1994), Aamo and Fossen (2000) and OCIMF (2008).
Design is informed by the factors of wind action, so that one considers changes in intensity and the longitudinal or transverse angle of incidence. Maximum current is also important: the effect of the ship’s draught interacting with under keel water clearance (OCIMF, 2008). In any case, by taking into account the effect due to wind and maximum current, as indicated in Schelfn and Östergaard, (1995), one can also deal with other factors. Aerodynamic studies will have to be reviewed (OCIMF, 1994; OCIMF 2008; Paulauskas et al., 2009). From the results obtained in wind tunnels for different types of ships, it will be possible to determine the application coefficients in every case studied.

Table 1 - Predictions made using these mathematical models and their general, underlying assumptions. Source: author’s own, based on Schelfn and Östergaard (1995).

| 1 | First-order ship responses at wave frequencies are modelled as linear responses to harmonic waves using response functions due to waves of unit amplitude (transfer functions) for the six degree of freedom motions of the ship. Effects of the linearised stiffness of the mooring system can be accounted for. |
| 2 | Low-frequency ship motions in surge, sway and yaw are not affected by first-order ship motions. They are modelled as responses to wind, current and wave drift forces acting on the moored ship. |
| 3 | To obtain total response, first-order (high-frequency) response is combined with low-frequency response using an appropriate method. |
| 4 | Dynamic behaviour of mooring lines and fenders does not significantly affect motions of the moored ship. Consequently, mooring forces are determined from the instantaneous position of the fairleads and from load-deflection characteristics of mooring lines and fenders. |

From these proposals, a set of rules can be created to represent mooring principles. Table 2 includes the characteristics that must be respected in the mooring lines in terms of angles, materials and length. Table 3 outlines the properties that the mooring arrangement must have to with the different components involved.

Table 2 - Mooring pattern arrangement.

<table>
<thead>
<tr>
<th>CHARACTERISTIC</th>
<th>LAYOUT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mooring lines</td>
<td>Symmetry in number and with the lines positioned between sides and hull of forward and stern.</td>
</tr>
<tr>
<td>Breast lines</td>
<td>Orientation perpendicular to the amidships line and positioned as far as possible from forward and aft.</td>
</tr>
<tr>
<td>Springs</td>
<td>As parallel as possible to the amidships line.</td>
</tr>
<tr>
<td>Head and stern lines</td>
<td>Positioned along forward and aft with the same length as the rest.</td>
</tr>
<tr>
<td>Angles</td>
<td>Maintained within low values.</td>
</tr>
<tr>
<td>Tails</td>
<td>With steel line, synthetic tails facilitate handling and elasticity. All of the same length.</td>
</tr>
<tr>
<td>Material</td>
<td>The same material and class.</td>
</tr>
<tr>
<td>Length</td>
<td>The same length between the mooring winches and the bollard on shore.</td>
</tr>
</tbody>
</table>
Table 3 - Guidelines for shipboard mooring equipment arrangements.

<table>
<thead>
<tr>
<th>COMPONENT</th>
<th>PROPERTY</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mooring arrangement</td>
<td>Symmetrical, with the possibility of being used equally from every side</td>
</tr>
<tr>
<td>Areas of manoeuvre</td>
<td>Placed to fore and aft as far as possible from mid-section</td>
</tr>
<tr>
<td></td>
<td>Fairleads as far and low as possible. Springs placed at the ends of fore and aft from the cylindrical body</td>
</tr>
<tr>
<td></td>
<td>Breast lines placed at the ends of the ship to prevent yawing</td>
</tr>
<tr>
<td>Winches</td>
<td>Suitable position for manoeuvre, with equal line lengths up to the chocks and bollards</td>
</tr>
<tr>
<td>Mooring areas</td>
<td>Spaces free for the manoeuvre, which can be clearly seen from this area. Perfectly aligned lines that runs from the winches, bitt and fairlead on the side</td>
</tr>
<tr>
<td>Spares</td>
<td>Provide additional bits and fairleads</td>
</tr>
</tbody>
</table>

3. Mooring analyses

The ship and its mooring system constitute an integrated system that share a dynamic response to environmental loads. However, a mooring system designed to counteract wind and tidal stream action could also do so with wave forces. This is supported by the fact that the static analysis method is frequently sufficient for determining how many mooring lines are needed (Schelfn and Östergaard, 1995). For instance, the OCIMF has met these design conditions at many port terminals worldwide (OCIMF, 2008). Nonetheless, when waves are the dominant action, to carry out a static analysis makes no sense and a dynamic analysis is needed instead (Nakajima et al., 1982; Natarajan and Ganapathy, 1995).

Over recent decades, there have been two trends in sea transport. On the one hand, vessels have grown in size, while the offshore terminals built in deeper waters to accommodate them are more exposed to swells (Paulauskas et al., 2009; Stopford, 2009). On the other hand, large ship motions along the horizontal plane – that is, surge, sway, and yaw – can occur at sheltered ports in the absence of higher waves (Shiraishi et al., 1999). The latter is related to resonant processes associated with low-frequency waves, also known as infragravity or long waves. The infragravity waves have periods in the order of minutes and, therefore, can match the natural periods of moored ships and/or harbours (Kubo et al., 2001; Sakakibara et al., 2001; González-Marco et al., 2008; Kwak and Pyun., 2013; López et al., 2012). Large ship motions induced by both swell and long waves can hinder cargo handling and ship mooring operations, break mooring lines or even damage fenders and quay walls as a result of the impact on the vessel.

On these grounds, the efficiency of a mooring system should be assessed in terms of its ability to restrain ship motions (Shiraishi et al., 1999). Moreover, it is important for motion criteria to be suitably defined by considering port requirements or the reference values for safe working conditions found in literature on different vessels and port operations (Elzinga et al., 1992; López and Iglesias, 2014). Nonetheless, defining how a moored ship will respond to wave action is not a straightforward task. This can be explained by dynamic coupling with the mooring system and the nonlinearities associated with resonant processes (Low and Langley, 2008).
As a first step in analysing the performance of a mooring system, the wave conditions at berth should be determined (Van der Molen and Moes, 2009; Sakakibara and Kubo, 2008a). With this aim, pressure or acoustic sensors can be deployed in the area of interest to obtain the time series of surface elevation. Alternatively, wave propagation numerical models can be used with an accurate description of the harbour and the corresponding bathymetry as inputs. However, numerical approaches should offer the possibility of reproducing phenomena such as diffraction, refraction and reflection, as well as nonlinearities associated with resonance and long waves (Van der Molen and Wenneker, 2008). The most common models involve solving Boussinesq equations (Dong et al., 2013), elliptic mild-slope equations (Bellotti, 2007) or nonlinear shallow water equations (Van Vledder and Zijlema, 2014). Recently, new techniques based on artificial intelligence have also been proposed to estimate wave conditions at berth (López et al. 2015).

Once wave loads have been defined, the interaction of the ship with the incident waves is evaluated by considering the effects of the mooring system. This can be achieved with both numerical and physical models. Thanks to the improved performance of computers and recent theoretical advances in the field, numerical models have gained force in recent years (Sasa and Incecik, 2014). Nonetheless, physical models are still deemed one of the most reliable tools in studying the behaviour of moored ships in harbours; they are needed to calibrate or and validate numerical models (Pessoa et al., 2015; Rosa-Santos et al., 2014).

Other methods are still under development. However, the most advanced numerical techniques solve ship-wave interactions with a two stage procedure based on potential flow theory (Hirdaris et al., 2014). First, hydrodynamic coefficients are calculated by means of linear frequency-domain methods, known as panel models (Lee and Newman, 2005). The second step involves simulating the behaviour of the moored ship in the time domain frame by applying the impulse response technique (Cummins, 1962). At this stage, the external forces due to the mooring lines and fenders can be introduced into the dynamic system (Kwak and Pyun, 2013).

Stiffness, friction and damping parameters can be used to model the compression action of the fenders. However, if reference values are unavailable, friction and damping will be neglected (Pessoa et al., 2015). Fender systems must withstand large loads during berthing. These loads have commonly been calculated with deterministic methods. More recent and sophisticated methodologies include: statistical procedures (Ueda et al., 2001), the Quick Fender Selection Method (Das et al., 2015) or the finite element method (FEM) (Jiang and Gu, 2010).

The traditional methodology for analysing mooring lines is done separately from the ship motion analysis. This approach calculates vessel motions using simplified representations or scalar coefficients of the mooring lines. The vessel motions then serve as inputs for the FEM; each line in the mooring arrangement can be analysed in isolation and different materials can be explored. However, this methodology presents several shortcomings, especially when underwater elements and deep water conditions are involved (Ormberg and Larsen, 1998). More advanced methods simulate the behaviour of both the mooring lines and vessel in the time domain. These interconnected methods have been developed for the offshore industry (Chen et al., 2006; Girón et al., 2014; Low and Langley, 2008; Yang et al., 2012).
4. Mooring equipment

4.1. Mooring lines

Taking the action of forces produced on the ship as a starting point, the mooring system distributes the forces along the mooring lines. These forces are transmitted to the fixed elements of the terminal. The effectiveness of each will depend on the vertical and horizontal angles that make up the mooring line and that will have to present the lowest possible value (OCIMF, 2008)

The elasticity of a mooring line will depend on factors like the material, as well as its diameter and length. For this reason, between two mooring lines that present different flexible capacity and undergo identical working conditions, the one with the highest diameter and shortest length will, in the end, be subjected to the greatest effort (Schelfn and Östergaard, 1995).

Material is a crucial decision considering the extreme environmental conditions that the ship has to withstand when safe mooring is carried out (OCIMF, 2008; Aamo and Fossen, 2000). A load value will be obtained for this mooring rope: SWL (safe working load). With the corresponding safety factor (SF) playing a role, the MBL (minimum breaking load) can be determined (Table 4). At this point it is necessary to take into account the material to be used for the mooring rope and therefore define the appropriate rope diameter (OCIMF, 2008). Moreover, the material will influence the diameter of the winch drum and the space needed for the manoeuvre, in addition to the type and curvature radius of the chocks, Panama-type fairlead, pedestal rollers and fairleads (Schelfn and Östergaard, 1995).

Low stretch ropes are made of steel cables or synthetic materials, like high modulus polyethylene (HMPE). Using this second option has many advantages and is therefore ideal for larger sized vessels in which high loads are involved. It is also extremely suitable for restricting the ship’s motions, in small ports of limited size and loading and unloading operations with those connections. Another area in which HMPE performs well is ones in which high dynamic loads appear (OCIMF, 2008). Its advantages and specific features have been studied in (OCIMF, 2008; Crump et al., 2008; Pederson et al., 2011; Carral et al., 2015b; Carral et al., 2016).

Table 4 - Strength criteria for steel, polyamide and other synthetic mooring lines. Source: author’s own, based on OCIMF (2008).

<table>
<thead>
<tr>
<th>ELEMENT</th>
<th>SWL</th>
<th>SF=MBL/SWL</th>
<th>% MBL</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mooring lines</td>
<td></td>
<td>Steel 1.82</td>
<td>55</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Polyamide 2.22</td>
<td>45</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Other synth (HMPE) 2.00</td>
<td>50</td>
</tr>
<tr>
<td>Tails for wire mooring lines</td>
<td></td>
<td>Polyamide 2.50</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Other synth (HMPE) 2.28</td>
<td></td>
</tr>
<tr>
<td>Tails for synthetic mooring lines</td>
<td></td>
<td>Polyamide 2.50</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Other synth (HMPE) 2.28</td>
<td></td>
</tr>
<tr>
<td>Joining shackles</td>
<td>Equal to mooring lines to which attached</td>
<td>2.00</td>
<td></td>
</tr>
</tbody>
</table>

(1) Highest load calculated for adopted standard environmental criteria
On the other hand, synthetic lines with high elasticity will be used in smaller sized ships where the presence of loads of lower value means that other criteria come to the fore. Among these criteria are ease of use, lower cost and the interchangeability with other lines (OCIMF, 2008).

The steel line and HMPE (low stretch ropes) ones usually have at their ends a small length of synthetic rope, called a pennant or tail. These have many positive points. Tails provide elasticity to the line, are easier for dock crew to handle and make it possible to connect with the bollard and thus protect the main material from abrasion. Their use and the way they are attached to the main mooring rope have been studied in Carral et al. (2016) and Schelfin and Östergaard (1995).

4.2. Mooring winches

In the mooring manoeuvre, winches play a vital role. They have to fulfil a dual purpose: handling the mooring lines during the manoeuvre and, after that, keeping them in a suitable place when the ship is in port. The mooring system of the ship has to be adapted when its displacement is altered, or in response to changes in tide or current conditions. The manoeuvre can be done by means of continuous manual adjustments to every rope, or automatically with constant tension winches.

In contrast with conventional winches, constant tension winches automatically maintain the tension of the mooring line, in a preset value (Table 5). If during the operation the line exceeds a certain value, its drum turns in order to render more line. When the tension decreases, the line is recovered and a fixed tension is reached. Table 5 is based on the content of ISO 3730 (2012). It describes the mooring functions of both conventional or constant tension winches (Lee et al., 2000; Kim, 2014; Carral et al., 2015c; Carral et al., 2015d; Ji et al., 2015) (Fig. 3 and Fig. 4).
Table 5 – Mooring functions of winches in accordance with ISO 3730 (2012). Source: Carral et al. (2015c)

<table>
<thead>
<tr>
<th>OPERATION</th>
<th>Conventional mooring winches</th>
<th>Automatic, constant tension mooring winches</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mooring</td>
<td>By means of drum</td>
<td></td>
</tr>
<tr>
<td>Mooring line stowage</td>
<td>In drum</td>
<td></td>
</tr>
<tr>
<td>Tension maintained with brake</td>
<td>In drum</td>
<td></td>
</tr>
<tr>
<td>Mooring line work</td>
<td>Optional, by means of drum</td>
<td></td>
</tr>
<tr>
<td>Tension maintained through</td>
<td>Not available</td>
<td>In drum</td>
</tr>
<tr>
<td>automatic device</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 4 - Block diagram with the tension control of hydraulically operated mooring winches.

Hydraulic power uses pressure to work the winch, a simple way to control the tension on the mooring line. By means of the pressure sensor, there is a pressure change in the fluid that circulates between the pump and the hydraulic engine. This change corresponds in a linear manner with another in the traction force that the equipment exerts on the mooring line. Source: author’s own

4.2.1. Operation

The two most common power systems for mooring winches are electrical and hydraulic (Fig. 5 and 6). Hydraulic engines withstand a wide range of speed changes so that a constant torque is maintained. These engines may be slow and have radial pistons or work at a higher speed and have axial pistons. The pistons in the former are commonly used with direct transmission, while those in the latter are coupled by means of a differential (Carral et al., 2015c).
Alternating current engines are more economical, easy to install and of low maintenance. However, they always have to be installed along with a differential. When less power is needed, the economic option is asynchronous, alternating current squirrel cage rotary engines with four poles. When larger engines are needed, a more suitable choice is one with six or eight poles. The choice depends on the cost of combining differential and electrical power. Whenever a constant speed is required, then it is necessary to use hydraulic transmissions with a variable flow pump and possibly variable engine, electrical engines with frequency converter (Carral et al., 2015c) or permanent magnet engines (Lamas et al., 2016). However, one disadvantage remains: the lower efficiency of hydraulic power (Fig. 7, Table 11).

The data that are necessary for defining the winch are provided in the reference (Table 6) (Carral et al., 2013), and these include the type of work that the winch will carry out, traction, and the rope length it needs to stow.
Table 6 - Design parameter values for the mooring winches in accordance with different regulations.

<table>
<thead>
<tr>
<th>COMPONENT</th>
<th>ELEMENT / PARAMETER</th>
<th>SSCC</th>
<th>ISO 3730</th>
<th>OCIMF (MEG 3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Components</td>
<td>ROPE DRUM</td>
<td>-</td>
<td>Capacity geometry</td>
<td>Capacity geometry</td>
</tr>
<tr>
<td></td>
<td>WARPING END</td>
<td>To withstand 0.8 MBL</td>
<td>ISO 6482 (1980)</td>
<td>-</td>
</tr>
<tr>
<td>Technical characteristics</td>
<td>TRACTION</td>
<td>0.24-0.33 MBL</td>
<td>Nominal, max,</td>
<td>Nominal, max</td>
</tr>
<tr>
<td></td>
<td>SPEED</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>BRAKING</td>
<td>Withstands 0.8 MBL</td>
<td>Withstands 0.8 MBL</td>
<td>Withstands 0.8 MBL</td>
</tr>
<tr>
<td></td>
<td>DRUM</td>
<td>-</td>
<td>Max, min, diameter lines</td>
<td>Max, min, diameter lines</td>
</tr>
<tr>
<td>Control and SAFETY</td>
<td>EMERGENCY STOP</td>
<td>-</td>
<td>Yes</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>PROTECTION</td>
<td>-</td>
<td>Yes</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>SPEED</td>
<td>-</td>
<td>Adjustable</td>
<td>-</td>
</tr>
</tbody>
</table>

Other data are sometimes provided to supplement basic information: type of engine desired for the operation, diameter of the end to be used, render and recovery speeds, braking capacity, the dimensions and type of drum and, lastly, the type of differential.

This procedure first involves knowing the traction or nominal pull. With these data it is possible to consider the material. The value for the rope diameter and the nominal speed can be obtained from the contents in ISO and MEG 3 (ISO, 2012; OCIMF, 2008). This information in turn (Carral et al., 2016) lets one determine the dimensions for the warping end and drum, the type of power and reduction ratio, the power of the winches and the brake’s dimensions.

4.2.2. Dimensions of the drum

The ideal drum will meet the basic requirement of storing the entire length of cable or rope that is envisaged. It also fulfils the following conditions (Carral et al., 2015c): the mooring rope is not damaged on being coiled, the linear speed of the rendered and recovered cable is kept constant, and that (drum) which in accordance with the drum size, the final dimensions of the mooring winch are reduced).

Complying with these three conditions simultaneously is impossible. It will therefore be necessary to establish a priority list. To minimise the losses that the mooring rope incurs during operation, drums have to be made with cores of a large diameter. On the other hand, if the aim is for the mooring rope to obtain a constant linear speed, it will have to be wound in only one layer. This will make it necessary to produce very wide drums and will affect the dimensions of the winches (Carral et al., 2015c).

ISO and MEG 3 (ISO, 2012; OCIMF 2008) distinguish between two types of drums: single and split. In the first type, the entire rope is reeled into a single chamber, so that the load and linear speed vary from layer to layer (Fig. 8). In the second type, the drum is split.
into a load or working space and another one for stowage by means of an intermediate element. In the load space, the rope is reeled into one layer, so that a constant speed and load are maintained (Fig. 9).

![Fig. 8 - Mooring winches with reel. Source: Carral Design Engineering Solutions.](image)

![Fig. 9 – Split drum mooring winches. Source: Carral Design Engineering Solutions.](image)

**4.3. Ship fittings**

The document IACS URA2 - “Shipboard fittings and supporting hull structures associated with towing and mooring on conventional vessels” (IACS, 2007b) standardises guidelines for designing and manufacturing the deck components of mooring systems, as indicated below: "fittings and supporting structures used for the normal towing and mooring operations. Shipboard fittings mean those components limited to the following: bollards and bitts, fairleads, stand rollers, chocks used for the normal mooring of the vessel and the similar components used for the normal towing of the vessel".

ISO 3913 (1980) contains standards for mooring bitts (double bollards). In the case of fairleads, bend radius values of 10:1 are recommended for the Panama fairleads. As for the elements that withstand an elevated friction force, a bend radius value of 12:1 is advisable. A similar situation occurs with pedestal rollers, which have a bend radius value of 10:1 and a deflection no higher than 90° (Schelfl and Östergaard, 1995; Villa, 2015).

**4.4. Fenders**

Fenders are devices used to prevent damage in vessels and/or berth structures with a dual function: (i) to absorb the impact energy during berthing and (ii) to reduce vessel motions during unloading operations by acting with a suitable line arrangement. Each combination of vessel, berth structure and berthing conditions has its own requirements. Ship
size, berthing method, location, tidal range and water depth are among some of the factors that can influence which fender is chosen (Gaythwaite, 2004).

When fenders are chosen, the basic parameter is the ratio between the force to be withstood by the berth structure and the energy to be absorbed by the fender, which is known as the fender factor. In general, an ideal fender absorbs a large amount of energy and transmits a low reactive load to the berth structure. In other words, it has a low fender factor (Liu and Burchart, 1999). However, with surface-protection fenders and other cases, a high fender factor is advantageous (PIANC, 2002).

The fender factor depends on the material, shape, dimensions and design of the fender. Several materials, such as wood, used tires and rubber cylinders, were commonly used in the past and are still used for small vessels. However, these solutions cannot satisfy the current requirements for large modern ships (Das et al., 2014). State of the art fenders are made of rubber and polyurethane elastomers with excellent elastic properties and can be manufactured with repeatability (Galor, 2007).

Even though a vast variety of fender types are found on the current market, modern systems can be classified into two groups: bucking and pneumatic fenders (Das et al., 2014). This division is according to how they absorb or dissipate kinetic energy from the ships. The former can deflect considerably under loading and return to their original shape after unloading, transforming kinetic energy into elastic work. Some fenders of this group are in direct contact with the ship, while others are equipped with a frontal panel that increases the contact area. As for pneumatic fenders, the pressure of the air confined in bags increases above its normal value by transforming kinetic energy into compression work. A representation of several types of current fenders is shown in Fig. 10.

<table>
<thead>
<tr>
<th>Type</th>
<th>Fender shape</th>
<th>Stiff L</th>
<th>Reaction</th>
<th>Energy</th>
<th>Performance curve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glacier shape fender</td>
<td>Circular shape</td>
<td>D/H: 500/300, 3000/2000</td>
<td>60</td>
<td>4</td>
<td>4840</td>
</tr>
<tr>
<td>Longitudinal shape fender</td>
<td>Rectangular shape</td>
<td>H/L: 300/600, 1800/2000</td>
<td>0</td>
<td>9</td>
<td>5688</td>
</tr>
<tr>
<td>Spacing fender</td>
<td>Rectangular shape</td>
<td>H/L: 400/500, 2500/4000</td>
<td>6</td>
<td>9</td>
<td>1250</td>
</tr>
<tr>
<td>Bucking fender</td>
<td>Spacing fender</td>
<td>H/L: 200/100, 1000/2000</td>
<td>150</td>
<td>15</td>
<td>2290</td>
</tr>
<tr>
<td>Air bag</td>
<td>D/H: 800/400, 3200/3000</td>
<td>130</td>
<td>15</td>
<td>6100</td>
<td>4900</td>
</tr>
<tr>
<td>Pneumatic fender</td>
<td>Rectangular shape</td>
<td>H/L: 300/1000, 2000</td>
<td>60</td>
<td>4</td>
<td>10000</td>
</tr>
<tr>
<td>Foam filled</td>
<td>Rectangular shape</td>
<td>H/L: 1000/500, 3500/8000</td>
<td>200</td>
<td>4</td>
<td>4000</td>
</tr>
<tr>
<td>Cylindrical</td>
<td>Rectangular shape</td>
<td>H/L: 500/1000, 2800/5800</td>
<td>80</td>
<td>3</td>
<td>6000</td>
</tr>
</tbody>
</table>

Table 3.1 Different types of energy absorbing elastic deformation rubber units.

Fig. 10. Different types of fenders. Source: PIANC (2002).
The reference document for fender system design is the report of Working Group 33 of PIANC (PIANC, 2002). This group suggests the procedure presented in the flow chart from Fig. 11. In addition to a detailed fender design procedure informed by statistical analysis (Ueda et al., 2002), the document provides further relevant information on whole life considerations and special cases, among other topics of interest. Different national guidelines for the design of port facilities are based on PIANC (2002) and provide further information and criteria especially relevant to port engineers and planners (e.g. AS 4997, 2005; BS 6349-4, 2014; MLIT, 2009; UFC 4-152-01, 2005; ROM 2.0–11, 2012). In the particular case of floating pneumatic fenders, international standards are also available (ISO 17357, 2014).

Functional requirements + operational requirements + site conditions + design criteria

Initial fender layout

Berthing vessel

Calculation of berthing energy

Calculation of fender energy absorption

Selection of appropriate fenders

Determination of:
  - Energy absorption
  - Reaction force
  - Deflection
  - Hysteresis
  - Angular compression
  - Hull pressure

Check impact on structure/vessel
  - Horizontal and vertical loading
  - Chance of hitting the structure
  - Face of structure to accommodate the fender
  - Implications of installation

Moored vessel

Mooring Layout
  - Locations of mooring equipment
  - Strength and type of mooring lines
  - Pre-tensioning of mooring lines

Assume fender system and type

Dynamic analysis of the system

Check results
  Vessel motions and acceleration
  Fender deflection, energy and reaction force
  Mooring line forces

Optimization via reanalysis of the mooring system and vessel

Final selection of the fender

Fig. 11. Flow chart for fender. Source: PIANC (2002).

To simplify the design procedure, a Quick Fender Selection Method (QFSM) was proposed by Das et al. (2014), who obtained a “Ship-fender matrix” derived from application of QFSM to various ships in a terminal of Cochin Port (India). The method was also applied to the design of a safe mooring arrangement for Very Large Crude Carrier (VLCC) in a marine oil terminal (Das et al., 2015).

During port operations, damage to fenders may have a serious impact on mooring facilities or berthing ships. Therefore, in the design stage, finite element methods (FEM) can
be applied to model their complex material constitutive relationships and deformations. These methods can also define their crash responses with accuracy (Jiang and Gu, 2010). As a countermeasure to possible failures or damage, Sakakibara and Kubo (2007) proposed a monitoring system to check the loads on pneumatic-type fenders on site and alert to several critical conditions. The system can be extended to other types of fenders and port operations, such as ship-to-ship transfer operations and side-by-side moored ships.

5. Evolution of mooring systems

5.1. Mooring system/arrangement modification

In general terms, problems with berth operability associated with excessive ship motions can be mitigated in two different ways: by reducing wave action on ships or by modifying the ship mooring system’s response. The first group includes ‘hard solutions’ involving changes in the port infrastructures. These may entail: reducing the wave reflection coefficient of the target berth (Uzaki et al., 2010); reducing the wave penetration by extending breakwaters (McComb et al., 2009) or reducing resonance by modifying the harbour layout (Bellotti, 2007). The second group are seen as ‘soft solutions’ that, by modifying the mooring system, can be an effective and low-cost countermeasure (Sakakibara and Kubo, 2008a).

As mentioned in Section 3, there are no general rules for defining the most suitable mooring system. Each particular case study requires a joint dynamic analysis of the berthed ship and its mooring system. Nonetheless, several studies analysed the mooring arrangement of different vessel types in ports all around the world. They proposed different solutions to improve operation safety and reduce downtimes. The main results of these works are summarised below.

Van der Molen et al. (2006) studied the response of a coal carrier to long waves in Tomakomai Port (Japan) through numerical modelling techniques. They found that the pre-tension of the lines is a very important parameter. This can be varied to avoid resonance with long waves by shifting the natural period of the mooring system from the predominant period of waves. Therefore, varying mooring line pre-tension can be a good solution for reducing the motions of moored ships in the presence of low-frequency oscillations. On the other hand, smaller motions are the result of increasing the pre-tension value, but line loads also increase.

By means of field observations and numerical modelling, Sakakibara and Kubo (2008a) investigated the excessive motions of a coal carrier in stormy weather. They proposed a mooring system with new characteristics to escape from the resonance between the natural periods of the moored ship and those of the long waves in the port basin. The solution involved modifying the mooring lines and fenders along with adding new mooring dolphins at the ship’s bow and stern. It resulted in an effective countermeasure to restrain the low frequency motions of the ship.

In another work, Sakakibara and Kubo (2008b) evaluated the motions of moored oil tankers. They defined an ‘asymmetrical parameter’ as the ratio of spring constants between fenders and mooring lines and observed that the parameter clearly influences the subharmonic motions of the ships. Moreover, they found that harbour tranquillity can be enhanced when asymmetrical mooring systems are changed to symmetrical or weak asymmetrical ones. To achieve this, pneumatic type fenders were used to great effect.

Yoneyama et al. (2009) presented an alternative that reduces low-frequency surge ship motions of a moored ship by preventing resonance with long waves. The method uses computer-controlled hybrid mooring winches to forcibly change the natural period of the mooring system. The effectiveness of this solution has been checked with field demonstration experiments in Japanese ports.
In Ahuja et al. (2010a) and Ahuja et al. (2010b), a downtime assessment related to the Dahej LNG terminal in India is presented and the need for the construction of a breakwater discussed. After several studies, including numerical observations and field observations, the original idea of constructing the breakwater to improve downtime was discarded. As safety measures, the number of mooring lines was increased and constant tension, shore-based winches were used to establish safety measures for reducing the downtime for the ships at berth.

Rosa Santos et al. (2014) analysed different mooring arrangements to reduce moored ship motions and to improve operational and safety conditions at berth in the Leixões oil terminal (Portugal). The results of wave tank experiments showed that high friction fenders can increase tension mooring efficiency and reduce a moored ship’s motions, especially in the vicinity of the system’s natural periods of oscillation. However, varying the magnitude of pretension forces or the type of ship–fender interface makes a limited contribution to reducing moored ship motions under extreme conditions.

De Bont et al. (2010) carried out field measurements and numerical simulations at the Port of Shalah (Oman) to analyse the reduction in surge motions of moored containerships by MoorMaster™ units, a system composed of vacuum pads and hydraulics to secure and control the response of moored ships at berth developed by Cavotec (see next section). They found that MoorMaster™ units had a reducing affect on surge motions of moored ships. However, several parameters were not measured in the study and the results were not conclusive. More recently, Van der Molen et al. (2015) investigated alternatives to the mooring configuration of bulk carriers in Geraldton Harbour (Australia), including the deployment of twelve MoorMaster™ units. Although vacuum paddles reduced vessel motions, these motions exceeded the thresholds for maximum excursions of the arms for one of the analysed conditions. According to the results, the highest reduction in vessel motions can be obtained by installing a combination of pneumatic fenders and constant tension winches or nylon breast lines.

Finally, an innovative hydraulic mooring system called ShoreTension™ has been developed by the mooring company at the port of Rotterdam (KRVE, 2016). This system is situated on the quay side in between two bollards and automatically keeps mooring cables tense in severe conditions. While one end is fixed to the quay bollard, the ship line is connected to the moveable part of the system and a second quay bollard is used for guiding the ship line (Fig. 12). The system was simulated with a dynamic analysis for a generalised Liquid Natural Gas (LNG) terminal under combined wind and wave conditions. Results showed that the motions of the moored vessel and the loads on mooring lines are significantly decreased with ShoreTension™ (Van der Burg, 2010).

![Fig. 12 – Hydraulic mooring system. Source: KRVE (2016)](image-url)
5.2 Novel mooring systems. Classification in accordance with physical principle used.

For thousands of years, the traditional mooring system with ropes remained unchanged. Nevertheless, changes are currently taking place, having a profound impact on basic mooring principles. These changes challenge time-honoured, traditional and recognised mooring system with ropes.

At present, novel mooring systems (NMS) present two options: a vacuum system and mooring by means of a mechanical arm. Undoubtedly the mooring principle of the first one turns out to be more innovative. At the same time, it is more practical for having eliminated the need to modify the side of the ship. Moreover, it provides greater flexibility to the alignment between ground devices and the ship (Table 7) (Villa, 2015; Villa, 2014).

In 1999, a mooring system called IronSailor was installed for the first time, the work of Mooring Systems Limited (MSL) (Villa, 2015). It was used on the Aretere, a 150-metre-long ferry built by HJ Barreras in Vigo (Spain). The automatic mooring system comprised four 20 tonne units, placed in pairs. There were two units at the bow of the ship and another two at the stern. Activated from the control bridge, the units would open up to attach themselves to a steel plate on the wharf (Fig. 13).

Currently specific facilities are no longer required on a ship; the devices found along the quays can be attached directly to the sides of most ships (Fig. 14). The fact that its storage is retractable when it is not in use is a major advantage. This allows the device to remain behind the fender line to protect itself from impact during docking. When it is activated, the support structure of the device unfolds towards the exterior; mooring connection by vacuum is activated in few seconds (Kim et al., 2014; Cavotec, 2013). This mooring system by means of vacuum was designed to be compatible with most ships. It boasts a range of key characteristics, as indicated in Villa (2015):

- Operates in three directions.
- Accurately positions the ship.
- Manages loads; remote control possible by means of a real-time, computer network that records the information obtained (Villa, 2015).
**Fig. 14** – Vacuum-operated mooring system deployed along the quay. Source: Cavotec (2013).

**Table 7** – Novel mooring system (NMS) with its operational characteristics specified. Source: author’s own, based on CAVOTEC (2013) and TTS Group (2016)

<table>
<thead>
<tr>
<th>MOORING PRINCIPLE</th>
<th>ADVANTAGES</th>
<th>CHARACTERISTICS</th>
</tr>
</thead>
</table>
| VACUUM            | - Versatility, applicable to any existing ship  
                   - Increased operation speed for docking and casting off as well.  
                   - Increased manoeuvre economy, with less dead time for sailors and tugs.  
                   - Crew need not be involved in the manoeuvring.  
                   - Docking line efficiently used in face of longitudinal displacement of the ship.  
                   - Reduced task time and hazard for crew members; continuous adjustment of the mooring ropes unnecessary.  
                   - Ground power consumption instead of ship power.  
                   - Reduction in gap between ship and quay.  
                   - Reduction in ship motions. |
|                   | DISADVANTAGES | - Much greater investment needed in port infrastructure; ships must maintain their traditional mooring system until the new one is in place worldwide.  
                   - Increase ship’s dependence on the shore. |
| MECHANICAL ARM    | ADVANTAGES | - Increased operational speed in docking and casting-off manoeuvres, but alignment operation necessary between ship and quay.  
                   - Economy of manoeuvres, less intervention from boat-men and tugs.  
                   - Crew need not be involved in the manoeuvring.  
                   - Reduction in work and hazards for the crew members; no need for continuous adjustment. |
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- Ground power consumption instead of ship power.
- Reduction in gap between ship and quay.
- Reduction in ship’s motions.

DISADVANTAGES
- Docking line underused once longitudinal movement of the ship is no longer possible.
- Much greater investment needed in port infrastructure; ships must maintain their traditional mooring system until the new one is in place worldwide.
- Adaptation of the side of the ship (piece male) to receive the harnessing of the mechanical arm. Need to standardise this element.

5.3. Mooring line using HMPE

To achieve greater efficacy in mooring lines, marine industries have recently adopted high modulus polyethylene (HMPE) as a material. It has great advantages. Among these is its high resistance, similar to that of steel when diameter equality is taken into account. Another benefit is the relationship between resistance and weight, which is superior to that of any other natural or artificial fibre. Moreover, HMPE has low specific gravity so that even line buoyancy is possible (Wardenier, 2011), as seen in Table 8.

Table 8. Comparing the properties of 72 mm lines of diverse materials. Source: Carral et al. (2016).

<table>
<thead>
<tr>
<th>Material</th>
<th>Diameter (mm)</th>
<th>Weight (kg/100m)</th>
<th>MBL (kN)</th>
<th>Lengthening to 40 % of the MBL</th>
<th>Lengthening to 100 % of the MBL</th>
<th>Specific gravity</th>
<th>Melt point (ºC)</th>
<th>Dynamic coefficient of friction against metal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polyester double braid</td>
<td>72</td>
<td>447.9</td>
<td>1054</td>
<td>8.5%</td>
<td>15-20%</td>
<td>1.38</td>
<td>250</td>
<td>0.12-0.15</td>
</tr>
<tr>
<td>Steel</td>
<td>72</td>
<td>2200</td>
<td>3500</td>
<td>0.8%</td>
<td>2-3%</td>
<td>7.85</td>
<td>1600</td>
<td>0.23</td>
</tr>
<tr>
<td>HMPE</td>
<td>72</td>
<td>318.5</td>
<td>3470</td>
<td>1.5%</td>
<td>4-5%</td>
<td>0.97</td>
<td>140</td>
<td>0.07</td>
</tr>
</tbody>
</table>

Table 9. Outstanding properties of HMPE mooring lines. Source: Carral et al. (2016).

<table>
<thead>
<tr>
<th>Resistance</th>
<th>Weight</th>
<th>Lengthening</th>
</tr>
</thead>
<tbody>
<tr>
<td>Associated with the fibre, independent of the manufacturer, superiority over other fibres</td>
<td>Lighter than polyester and even more than steel, buoyancy</td>
<td>Low, which means little capacity for damping</td>
</tr>
</tbody>
</table>
With mooring operations in which constant tension is involved, it is necessary to for the drum to keep the line in continuous action. Here the mooring winch plays an active part and, as a result, line wear is a major concern (Carral et al., 2016). Along with all the other properties presented in Tables 8 and 9, a new aspect to take into account is line durability. Crump et al. (2008) have determined that, after 1500 work cycles, the residual resistance of the line end is reduced to 50% of its initial value in the tail and 30% in the middle section of the main line. Therefore, it is recommended that the line is replaced once this number of work cycles has been reached.

With these concerns in mind, operators have designed solutions to extend the line’s service life (Crump et al., 2008). The line can be rotated or its end can be cut when it has been severely grazed. Moreover, additional tails can be used at the end of the mooring line in the area withstanding the mechanical action of bits and fairleads. These additional lines could be made of the same material as the main line, or a traditional polyester or polypropylene rope can be used (Wardenier, 2011).

Carral et al. (2016) outline the characteristics common to fairleads used in conjunction with this material. Several factors must be obtained. The fairleads must be manufactured to be robust. They must have the appropriate radius for the line contact surface. Moreover, their surface finish must be of such a high degree of quality that the effect of grazing is minimised (Fig. 15). Another key factor is the inalterability of the fairleads’ surface over a period of time and in the face of harsh outdoor conditions.

5.4. Operating mooring winches by means of permanent magnet engines

In industry, permanent magnet engines (PMM) are used when constant torque and high efficiency are important factors for changes in speed. At the same time, these engines are
being used more in devices like lifts, where a reduced torque, as well as a lower noise and vibration level, are needed. In recent years, they have been perfected to offer a high degree of precision and reliability in applications involving a high transfer torque and low speed. One such case is the mooring winch. This innovative PMM technology in part makes it possible to dispense with reducers in numerous industrial sectors (Ikäheimo, 2002).

When compared with an induction engine of equal power, permanent magnet engines have a notably longer service life. However, their volume is smaller by 47 %, resulting in a high torque / volume relationship, and their weight is nearly 36 % lower (Kverneland, 2008). In terms of their efficiency, losses are 15-20 % lower than those caused by induction engines (Munz, 2014). A summary of these advantages can be found in Table 10.

Table 10 - Advantages of PMM. Source: Lamas et al. (2016).

<table>
<thead>
<tr>
<th>ADVANTAGES</th>
<th>DISADVANTAGES</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Greater precision and reliability when high transfer torque and low speed required</td>
<td>- Its power is not very high</td>
</tr>
<tr>
<td>- Greater performance</td>
<td>- Prone to demagnetisation</td>
</tr>
<tr>
<td>- Lower maintenance</td>
<td>- The characteristics of the machine cannot be modified</td>
</tr>
<tr>
<td>- Soft start motor</td>
<td>- Expensive</td>
</tr>
<tr>
<td>- Capable of reaching high speeds</td>
<td>- Technology still under development</td>
</tr>
<tr>
<td>- Capable of increasing power factor</td>
<td></td>
</tr>
</tbody>
</table>

Nevertheless, in the naval sector, the PPM has been, until now, less commonly used, with the exception of a few well-known cases linked to main propulsion (Rojas et al., 2009). Only very recently have they started to be applied to anchor hauling winches and other offshore applications (Vacon, 2016). Lamas et al. (2016) make a strong case for their future application in deck equipment. Table 11 compares this engine with its more conventional counterparts.

Table 11 - Types of winch engines according to the operating mode. Based on Lamas et al. (2016).

<table>
<thead>
<tr>
<th>MODE</th>
<th>HYDRAULIC</th>
<th>ASYNCHRONOUS ELECTRICAL</th>
<th>ELECTRICAL PMM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor type</td>
<td>Hydraulic low pressure</td>
<td>Triphasic asynchronous</td>
<td>PMM</td>
</tr>
<tr>
<td>Mechanic interface</td>
<td>Planetary reducer</td>
<td>Planetary reducer</td>
<td>Reducer with 2/3 phases or direct connection</td>
</tr>
<tr>
<td>Control speed and torque</td>
<td>High control of hydraulic flow in power unit</td>
<td>Mid-level CONTROL SPEED</td>
<td>Elevated CONTROL SPEED</td>
</tr>
<tr>
<td>Performance</td>
<td>54%</td>
<td>70%</td>
<td>90% (expected)</td>
</tr>
<tr>
<td>Surges overcapacity blockage</td>
<td>Elevated</td>
<td>Medium</td>
<td>Lower</td>
</tr>
<tr>
<td>Contamination</td>
<td>Spilled oils</td>
<td>No</td>
<td>No</td>
</tr>
<tr>
<td>Maintenance</td>
<td>High</td>
<td>Low</td>
<td>Lower</td>
</tr>
</tbody>
</table>
6. CONCLUSIONS

The term mooring has evolved. What once just referred to the system that secures the ship to the terminal can now also be applied to single point or multi-buoy mooring (MBM), floating production storage (FPSO) and the offloading of vessels or ship to ship transfers. A broader definition of mooring means that specialised fittings or gear is required. Therefore, widespread progress has been made in research and in their application, with the equipment itself and special fittings needed to adapt to these trends.

In this study, proposals have been made in relation to the principles of mooring. In these guidelines, the requirements – including angles, materials and length – of mooring lines were specified. Moreover, the properties required for the various components that make up the mooring system were outlined.

Also important for port operations are studies on reducing ship motions while the vessel is moored but in the presence of waves. These conditions have to be taken into account in the effort to improve the safety of moored vessels, as well as the design and efficiency of the harbour. The forces induced by wave action are compensated by the necessary elasticity of the spring mooring lines, fenders and other devices. Changes in the vessel’s elevation are also a concern due to changes in its displacement or tidal range. Line length has to be adjusted so that these changes can be compensated.

Key data for defining mooring winches are provided in Table 6. Other factors are included: the function that it is going to be carried out; tension or pull and the length of mooring rope that must be stored. By determining the tension or nominal pull, along with the physical characteristics of the material, one can obtain the value for the rope diameter and the nominal speed in accordance with ISO and MEG 3 guidelines. With all this information it is also possible to define drum dimensions, operation mode, reduction ratio, the winch power and brake dimensions.

Berth operability issues associated with excessive ship motions can be generally mitigated in two different ways. One is to reduce wave action on ships; the other is to modify the response of the ship’s mooring system. The first option involves ‘hard solutions’ through changes in the port infrastructure. These may entail reducing one of three elements: the wave reflection coefficient of the target berth, wave penetration by extending the breakwaters or resonance by modifying the harbour layout. With the second options, there are ‘soft solutions’, through a modification of the mooring system. These can be an effective and low-cost countermeasure.

Regarding novel systems of mooring (NMS) two systems come into play: one based on a vacuum and another which employs a mechanical arm. The first is more innovative, yet practical; it eliminates the need to modify the shell side of the ship. Moreover, it facilitates the alignment between the device placed on the wharf and the ship (Table 7).

If HMPE lines are used in the mooring operation based on constant tension, the line is continuously moved by the drum within mooring winches, and this movement leads to line wear. For this reason, certain properties (Table 8 and 9) must be considered, along with the additional concern of line durability.

The last topic explored in this study is related to the use of the permanent magnet motor in marine operations. In recent years, this type of motor has started to be used with anchor handling winches and other offshore applications. Lamas et al. (2016) are confident that the PMM will play an important role in to the operation of deck machinery.
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PROBING INTO THE EFFECTS OF CAVITATION ON HYDRODYNAMIC CHARACTERISTICS OF SURFACE PIERCING PROPELLERS THROUGH NUMERICAL MODELING OF OBLIQUE WATER ENTRY OF A THIN WEDGE

UDC 629.5.016.7:629.5.035.5:519.6
Original scientific paper

Summary

The current paper investigates flow around a blade section of a surface piercing hydrofoil. To this end, a thin wedge section is numerically modelled through an oblique water entry. The flow is numerically studied using a multiphase approach. The proposed numerical approach is validated in two steps. First, pressure and free surface around a wedge entering water are simulated and compared against previously published analytical results. Subsequently, cavitation phenomenon around a submerged supercavitating hydrofoil is modelled and analyzed. It is observed that cavity length, pressure, and lift force are accurately predicted. Subsequently, the main problem has been studied for two different cavitation numbers for a range of advanced ratios equivalent to fully, transition and partially ventilated conditions in order to investigate the effect of ambient pressure on hydrodynamics of the water entry of the foil. The numerical findings reveal that, when the cavitation number decreases, the start of transition mode is postponed and this mode is expanded for the larger range of velocity ratios. This implies that fully ventilated velocity ratio modes are expanded, too. However, in the transition mode, the cavitation number plays an essential role and may lead to a decrease in the pressure difference across the surface piercing hydrofoil which yields a decrease in the resultant force.

Key words: Surface piercing propeller; oblique water entry; thin wedge; Cavitation number; Numerical modelling;

1. Introduction

Surface piercing propellers (SPP) have various applications for different high-speed marine crafts including planing boats and surface effect ships (SES). When the blade of such a propeller enters the water, ventilated air is generated by gas formation. In addition, for many applications of these propellers where the pressure drastically drops, the possibility of this blade undergoing cavitation phenomenon increases. Consequently, a combination of water,
vapor, and air is generated near the blade, as it enters the water and the physics of the flow becomes quite complicated. From an engineering point of view, it is imperative to quantify the thrust force and the generated moment in the presence of cavitation which can be quite beneficial in the design process.

Shiba [1] is considered a pioneer in investigating the ventilation and cavitation mechanism of surface piercing objects. To ascertain the effective parameters on these phenomena, he conducted different experiments on the surface piercing (SP) hydrofoils. Based on the obtained experimental results, he concluded that ventilation occurs when the pressure in downstream of the surface piercing object drops below the ambient pressure minus the pressure associated with the surface tension. The feasibility of using supercavitating sections for the propellers in partially submerged conditions was first investigated by Hadler [2] and Hacker [3]. They conducted experimental measurements on the SP hydrofoils and SP propellers. Three main flow regions of fully ventilated, transition, and partially ventilated were initially introduced by them. Krupp [4] experimentally modeled surface piercing propellers and reported some results on the effects of Froude numbers and Reynolds numbers on scaling. Meanwhile, Brandt [5] used some of his experimental data and concluded that the point at which the transition mode starts depends on the cavitation and Froude numbers. Olofsson [6] measured the forces and torque of a SPP model 821-b. He reported these parameters during one cycle. By conducting a series of experiments, he found that during water entry process of the SP blades, the decrease in cavitation number may lead to an increase in the forces and moments, which was declared as quite unusual by him. However, experimental research by Shen [7] on 2D and 3D hydrofoils showed both increase and decrease in the lift force with the appearance of cavitation.

Numerical methods have also been utilized for investigating the submerged and surface piercing propellers. Furuya [8] presented the results of numerical simulation of the flow around a surface piercing propeller. His results were restricted to fully ventilated condition. Wang [9], on the other hand, used a lifting surface method to numerically model these propellers. Later, Kudo [10] used VLM method to model the SPPs. Also, Young and Kinnas [11] used surface panel method and BEM and analyzed flow around the surface piercing hydrofoils and propellers and reported the existence of cavitation phenomenon around the SPP blades. Through the same approach, Young and Savander [12] conducted hydroelastic analyses of large size SPPs and examined the effects of diameters on their hydroelasticity. Ghassabzadeh et al [13] have also applied this approach to evaluate hydroelastic behavior of the submerged propeller. Similar to the SPPs characteristics, operation close to the water free surface or percents of cavitation sheets could also affect the common marine propeller performances. The effects of these phenomena have been numerically studied by Ekinci et al [14], Bagheri et al [15], Kinaci et al. [16] and Samir et al [17].

Investigation of flow around surface piercing propellers can alternatively be accomplished through an oblique wedge water entry. Theoretical research of Yim [18] may be considered as one example of such approach. He presented simulation of water entry of thin wedges which were infinite and reported that such a solution may be useful for analyzing flow around surface piercing propellers. Wang [19] further developed these solutions in order to consider the fully ventilation mode of the flow for a hydrofoil. Later, Cox [20] experimentally studied water entry problem of thin wedges where the solution by Wang [19] was modified in such a
way that gravity effects is taken into consideration. Judge et al. [21] presented a numerical simulation for the potential flow around a wedge entering water with an oblique speed. They presented new boundary conditions for solving this problem. Afterward, Faltinsen and Semenov [22] presented a nonlinear theoretical simulation of the flow of an oblique water entry, where they provided a specific conformal mapping for solving of the problem. On the other hand, Xu et al. [23] presented numerical simulation of the oblique water entry of asymmetric wedges. Xu et al. [24] later extended their simulation in such a way that freefall of the wedge was considered. Recently, there has been good progress in numerical modeling of the flow related to water entry problems. Application of analytical method by Ghadimi et al. [25], URANS method by Ghadimi et al. [26-28], as well as SPH simulations by Farsi and Ghadimi [29-31] and Feizi Chekab et al. [32] in simulations of water entry of different objects have suggested that numerical solution of the viscous flow may be an appropriate choice for analyzing flow around wedges.

The main objective of the current paper is to offer further contribution to the understanding of cavitation phenomenon in surface piercing propellers through the usage of computational fluid dynamics and by investigating flow around a wedge. In this framework, different parametric analyses are conducted on a particular 2D blade section for two different cavitation numbers and various advanced ratios ranging from 0.23 to 0.64. Three different operating conditions including fully ventilated, transition, and partially cavitated are also considered. A purposive validation is presented through comparison in two steps. Finite volume based software ANSYS-CFX is employed as solver. Ultimately, the main results of the paper, which include the free surface, pressure, and forces acting on the wedge, are presented and it is illustrated how these factors are affected, as the impact velocity and cavitation number changes.

2. Problem Definition

2.1 Geometry of the problem

A surface piercing hydrofoil is considered to penetrate the water with vertical velocity of V, which may also have been influenced by an inlet velocity of U. This section is similar to a wedge whose deadrise angle is denoted by β. The ambient pressure is denoted by P\text{atm} which can be varied. Variation of this pressure may help initiate different cavitation numbers. During this type of water entry, three different conditions can be observed. These conditions may also occur for a surface piercing propeller. The first condition is called a fully ventilated mode in which water surface separates from the wedge apex in the leading edge and consequently, the air is sucked into the water (Fig. 1a). In this condition, the fluid behavior is perfectly steady and the hydrodynamic pressure acts only on the pressure side. For the constant inlet velocity U, if the impact velocity V decreases, a different condition appears which is called transition mode (Fig. 1b). In this condition, the wall of air cavity impacts the foil body and as a result, the cavity is disconnected from the atmosphere. Meanwhile, the pressure does not display a steady behavior and is highly affected by the speed ratios. As the impact velocity continues to decrease, a third condition called partially cavitation mode occurs in which the cavitation sheet begins from the leading edge and can remain on the suction side. As a result, no ventilation pattern is observed on the free surface (Fig. 1c).
2.2 Application of the considered problem for the surface piercing propellers

In Fig. 2. Through a close scrutiny of the blade of this type of a propeller, its section, and the presented velocities attacking the section (Fig. 2), one may conclude that a three-dimensional problem can be changed to a 2D water entry problem with an oblique speed, which is the exact replica of the problem introduced in the previous sub-section.

The propeller has a diameter $D$ and is considered to rotate with an angular speed of $n$. The advanced ratio of the propeller is determined by

$$J = \frac{U}{nD} \quad (1)$$

The dominant thrust and torque on the rotating propeller are found to be in the range of 0.5$r$ to 0.7$r$. Therefore, a better approach which can bring about meaningful results may be the consideration of a section between these two radii. Accordingly, the current paper focuses on the section 0.55$r$ from the hub. The impact velocity of this section can be computed by

$$V = 0.55\pi nD \quad (2)$$

The ratio of the inlet velocity to the impact velocity is defined by $U/V$ which is found by
Probing into the effects of cavitation on hydrodynamic characteristics of surface piercing propellers through numerical modeling of oblique water entry of a thin wedge

\[ \varepsilon = \frac{U}{N} = \frac{J}{0.55\pi}. \] (3)

The Froude number associated with the propeller is also defined by

\[ Fr = \frac{U}{\sqrt{gD}}. \] (4)

In most of the research papers conducted on propellers, the cavitation number is defined by

\[ \sigma_a = \frac{(P_v - P_0)}{0.5\rho_w U^2}, \] (5)

where \( P_v \) is the vapor pressure of the water and \( P_0 \) is the hydrostatic pressure at the propeller hub and since the hub of SPP is not submerged, \( P_0 = P_{atm} \) condition is assumed. The pressure acting on the blade section is normalized using the relation

\[ C_p = \frac{(P - P_0)}{0.5\rho_w U^2}. \] (6)

The force acting on the 2D wedge, which is obtained by integrating the pressure over the blade section, is non-dimensionalized using the relation

\[ K_F = \frac{F}{0.5\rho_w V^2 h} = \frac{F}{0.5\rho_w V^3 t}, \] (7)

where, \( h \) is the foil submergence at time \( t \) which is equal to \( V.t \). Meanwhile, the efficiency of the water entry process of a wedge, similar to the definition of propeller efficiency, is computed by

\[ \eta_{Wedge} = \frac{F_x U}{F_y V}, \] (8)

where \( F_x \) and \( F_y \) are the horizontal and vertical forces acting on the wedge, respectively.

2.3 Governing equations of the fluid

Since the present study deals with three different phases of vapor, water, and air, the homogeneous mixture model is used to calculate the mixture pressure and velocity. Accordingly, based on this model, the continuity and momentum URANS equations are implemented as

\[ \frac{\partial (\rho_m)}{\partial t} + \nabla \left( \rho_m \vec{V} \right) = 0, \] (9)
\[
\frac{\partial (\rho_m \vec{V})}{\partial t} + \nabla \cdot (\rho_m \vec{V}) = \rho_m g - \nabla P + \nabla \left[ \mu_m \left( \vec{V} \vec{V}^T - \frac{2}{3} \nabla \vec{V} \right) \right],
\]

where \( I \) is the unit tensor, while \( \rho_m \) and \( \mu_m \) are the mixture density and viscosity, respectively, which can be computed by

\[
\rho_m = \sum \alpha_n \rho_n = \alpha_a \rho_a + \alpha_v \rho_v + (1 - \alpha_a - \alpha_v) \rho_w, \quad (11)
\]

\[
\mu_m = \sum \alpha_n \mu_n = \alpha_a \mu_a + \alpha_v \mu_v + (1 - \alpha_a - \alpha_v) \mu_w.
\]

In the above equation, subscripts \( a, v, \) and \( w \) denote air, vapor, and water fluid, respectively. \( \alpha_n \) is the volume fraction of the \( n \)-th fluid in a cell, and it is known that

\[
\sum \alpha_n = 1. \quad (12)
\]

The mass continuity equation for the existing mixture can be written in the form of three different equations as in

\[
\frac{\partial (\alpha_a \rho_a)}{\partial t} + \nabla \cdot (\alpha_a \rho_a \vec{V}) = 0, \quad (13)
\]

\[
\frac{\partial (\alpha_w)}{\partial t} + \nabla \cdot (\alpha_w \vec{V}) = -\frac{\dot{m}}{\rho_w}, \quad (14)
\]

\[
\frac{\partial (\alpha_v \rho_v)}{\partial t} + \nabla \cdot (\alpha_v \rho_v \vec{V}) = \dot{m}, \quad (15)
\]

\( \dot{m} \) is the mass transfer rate that is only observed between the water vapor phases. By using the proposed evaporation and condensation model of Zwart [33], it can be written that

\[
\dot{m} = \dot{m}_e - \dot{m}_c, \quad (16)
\]

\[
\dot{m}_e = C_e \frac{3(1-\alpha_v)\alpha_{sv} \rho_v}{R_b} \sqrt{2(P_v - P_a)/3 \rho_w} \text{sgn}(P_v - P_a), \quad (17)
\]

\[
\dot{m}_c = C_c \frac{3\alpha_v \rho_v}{R_b} \sqrt{2(P_o - P_v)/3 \rho_w} \text{sgn}(P_o - P_v), \quad (18)
\]
Probing into the effects of cavitation on hydrodynamic characteristics of surface piercing propellers through numerical modeling of oblique water entry of a thin wedge

where $C_e$ and $C_c$ are empirical values being equal to 50 and 0.01, respectively. $R_b$ is the mean radius of the bubble and is assumed to be $1 \times 10^{-6}$ m, as proposed by Ji et al. [34]. $\alpha_v$ is the vapor fraction which is considered to be $5 \times 10^{-4}$, as proposed by Mejri et al. [35].

3. Problem Set Up

For solving the problem, the displayed domain in Fig. 3 is considered. Figure 3 shows the geometry of a wedge (supercavity hydrofoil) penetrating the water with impact speed of $V$, while the water enters the domain from the left side with a speed of $U$. Initially, the wedge apex is located at the water free surface.

Fig. 3. The considered foil section geometry and domain of the current problem.

The computational domain is discretized using a structured mesh. The cells have a hexahedron shape. The wedge walls are discretized using boundary layer mesh approach. Three different sub-domains are taken into consideration. As evident in Fig. 3, the moving zone cells move rigidly toward the set free surface, which cause the upper and lower zone cells to be decompressed and compressed, respectively. Therefore, in order to properly mesh the stationary sub-domains, a dynamic meshing approach is adopted. As a result, 0.5 million cells are produced. The discretized domain is illustrated in Figs. 3 and 4.

Fig. 4. The generated mesh.

4. Validation

The considered validation of the proposed numerical model consists of two different parts. In the first part, a wedge water entry is studied, while in the second part, cavitation around a supercavitating hydrofoil is modeled. In each of these parts, one aspect of the current simulation is validated.

4.1. Wedge entry to a calm water tank

Tveitnes [36] experimentally studied hydrodynamic characteristics of a series of symmetric wedges water entry. These experiments were conducted in calm water condition. The schematic of the wedge and simulated domain setup are displayed in Fig.5.
The validation case involves the water entry of a wedge whose length is \( L = 0.3 \) m and deadrise angle is 45 degrees. The wedge impacts the calm free surface with a constant vertical speed of \( V = 1.19 \) m/s. Free surface pattern of the wedge at \( V \cdot t = 0.5 \) m is illustrated in Fig. 6 (left) and the obtained dimensionless forces during the wedge motion are compared against the experimental results of Tveitnes [36] in Fig. 6 (right).

As evident in Fig. 6 (right), there is favorable agreement between the computed force coefficients and experimental data.

### 4.2 Cavitation

Here, a supercavity foil is numerically modeled by the proposed approach and the results are compared against available data. This foil was experimentally studied by Dinh [37]. This foil is a supercavity foil which is attacked by the flow at 9 degrees. The profile of this foil is displayed in Fig. 7.

This hydrofoil is modeled for five different cavitation numbers, cavity length, the pressure, and the acting lift on the body are computed. The computation is performed with 100,000 cells. The cavity shape is shown in Fig. 8 (\( c \) is the cord length). As evident in this figure,
Probing into the effects of cavitation on hydrodynamic characteristics of surface piercing propellers through numerical modeling of oblique water entry of a thin wedge

throughout the current simulation, length of the cavity increases as the cavitation number decreases, which is similar to what happens in reality.

The computed cavity length and the experimental data of Dinh [37] are displayed in Fig. 9. Based on the reported data, the results of the current simulation display relatively good agreement with the experimental results of Dinh [37] and there is only a slight under-prediction. It should be noted that radius of hydrofoil’s leading edge also has important effects on the generated cavity length. Since this value is unknown, it is set to be zero which leads to some errors. However, the RMS of the errors between the computed cavity length and that of Dinh’s experiment is determined to be 0.1 which is considerably small (Table 1).

![Fig. 8. Obtained cavity shapes.](image)

![Fig. 9. Comparison of the computed cavity length with experimental data of Dinh[37].](image)

<table>
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<tr>
<th>$\sigma_n$</th>
<th>Error%</th>
<th>RMS Error</th>
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<tr>
<td>0.21</td>
<td>11%</td>
<td>10%</td>
</tr>
<tr>
<td>0.236</td>
<td>8%</td>
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<tr>
<td>0.26</td>
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<td>0.32</td>
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</table>

The computed pressure and experimental data of Dinh [37] are displayed in Fig. 10. The results of free-streamline theory by Wu [38] are also presented in this figure. As evident in this figure, the prediction of the pressure distribution shows proper accuracy which confirms the effectiveness of the proposed model. Finally, the computed lift forces are illustrated in Fig. 11. In this figure, the measurements of Dinh [37] and the analytical results of Wu [38] displayed. As observed in Fig. 11, the current simulation displays proper accuracy for all the cavitation numbers.
5. Results and Discussion

A supercavitating blade section with deadrise angle of 120.2 degrees on the left side and 55 degrees on the right side (Fig.3) is investigated for different cavitation and Froude numbers (Table 2). The corresponding results are discussed in this section. Two different ambient pressures corresponding to two cavitation numbers are considered. First, it is considered that the ambient pressure is equal to the atmospheric pressure. It is then assumed that ambient pressure is equal to 1/7 of the atmospheric pressure. The later pressure is selected based on Olofsson’s [6] experimental results on SPP 821-b. During his experimental investigations for the ambient pressure below the stated value, the effects of cavitation number on the characteristic curves were proved to be significant. Through such approach, the effects of cavitation on the flow pattern, pressure, and forces may be determined easier. Also, three different $U/V$ ratios ranging from 0.23 to 0.64 are taken into consideration.

<table>
<thead>
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<th>Parameter</th>
<th>Value</th>
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</thead>
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<td>Velocity ratio $(U/V)$</td>
<td>0.23 - 0.64</td>
</tr>
<tr>
<td>Froude number</td>
<td>6</td>
</tr>
<tr>
<td>Cavitation Number</td>
<td>0.25, 2.3</td>
</tr>
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</table>

5.1. Ventilation Pattern

Ventilation regime on the water encountering the hydrofoil can be detected by the observed free surface profile and the recorded pressure on the foil's walls. Therefore, in this section, variation of these two parameters are computed for different $U/V$ ratios and presented in Fig.12. All the results in this sub-section are presented for the cavitation number of 2.3. As evident in Fig. 12, through an increase in $U/V$ (or increase in the advanced ratio), the free
surface on the cavity wall tends to get closer to the back side of the wedge. However, this phenomenon displays different conditions for different velocity ratios. At lower velocity ratio (high vertical wedge velocity), i.e. $U/V=0.23$, the fluid flow detaches from the leading edge and free surface pattern corresponding to the fully ventilated regimes appears. Based on Shiba’s observations [1] outlined in the literature, higher angle of attack of the hydrofoil causes higher pressure difference during the water entry and air is thus perfectly sucked into the low pressure region. The cavity peak point C (shown in Fig.14) is far enough from the wedge to balance the pressure in the cavity with the ambient pressure. The hydrodynamic pressure only acts on the face side of the foil and it has a steady behavior in this case (Condition I in Fig.12). Afterward, as $U/V$ increases ($U/V=0.38$), cavity peak point C is adhered to the back side of the wedge and the cavity is disconnected from the ambient air. This condition was earlier defined as the transition mode (Condition II in Fig.12). When the ventilated cavity collapses, the closed vapor cavity near the leading edge remains on the back side of the wedge and partially cavitation regime occurs. Accordingly, the pressure would have a steady condition and the resultant force substantially drops (Condition III in Fig.12).

Fig. 12. Pressure distribution and water surface profile for different values of $U/V$ at $Fn=6$ and $\sigma=2.3$. $Vt$ or $V.t$ (product of velocity and time is height) is the foil submergence at time t.

The horizontal and vertical forces corresponding to each value of $U/V$ are displayed in Fig. 13 which include the results in all three considered conditions or regimes and may help understand the effect of each of these conditions on the forces. As observed in Fig.13, through an increase in $U/V$ in both fully ventilated and partially cavitation conditions, dimensionless values of $F_x$ and $F_y$ are reduced. This is while the efficiency of the blade section increases. This may be attributed to the reduction of the spray height on the pressure side of the wedge and consequently the decrease in the wasted energy, as the velocity ratio increases. As a result, the efficiency should increase in both of these modes. However, there is an exception in the case of partially cavitation mode, where the impact velocity highly decreases and hence the wetted area increases considerably in such a way that the efficiency drops. Such phenomena can be seen for $U/V=0.64$. The main aspect and the most important fact which should be considered here are the resultant forces in the transition mode. In this mode, since the ventilated cavity is disconnected from the atmosphere, the pressure drops in this area (see pressure distribution and cavity profile for $U/V=0.38$ in Fig.12.II). Through this sudden drop, the difference of the pressure between the sides highly increases and the resultant force increases.
5.2. Effects of the ambient pressure

Fully Ventilation. A comparison is presented in Fig.14 between the pressure distribution and water surface profile of the fully ventilated condition \((U/V=0.23)\) for two cavitation numbers of \(\sigma=2.3\) and 0.25. A close scrutiny of this figure indicates that there is no significant difference between the free surface profiles.

This may be attributed to the fact that cavity pressure at both cavitation numbers is equal to the ambient pressures \((P_0: 1 \text{ atm}, P_0': 1/7 \text{ atm})\) and that there is enough space for the air in both cases to penetrate the ventilated area. Therefore, only pressure in the case of lower ambient pressure, drops. The pressure and water surface profile for a larger \(U/V\) (i.e. \(U/V =0.38\)) in the fully ventilated condition are shown in Fig.15. With a decrease in \(V\), the position of point C (shown in Fig.14) gets closer to the back side of the wedge. As a result, the air moving toward the foil’s leading edge exhibits a pressure drop at point C. At point C (hereby called “Critical C”), a particular region is established due to this pressure drop in which the separation of the air cavity is expedited. This phenomenon may block the path of air moving toward the leading edge faster which itself leads to a separation. If the ambient pressure is lower (e.g. \(\sigma =0.25\)), through a slight pressure decrease in the closed cavity region, the pressure may reach the vapor pressure and the cavity wall hence experiences cavitation. This cavitation causes an injection of vapor into the cavity and prevents point C from being further sucked down. This is why point C is located at an upper position for the case of lower cavitation number.
Probing into the effects of cavitation on hydrodynamic characteristics of surface piercing propellers through numerical modeling of oblique water entry of a thin wedge

Transition condition: As stated and explained earlier, with an increase in $U/V$ ratio, the flow may turn into another mode called transition mode, in which air is divided into two regions. In order to explore the effects of cavitation number on the free surface and pressure in this flow mode, these two factors are computed and displayed in Fig. 16 at $U/V=0.4$ for two different cavitation numbers. As mentioned in the previous subsections, there should be a large difference between the pressures ($\Delta P$) of both sides in this case. However, for the case of lower ambient pressure, as evident in Fig. 15, the vaporization does not allow the water surface to come down and adhere to the back side of the wedge. Therefore, the transition regime signs are not yet observed. In this phenomenon, the pressure difference ($\Delta P'$) in the case of lower cavity number, decreases.

Free surface pattern and pressure distributions of both considered ambient pressures for $U/V=0.48$ are shown in Fig. 17. Based on the obtained results, it may be concluded that the transition in the case of lower Cavitation number occurs with a delay, in comparison with the case of larger Cavitation number. Therefore, while the transition happens at $U/V\sim0.38$ for the case of $\sigma=2.3$, this phenomenon is shifted for the case of $U/V\sim0.48$ at $\sigma=0.25$. As observed in Fig. 17, the free surface in the case of smaller Cavitation number experiences the transition mode. However, pressure is the important aspect of this physical phenomenon. Although the case with smaller Cavitation number exhibits transition condition, the difference between pressures of both sides is smaller for this case, in comparison with that of larger Cavitation number.
Partially cavitation condition: As the velocity ratio increases (i.e. $V$ decreases), no ventilation occurs and only cavitation sheet is observed near the leading edge, in which case partially cavitation mode occurs. The free surface and pressure distribution for $U/V = 0.52$ are shown in Fig.18. Based on the free surface plots, it can be concluded that the free surface profiles for both cases are similar. According to the plots of pressure distributions, it may be claimed that pressures of the face sides for both cases are also similar and there is a downward shift, similar to the previous cases. However, on the suction side area, the generated vapor cavity in the case of lower ambient pressure, is longer than the case of higher ambient pressure.

Figure 19 illustrates the results for another $U/V$ ratio in a different partially cavitation case. The considered ratio corresponding to these results is $U/V = 0.64$. In this situation, it is also observed that free surface profile and pressure distribution are similar in both cases, except for the fact that pressure is shifted downward. Similarly, equivalent forces are expected to be produced in both cases which will be presented in the next subsection. Therefore, one may conclude that when $U/V$ ratio increases enough to avoid cavitation occurrence, the cavitation number would not be a significant contributor.
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Fig. 19. Pressure distribution (left) and water surface profiles (right) at U/V=0.64 and Fn=6.0 for cavitation numbers \( \sigma_n=0.25 \) and \( \sigma_n=2.3 \).

Effects of cavitation number on the performance curves: The computed forces as well as the efficiency are displayed in Fig. 20, as a function of \( U/V \). All the run conditions are related to Froude number of \( Fn=6 \) and \( U/V \) ratios ranging from 0.23 to 0.64. In the previous subsections, the pressure distribution and the effects of cavitation number were discussed. The results in Fig. 20 may help derive a general conclusion regarding the effects of cavitation number on the water entry process of supercavity blade section. Based on the force plots, it can be inferred that Cavitation number or better yet the ambient pressure may not significantly affect the forces in both fully ventilation and partially cavitation conditions. It is evident that magnitudes of the forces are approximately equal for both Cavitation numbers. However, for the transition condition, it is observed that the case with larger Cavitation number/ambient pressure have larger forces. This may be attributed to the intensive vaporization in the cavity area of the case with smaller cavitation number, which was explained earlier. It is also evident that transition condition of the case with smaller Cavitation number exhibits a delay, in comparison with the other case. Finally, the plots of the efficiency show that there is no remarkable change in efficiency for both of the considered cases. On the other hand, it can be stated that Cavitation number may not significantly affect the efficiency for the investigated velocity ratios. This may be attributed to the reduction of both horizontal and vertical forces in the case of smaller Cavitation number.

![Fig. 20. Forces (right) and efficiency (left) of the foil for two different cavitation numbers.](image)

6. Conclusions

In the current paper, numerical simulation of flow around a thin wedge is presented in order to provide a better understanding of the effects of Cavitation number and advance ratio/velocity ratio on the performance of surface piercing propellers in different conditions. Coupled URANS equation, VOF scheme, and Zwart’s cavitation model [33] are utilized to conduct the targeted numerical simulations.
The validity of the proposed model is assessed in two steps. First, the water entry of a wedge is modeled and the obtained results indicate good accuracy for the computed pressure and free surface elevation. Subsequently, cavitation phenomenon around a supercavitating hydrofoil is modeled. It is demonstrated that cavity length, pressure, and lift force are predicted with good accuracy.

The proposed model is then used to investigate the defined problem for two different Cavitation numbers. The produced results demonstrate that in fully ventilated condition, smaller ambient pressure may not affect the pressure distribution as well as the free surface profile. However, in the transition condition, it is observed that the case with smaller Cavitation number has lower pressure difference on the wedge’s sides, since vapor is produced. Also, the results of horizontal and vertical forces in this regime reveal that these forces reduce, as the ambient pressure decreases. It should also be noted that as the Cavitation number decreases, the transition mode appears at a larger velocity ratio. The results of efficiency coefficient indicate that a decrease in the Cavitation number may not affect this coefficient for the investigated range of velocity ratio.

Future studies may include numerical modeling of the fluid-structure interaction for these sections by considering both one-way and two-way coupling. Numerical modeling of the problem in three-dimensional condition may be considered another future task.

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References

Probing into the effects of cavitation on hydrodynamic 
characteristics of surface piercing propellers through numerical 
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FEASIBILITY OF INVESTMENT IN RENEWABLE ENERGY SYSTEMS FOR SHIPYARDS

UDC 629.5.081: 629.5.082
Professional paper

Summary

Shipbuilding industry is generally considered as one of the key global industries. It consumes a significant amount of energy and leaves a strong footprint on the environment. It is therefore required to try to increase renewable energy sources in the shipbuilding industry in accordance with the adopted international legislation. This will improve energy efficiency of shipyards and reduce their impact on the environment. The present preliminary study contributes to those goals as application of sustainable renewable energy systems in ship production industry is considered. The analysis is performed for solar panels, tidal and wind turbines with respect to their technology readiness and investment costs, as well as for the total energy consumption. While in the present work the research hypothesis is studied on example of a typical Croatian shipyard, the general findings are expected to be widely applicable to all shipyards. The solar panels are considered as the most suitable renewable technology, while the investment return period for introducing solar panels to satisfy the entire electrical energy demand in this particular case is approximately seven years. The impact of introducing this green shipbuilding approach is considered beneficial in comparison with current conventional electrical energy production based on conventional energy sources.

Key words: Shipyard design; Renewable energy; Solar energy; Green industry; Case study

1. Introduction

Shipbuilding is globally considered as one of the key industries. It focuses on a production of technologically very demanding and complex products that requires large energy consumption and makes considerable footprint on the environment. New measures related to reduction of carbon dioxide footprint are successfully adopted in Maritime Pollution (MARPOL) amendments based on Energy Efficiency Design Index (EEDI), [1]. This enables estimation of carbon dioxide emission for a specific ship with characteristic design parameters, [2]. The goal is to achieve energy efficient and less polluting equipment and engines, e.g. [3], which consequently leads to an improved ship design with respect to the environment. However, application of renewable energy sources and energy efficient concepts to enhance carbon dioxide reduction is quite rare in ship production industry. It is
therefore required to analyze this possibility on example of a typical shipyard, whereas the performed analysis yields an approach that is to be applicable in the global ship production industry.

The application of renewable energy sources in ship production industry is in line with United Nations (UN) policies for environment-friendly industry and greener industrial footprint initiative [4]. The main pathway towards sustainable industrial development is by rationally exploiting material and energy resources. An accomplishment of such a demanding and challenging task is quite complex, as it involves multiple stakeholder issues, e.g. industrial, financial and energy sector, local and state government, local community, investors, international agreements, which all need to be adequately addressed. The UN initiative, [4], is based on four fundamental green industry parameters considering environmental friendly approach in material, waste, water and energy management. This approach is based on adopting obligatory legal framework, as well as on environment-related labeling of low-carbon products.

This initiative is particularly concerned with optimal energy consumption to reduce electricity generated from conventional energy sources, i.e. fossil fuels. With this respect, an application of renewable energy sources, such as solar, offshore and wind energy, e.g. [5], is strongly recommended, as it decreases production costs, creates new business opportunities, prevents an increase in energy price, ensures energy independent and diverse production systems, and enhances product competitiveness and enterprise ranking, [6].

This approach is in line with the European Union (EU) mandatory 20% share in renewable energy sources in final energy consumption by the year of 2020, [7]. Given those facts the Croatian Government set three basic objectives, i.e. to develop secure, competitive and sustainable energy sector with emphasis on renewable energy sources, [8]. These recommendations are to be widely implemented in practice as to fully account for multiple benefits of this advanced approach for the energy and the environment. It is therefore the purpose of the present study to investigate a possibility of applying renewable energy sources in the shipbuilding industry.

A thorough literature survey reveals that the green shipbuilding concept has not been accordingly addressed in the global energy community so far. The studies that address complex interaction between general industry and energy management usually report on possible greenhouse gasses mitigation policies including continuous economic growth, environmental preservation, cost-benefit analysis of sustainable energy development as well as socioeconomic benefits and drawbacks, [9]. Various renewable energy scenarios were analyzed in [10] demonstrating health benefits as one of the most important socioeconomic indicators justifying the compliance costs associated with sustainable energy development. A concept of green industry should be considered in the context of the low-carbon society taking actions that are compatible with the principles of sustainable development, [11]. It is, therefore, necessary to demonstrate a high level of energy efficiency and application of the low-carbon production technologies, [12], as well as consumption and behavior patterns leading towards low levels of greenhouse gas emissions, [13].

A comparative policy scenario, exploring emission mitigation, renewable energy and energy efficiency in the United Kingdom and European Union (EU) industry using process-oriented modelling approach is presented in [14]. It demonstrates critical importance of energy-efficient, cost-efficient and long-term decarbonisation of the industrial production. In line with that, an impact of two possible green-growth scenarios assuming conventional and large-scale renewable energy development on China GDP until 2050 is analyzed in [15] using dynamic computable general equilibrium model. An assessment of energy consumption and pollution emissions in case of an iron and steel industry is addressed in [16] resulting with
performance indicators demonstrating inevitable necessity of rational and sustainable energy management for manufacturing industry. A similar study [17] reports an interaction between waste energy recovery an energy efficiency improvement in China’s iron and steel industry.

An energy supply chain in the context of resource, technology, environment, infrastructure and socio-economic development interaction is considered for manufacturing sectors in China, [18], where critical energy sectors are identified using input-output analysis. A more detailed literature review and case study of energy management for a typical process industry company is analyzed in [19] emphasizing the importance of such an approach in the industrial sector. Along with that, current situation, regulatory changes and promotion of renewable energy sources in Spain are addressed in [20] while pointing out the necessary policy instrument leading towards achievement of future renewable energy goals. Some of these goals are analyzed in [21] using the system thinking approach with particular focus on energy supply, diversification, energy security and support schemes of Nordic countries.

The scope of the present study is to analyze a possibility of introducing renewable energy sources in a ship production industry. This is performed on an example of a typical shipyard, while the performed analysis yields an approach that is expected to be globally applicable in the ship production industry. This methodology is therefore expected to enhance decoupling of shipyards from conventional energy sources, and decreasing an environmental footprint of ship production industry. To achieve this goal, the annual electrical energy consumption in the typical shipyard is analyzed to determine its energy demand. An overview of renewable energy sources that can be considered for this implementation is outlined. An optimal solution is selected by considering the impact on the environment, society and shipyard business.

2. Total electrical energy consumption for a typical shipyard

The total annual energy consumption is analyzed for a typical shipyard (Uljanik, Croatia). Typical ship production process is reported in Fig. 1. It consists of several subprocesses, i.e. a) metal process, b) outfitting, c) surface treatment, d) material handling, e) quality control and f) ancillary services. During these processes, various operations are performed, i.e. flattening, cleaning, conserving, cutting, forming, welding, machining, assembling, dislocating, transporting, testing, air-conditioning, ventilating, lightening as well as the operations of a general consumption. These actions are performed to exploit and modify physical, chemical and mechanical properties of the used material, [22].

Almost all technology operations directly use electrical energy, while some of them, like flattening, conserving and thermal cutting rely on heat produced by mixture of oxygen (O₂) and acetylene (C₂H₂) combustion. As relatively large amount of oxygen and acetylene is used during production process, shipyards are commonly equipped with separate facilities that produce those gases using electrical energy.

Along with shipyard core business, i.e. production process, the electrical energy is commonly consumed by ancillary services that enable operations like heating, air-conditioning and ventilation (in each workshop, working place and office), lightening and general consumption.

Nevertheless, some estimates of the electrical energy consumption can be found in [23] where an approximate electrical energy consumption of 750 kWh per ship lightweight ton is assessed with share of a) 18% for oxygen, b) 12% for carbon dioxide, c) 17% for acetylene, d) 19% for surface treatment, e) 21% for welding, and f) 13% for general consumption, Fig. 2.
While this overview provides guidelines about the structure of the total energy consumption for a typical shipyard, this statistics may vary for various shipyards depending on a particular production program, order book state, ship complexity and state of production lines. Nevertheless, the general approach developed for this typical shipyard may be well adopted for any shipyard by taking into account its particular electricity demand.
3. Renewable energy sources appropriate for ship production

Current development and application of renewable energy sources for ship production is mainly driven by several factors, i.e. increasing energy production demands, increasing awareness of conventional sources impact on climate change and strong ambition to exploit huge, unused and renewable energy potentials. In general, renewable energy sources are classified as bioenergy, solar energy, geothermal energy, hydropower, ocean energy and wind-energy sources, e.g. [5]. These renewable energy sources can be onshore and offshore.

Having in mind a rather complex nature of ship-production process, heavily exploited shipyard area and complex geographical placement, while assuming that renewable energy source is to be placed within the shipyard and managed by its personnel, only solar, ocean and wind energy sources are considered suitable. Therefore, the main properties of those particular renewable energy sources relevant for this study are briefly outlined.

3.1 Photovoltaic electricity generation

In comparison to other renewable energy sources, the solar energy is considered the cleanest and most abundant. It has an inexhaustible capacity of more than $1.5 \times 10^{18}$ kWh annually, [5]. It enables noiseless, carbon free and rather simple electricity production.

The most effective way to exploit solar energy is based on photovoltaic effect. This includes a direct conversion of solar input into electricity. Photovoltaic systems are composed of quite simple and easy-to-maintain devices that are widely applicable, from hand-size systems with output in microwatts up to large industrial and space applications and power plants with capacities in megawatts, e.g. [24]. Consequently, extensive experience in this field of renewable energy is gained mainly due to its steady development, its superior availability, cost effectiveness, capacity and efficiency, [25].

Commonly used technical solutions involve photovoltaic module, inverter, energy storage, energy consumer, and connection to electrical network to balance electricity deficiency and excess, Fig. 3. They can be arranged as stand-alone or hybrid systems combining solar power with other conventional or renewable energy sources. Light absorbing material is of particular importance for cell efficiency. Monocrystalline and multi-crystalline silicon technology is developed along with new concepts using amorphous and crystalline silicon, organic and thin film cells with typical efficiency of approximately 19%, e.g. [24].

The development in photovoltaic technology is accompanied by regulatory instruments, educational activities, as well as by financial and environmental initiatives aiming towards successful and rapid implementation of solar energy into energy systems.

It is estimated that solar electricity will have about 11% share of the world energy supply by the year 2050. This will positively influence photovoltaic industry resulting in a decrease in installation and maintenance costs, [26]. Current installation costs are estimated to be around 2.5 $/W with a goal to reduce them to 1 $/W until 2020 due to innovation and production development. Annual operative and maintenance costs for such a system are approximately 1.5% of the investment cost, [5].
The motivation for such a rapid development in high-end technology is obtained through international and national activities. They enable long-term targets and policies to be achieved by creating favorable and stable business climate that attracts investments in manufacturing, research and development activities. Some nations like Spain, Germany, United States, Brazil, Japan and China currently play a leading role in application of photovoltaic electricity generation with prominent implementations like building integrated systems as shading devices, [27], vertically positioned power sources in opaque building surfaces, [28], desalination plants, [29], Earth orbiting solar power satellite, [30], home and pumping systems, [31], and solar power plants, [32].

3.2 Ocean energy

Total annual energy capacity of ocean energy sources is estimated to be about $2 \cdot 10^{15}$ kWh, [5]. This is approximately 0.13% of the available solar energy. At this moment, ocean energy sources are barely harvested, mainly due to development and conceptual stage of this technology based on ocean energy converters typically sighted near the coast line or at the open sea, [33]. A typical system consists of hydrokinetic devices that transform sea current and wave kinetic energy into electricity. Some solutions make use of a salinity gradient and thermal energy of the sea. With respect to ocean energy sources, a development of these complex subsea structures with low production rate is currently addressed, e.g. [5].

At this point, research and development of hydrokinetic devices is dominated by small companies producing design solutions at high readiness level, [34]. Currently, there are three typical hydrokinetic devices, i.e. axial-flow, cross-flow, and oscillating systems. Their prototypes commonly have the power output up to 2 MW. Some locations where those hydrokinetic devices are used in water depths up to 35 m include United Kingdom, Ireland, Scotland, France, Spain, China, Japan, South Korea, Canada and United States, [35].

Axial and cross-flow hydrokinetic devices operate as lift-based systems similar to wind turbines and aircraft wings. Rotational motion about the axial or transversal axis is induced by torque as a combination of drag and lift forces acting on a blade composed of two-dimensional hydrofoil cross-sections, e.g. [36], Fig. 4. Current intensive development of lift-based hydrokinetic devices is based on favorable flow properties like a significant kinetic energy density across the rotor-disc area, simple manufacturing, handling, transport, installation and maintenance, as well as mitigation of visual pollution and sea routes.
What makes this renewable energy source particularly attractive is the fact that sea currents are highly predictable, which simplifies their design and ensures favorable working conditions. Some drawbacks are related to corrosive sea environment, sea fouling, non-uniform velocity profile of sea currents due to a friction between the sea flow and the sea bottom, as well as possible creating of underwater noise, e.g. [6].

Various types of oscillating systems are developed for a wide range of operating conditions in order to convert wave or flow vortex energy into electricity. They typically consist of three components, i.e. primary interface that interacts with fluid and transfers its energy to the second subsystems that incorporates a direct drive or serves as a short-term energy storage. The third interface converts energy by means of electromechanical processes, [5].

Wave energy converters differ from one another primarily with respect to the type of their interaction with the wave motion (heaving, surging, pitching), as well as with respect to the water depth and distance from the shore, [37]. Principal methods of energy conversion are oscillating water columns, oscillating body systems and overtopping devices, Fig. 5.

Oscillating water column devices, Fig. 5a, convert wave energy by means of varying air pressure induced by wave motion. Air flows through an air turbine, thus inducing its rotation. Consequently, this kinetic energy is transformed into electricity.

Oscillating body system, Fig. 5b, also known as the point absorber, transforms wave-induced heave motion into electricity by means of a relative motion among two bodies.

A prominent example of an energy storage conversion system is an overtopping device, Fig. 5c. Its basic operational principle is transformation of wave kinetic energy into potential by means of water accumulation. Once accumulated, the water is drained through the hydraulic turbine to induce its rotational motion.
An interesting concept of oscillating system based on vortex-induced vibration (VIV) is briefly outlined in [34]. Its operating principle is based on shedding vortices that form a characteristic von Kármán street in the wake of the body. This includes alternating low-pressure regions and consequently periodic lifting force acting as vibration excitation. This energy due to vibrations is harvested using linear power-take-off elements. An important advantage of this oscillating system in comparison with lift-based systems is its ability to exploit slow water flows with a sea-current speed as low as 0.4 m/s, [38].

Current research in generating ocean renewable energy by using hydrokinetic systems is focused on a) development of resource assessment methodologies, b) advanced turbine design with particular emphasis on efficient blade development, turbine wake and array modeling, and c) environmental and biodiversity impacts, [39]. A considerable discrepancy in investment price for hydrokinetic devices from 4500 $/kW to 14300 $/kW, [6] enables a wide range of practical solutions.

3.3 Wind energy

Global annual wind energy capacity is estimated at 1.662·10^{15} kWh, [5], which is approximately 0.11% of the available solar energy. Onshore and offshore wind-energy harvesting is predominantly developed in Denmark, Spain, Germany, Sweden, United Kingdom, and United States of America. Typical solution is horizontal-axis wind turbine that transforms wind flow into rotational motion and then into electricity. A typical power output of these wind turbines is up to 8 MW of installed power. This enables satisfying energy demand for more than 10,000 homes, [40]. Commonly installed power is between 2 MW and 4 MW, similar for both onshore and offshore wind farms.
A typical onshore and offshore wind-turbine structure consists of tower, rotor, and nacelle with installed power take-off system. While the onshore wind turbines are fixed to the ground, the offshore wind turbines can be fixed to the sea bottom or placed at the characteristic offshore platforms. For the fixed offshore wind turbine, the wind turbine is placed at a fixed monopile, gravity-base or jacket supporting structure. For the floating offshore wind turbine, the floating structures include spar, tension leg or semisubmersible supports, [41].

In comparison to an onshore wind turbine, an offshore wind turbine has some favorable properties like nearly constant average velocity at the hub height, weaker atmospheric turbulence, lower structural fatigue, possibility to install longer blades and consequently create larger power output. However, some drawbacks are mainly related to maintenance, sea loading and harsh sea environment, [42].

Once built, offshore wind turbine has to be successfully installed and maintained within harsh sea environment which turns out to be quite a demanding task, [43]. Moreover, its operation properties and output have to be continuously monitored for a long period that can be accomplished only by applying specific data integration framework optimized towards recovery rate improvement, [44]. It is also important to emphasize an increase of activities related to offshore wind farming in the Adriatic Sea, [45], which could act as a leaver significantly influencing a development of the Croatian economy, [46].

Current research is mainly focused on wind-turbine and wind-power-plant design and optimization, advanced telescopic and self-erecting towers, application of advanced materials enabling longer, lighter and more elegant blades that enhance efficiency and power output of wind turbines. An important and still open issues are related to wind characteristics around wind turbines placed in topographically complex terrain, [47], and in regions characterized by non-standard transient winds, [48], particularly in case of extreme weather conditions encountered during summer, [49], and winter, [50]. These untypical wind conditions can considerably increase structural fatigue of wind turbines and thus shorten their lifetime, while the power output of wind turbines in these transient wind conditions is smaller than it is the case for quasi-steady flow conditions.

A particular focus is on development of offshore floating wind turbines in order to decrease their installation and maintenances cost, develop methodologies to address coupled hydro-structural issues, develop turbine-transportation and offshore-assembly operations. The cost of onshore and fixed offshore wind turbines is estimated to be between 2000 $/kW and 5000 $/kW with annual operative and maintenance costs ranging from 170 $/kW to 350 $/kW, [5].

A comparison of costs for the solar, ocean and wind renewable energy sources is reported in Table 1.

4. The Croatian shipyard case study

Ship production industry in Croatia encompasses 5 large, 14 medium and 352 small shipyards that produce, maintain and repair ships and offshore structures. They employ approximately 20 000 personnel and create an annual income of approximately 1.3 billion $, [51]. Croatian shipyards are considered to be an industrial primemover, as they enhance regional community development with multiplicative factor of about 2.8, [52]. Moreover, 5% of the total Croatian industry income is generated by shipyards’ export, thus contributing annually with 0.8 billion $ to the national GDP.
Table 1 A comparison of considered renewable energy sources with respect to their commonly installed power and installation and maintenance costs, [5]

<table>
<thead>
<tr>
<th>Source</th>
<th>Commonly installed power, kW</th>
<th>Investment cost, $/kW</th>
<th>Annual maintenance cost, $/kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar</td>
<td>Photovoltaic modules</td>
<td>Microwatts to Megawatts</td>
<td>2500</td>
</tr>
<tr>
<td></td>
<td>Tidal turbine</td>
<td>2000</td>
<td>4500-14300</td>
</tr>
<tr>
<td>Ocean</td>
<td>Oscillating systems</td>
<td>750</td>
<td>6200-16100</td>
</tr>
<tr>
<td></td>
<td>VIV oscillating systems</td>
<td>NA</td>
<td>NA</td>
</tr>
<tr>
<td>Wind</td>
<td>Offshore turbine</td>
<td>2000-4000</td>
<td>2000</td>
</tr>
<tr>
<td></td>
<td>Offshore fixed turbine</td>
<td>2000-4000</td>
<td>2000-5000</td>
</tr>
</tbody>
</table>

The largest Croatian shipyards, and thus the largest energy consumers, are located in Croatian coastal cities of Pula, Rijeka, Split and Trogir, Fig. 6. They manufacture custom-made bulk carriers, chemical and oil tankers, car and wagon carriers, heavy lifters, passenger ferries, submarines, and dredgers. The Croatian shipbuilding industry, although relatively minor player at the global scene, plays a significant role in the total EU context with 27% of share in new orders, 14% in order book and 12% in deliveries.

Furthermore, a ratio between the total number of compensated gross tons (CGT) and the gross tons (GT) is approximately equal to one, thus indicating a high complexity of produced ships. It is important to mention that this matches the capability to produce the most complex ships of the world leading shipbuilding nations like China, Japan, and South Korea. An average annual production of Croatian shipyards is approximately 15 ships, [53]. At the moment, the entire production of Croatian shipyards is based on conventional energy sources.

This methodology is developed on example of a typical Croatian shipyard Uljanik, Fig. 1. It is based on the analyzed energy demands and possible renewable energy technology

Fig. 6 Geographical position and typical production program of large Croatian shipyards, [52]
implementations for shipyard production. In case it is anticipated that in average a typical Croatian shipyard annually produces up to 5 ships of 5000 lightweight tons, the total annual electrical energy consumption is approximately 19 GWh. For the purposes of this study, the total annual electrical energy consumption can be approximated with 20 GWh per year in accordance with e.g. [54].

Since some of the considered technologies are still insufficiently developed, the present case study is focused on application of mature solutions, i.e. photovoltaic modules, horizontal-axis tidal turbines and fixed offshore wind turbines. For this purpose, solar, sea current, and wind potentials are specified for the studied local environment.

Croatia is considered to be among the top insolated EU countries with approximately 4.2 kWh/m$^2$ to 5.2 kWh/m$^2$ of average daily insulation. The most significant solar potential in Croatia is at the Adriatic Coast. It is annually from 1.45 MWh/m$^2$ to 1.60 MWh/m$^2$, [45] and [55], for a horizontal insolated area, Fig. 7. The required area of a solar system is determined as a ratio between the energy demand, $E_D = 20$ GWh, and energy supply, $E_S = 1.45$ MWh/m$^2$, whereas the transformation efficiency $\eta = 0.19$ needs to be taken into account, [5].

In this case, the required area is approximately equal to 0.073 km$^2$, where approximately 36300 units of 2 m$^2$ solar panels can be placed. In general, solar panels in shipyards can be conveniently placed at flat building roofs. A single panel can therefore annually produce 0.551 MWh of energy.

An annual average of approximately 2000 sunny hours is taken into account, [56], whereas an installed single solar panel power yields about 275 W, i.e. 10 MW in total. Based on the required installed power, an investment cost is determined using the data reported in Table 1 as 25 million $ that is approximately 6.25 times more than the cost of the estimated

![Fig. 7 Average annual insolation for Croatia, adapted from [45]](image-url)
20 GWh of the annual energy consumption, in case a typical electrical energy cost in Croatia of approximately 0.2 $/kWh is considered, [57].

Therefore, taking into account the maintenance costs, a return period can be estimated to be approximately 7 years. Moreover, the return period can be additionally reduced if the Environmental Protection and Energy Efficiency Found (EPEEF) investment co-funding up to 40% of the expenses is included, [58].

A possibility for renewable energy production using hydrokinetic and aerokinetic devices is analyzed in [6] with respect to renewable energy potentials of sea currents and wind at the Adriatic Sea. It is estimated that a single horizontal-axis tidal turbine is able to annually produce approximately 4.5 MWh of electrical energy, whereas a wind turbine yields approximately 4 GWh for the same period. An estimated energy output is rather low, mainly due to small wind and sea current velocities that results with rather low efficiency of the installed capacities.

Consequently, an installation of horizontal-axis tidal turbines for annual production of 20 GWh does not seem to be a realistic solution, particularly if the installation cost of approximately 4450 turbines is considered.

On the other hand, the same energy can be produced using 5 wind turbines of installed power of 5 MW. In this case, the investment price is about 50 million $, while the return period is twice as long as it is the case for the solar panels. Moreover, further analysis of wind potential at the Adriatic Sea will result with more reliable data.

5. Impact on energy, environment and shipyard business

Application of renewable energy sources for purposes of ship production leads to implementation of green production principles promoted by UN. This approach promotes green and environment-friendly shipbuilding industry. As ship production is considered to be a significant industrial component and large energy consumer, an approach outlined in the present study contributes to reduction in greenhouse gas emission.

According to a current Eurostat database, manufacturing and electricity, gas steam and air conditioning in 2013 participate with 45.4% in total greenhouse gas emissions, [59]. Such a significant share can be effectively decreased by applying renewable energy sources in production process. Namely, according to National Energy Technology Laboratory, [60], CO₂ emission rates in case of electricity generation using coal and natural gas are approximately 1000 g/kWh and, respectively 450 g/kWh, yielding between 9000 and 20000 t less CO₂ emission annually only due to application of renewable energy sources in the analyzed Croatian shipyard (Uljanik).

Along with that, one of the immediate factors of introducing green shipbuilding concept is increasing the shipyard profitability, which is strongly enhanced by such an approach. Application of renewable energy sources in shipyards significantly improves their energy efficiency, sustainability, and operational effectiveness (particularly in case of transport) with a final consequence of an increased profitability.

A decreased shipyard share in greenhouse gas emission promotes its significance at a local, state and global level, particularly as such activities are generally well accepted in public. Therefore, a ship produced using renewable energy is to be considered as a novel product, which is completely designed, developed and manufactured without a significant change in current ship production procedures. In such a way, shipyards will gain modern and competitive production program which, will be attractive for ship owners and investors.
Based on the outlined benefits, a shipyard has an opportunity to strengthen an advance its position during negotiation with national or supranational governments and investors. A set of anticipated immediate and mediate effects is reported in Table 2.

Table 2 A summary of immediate and mediate benefits of introducing renewable energy sources in shipbuilding process

<table>
<thead>
<tr>
<th>RENEWABLE ENERGY IN SHIPBUILDING PRODUCTION</th>
<th>IMMEDIATE BENEFITS</th>
<th>MEDIATE BENEFITS</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Profit</td>
<td>Public perception</td>
</tr>
<tr>
<td></td>
<td>Energy efficiency</td>
<td>New product</td>
</tr>
<tr>
<td></td>
<td>Operational effectiveness</td>
<td>Competitive product</td>
</tr>
<tr>
<td></td>
<td>Sustainability</td>
<td>Negotiations with investors</td>
</tr>
</tbody>
</table>

6. Conclusion

Shipbuilding industry is globally considered as one of the key industries. It has a significant electrical energy demand that is required for production purposes. In such a way the shipbuilding industry considerably and adversely contributes to the overall CO₂ emission and global warming. To diminish this adverse effect on the energy and the environment, various measures with respect to energy efficient equipment and ship engines are adopted. Nevertheless, a serious lack of initiatives in view of energy consumption in ship production process and its environmental impact is noticed.

The present study thus focuses on developing an approach that outlines possibilities with respect to introducing sustainable renewable energy systems in ship production industry. While this hypothesis is studied on example of a typical Croatian shipyard, the general findings are expected to be widely applicable by taking into account particularities of any other shipyard.

The analysis is performed for solar panels, tidal and wind turbines. The focus is on their technology readiness, investment costs, and the total energy demand. For the studied case, the solar panels are selected as the most suitable renewable technology with the investment return period of approximately seven years.

The analysis clearly outlines beneficial effects of the green shipbuilding approach on the energy, environment and the shipyard business in general, while there is a number of other favorable immediate and mediate effects, e.g. profitability, sustainability, energy efficiency, operational effectiveness, positive public perception, new and competitive products, favorable position in deal making with investors.

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Hull Lines Reliability-Based Optimisation Design for Minimum EEDI

Summary

Pointing to marine greenhouse gas emissions, the EEDI (Energy Efficiency Design Index), a mandatory regulation, has become a significant factor in the design of ships’ hull lines. EEDI is closely related to many design parameters of ships’ hulls, which were conventionally set to be constant when involved in the design. However, it is often the case that considerable parameter fluctuations happen during actual navigation (such as travelling speed, draft, etc.), so it is more reasonable to state the important parameters as random variables. The reliability and quality level requirements of design results are also of concern. Accordingly, a reliability-based optimisation design (RBOD) method is introduced in this research. Furthermore, the design of experiments and reliability analyses via a Monte Carlo simulation and four reliability methods are employed to measure the sensitivity of the design variables and their reliability. Upon comparison with deterministic optimisation design (DOD) via adaptive simulated annealing (ASA), ROBD shows excellent adaptability and reliability in minimum EEDI ship hull lines’ designs.

Key words: EEDI; Reliability-based optimisation; Ship hull lines; Sigma level

1. Introduction

The Energy Efficiency Design Index (EEDI) is a significant index for the energy efficiency of naval architecture, and it has been a mandatory regulation for all newly built ships since January 1, 2013 [1]. EEDI expresses the environmental cost, stated as the vessel’s CO₂ discharge, which is generated by the social benefit of each tonnage unit (quantity of shipments). The compulsory execution of EEDI accelerates the pace of energy saving and emission reductions in the shipping business, and higher requirements are proposed for the development of ‘green’ ships. It is well recognised that EEDI is closely related to ships’ hull lines, which can greatly affect their sailing performance and other features. Therefore, along with the development of new ship hull lines, reducing EEDI and trends toward energy savings are becoming increasingly important.
Currently, a number of internal and external parameters (such as resistance, travelling speed, draft, etc.) are normally set as constants in the design of hull lines (2014) [2,3]. However, the fluctuation of these parameters is unavoidable during actual navigation [4], which leads to the inevitable generation of errors if they are always considered as constant values. Although errors are small in most cases, during the optimisation design process, continuous iteration and mutual couplings may cause large deviations in the final results, and greater risks to quality come into view. Apparently, it is more reasonable to set such parameters as random uncertainty variables during the hull lines design process, so that their effect on target responses can also be considered.

In recent years, during ships’ preliminary design stages, the influence of uncertain parameters has gradually come to be taken more seriously and subject to greater levels of research. Diez (2009) [5][6] took the lead in considering the influence of uncertain parameters on ship design: the principal dimensions of container ships were optimised using robust design (RD) and robust optimisation design (ROD). Afterwards, Diez’s team (2011–2015) [7-10] introduced RD and ROD to ship hull simulation based design (SBD) systems, and a series of studies were conducted. Their design objective was developed from principal dimensions to hull lines, single uncertain parameters to multiple ones, and speed disturbances to wave responses, which reflected the development idea of 'from coarse to fine, from simple to complex'. Other works, such as that by Hannapel (2010) [11], introduced uncertain factors to constraint conditions, and the expectations and variances of objective functions were also considered. Papanikolaou and colleagues (2014) [12] made a review of methods on the uncertainties in the estimation of wave loads, and they studied ships’ responses while in operation; in a later case, reliability assessment methods (FORM, SORM, MC) were used to account for the variations in ship and environmental parameters. The common characteristic of the above studies was that uncertain parameters were introduced into the fixed optimisation system as a priori knowledge, but the results’ quality and reliability were not evaluated. Had they been, it would have led the design results to being of low quality with a high failure probability when applied to the further detailed designs.

From a statistical point of view, reliability-based optimisation design (RBOD) (2016) [13-15] considers the uncertainty of constraints as probabilistic at the beginning of the design process. By means of reliability analyses and quality optimisation, RBOD can yield the high quality results that strictly meet the requirements of performance, reliability and constraints. RBOD works by finding the flat area in a design space that minimises the output fluctuation caused by uncertain parameters and meets all quality requirements and probabilistic constraints.

In summary, with respect to previous approaches, Beyer (2007) lists the key factors of different definitions as follows [16]:

- Robust design (RD): Aiming at the worst case, attention is mainly on variances or standard deviation;
- Robust optimisation design (ROD): An optimisation process considering uncertainty in the objective or target function;
- Reliability-based optimisation design (RBOD): The attention is focused on the constraints of the design, which are treated as probabilistic inequalities and give a statistical feasible region.

While RD and ROD are mainly focused on expectations or variances of the objective function, RBOD concentrates on the probabilistic handling of constraints. Thus, RBOD can be a fine method for solving the way to approach quality requirements and probabilistic constraints. In this study, principal dimensions and ship form coefficients are combined to form the design space, and constraints are stated as hull moulded volume and resistance; thus,
an optimisation model for minimum EEDI as the target is established. After the deterministic optimisation design (DOD) via adaptive simulated annealing (ASA), the design of experiments and reliability analyses via Monte Carlo simulation and several reliability methods are implemented to measure the sensitivity of design variables and the reliability of the optimisation results. Then, the RBOD is conducted for the final minimum EEDI hull line, which could well meet the quality requirements and probabilistic constraints.

2. EEDI Calculation

2.1 EEDI formula and parameter

EEDI is expressed by the ratio of CO\(_2\) emissions and quantity of shipments, and is related to the ship’s fuel consumption, engine power, auxiliary power, energy-saving equipment, speed, tonnage and other factors [2]. Its calculation formula is shown as in Eq. (1):

\[
EEDI = \frac{E_{ME} + E_{AE} + E_{PTI} + E_{eff}}{f_i \cdot \text{Capacity} \cdot v \cdot f_w}
\]

Where:

\[
E_{ME} = \left( \prod_{j=1}^{n} f_j \right) \left( \sum_{i=1}^{n_{ME}} P_{ME(i)} \cdot C_{FME(i)} \cdot SFC_{ME(i)} \right)
\]

\[
E_{AE} = P_{AE} \cdot C_{FAE} \cdot SFC_{AE}
\]

\[
E_{PTI} = \left( \prod_{j=1}^{n} f_j \sum_{i=1}^{n_{PTI}} P_{PTI(i)} - \sum_{i=1}^{n_{eff}} f_{eff(i)} \cdot P_{AEeff(i)} \right) C_{FME} \cdot SFC_{ME}
\]

\[
E_{eff} = \left( \sum_{i=1}^{n_{eff}} f_{eff(i)} \cdot P_{eff(i)} \cdot C_{FME} \cdot SFC_{ME} \right)
\]

And where: \(E_{ME}\) is CO\(_2\) discharge of main engine, and \(E_{AE}\) is of auxiliary engine, \(E_{PTI}\) is shaft belt device, \(E_{eff}\) is energy-saving equipment. \(C_F\) is carbon conversion coefficient, \(v\) is travelling speed, \(kn\); \(SFC\) is fuel consumption rate in 75% rated power, \(g/(kW \cdot h)\); \(Capacity\) is deadweight tonnage, \(t\); \(P\) is power for main or auxiliary engine, \(kw\); \(f_j\) is modifying factor for ship special design, \(f_i\) is modifying factor for ice strengthened ship, and is taken 1.0 for non ice strengthening; \(f_{eff}\) is innovation factor, and is taken 1.0 for waste heat recovery unit, and for other energy or technology, it should be evaluated by classification society; \(f_w\) is wind wave correction factor.

Parameters in this research are taken as in table 1.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Magnitude</th>
<th>Parameters</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>(SFC_{ME})</td>
<td>190 (g/(kWh))</td>
<td>(SFC_{AE})</td>
<td>215 (g/(kwh))</td>
</tr>
<tr>
<td>(P_{ME})</td>
<td>75% (MCR_{ME})</td>
<td>(f_i, f_j, f_w)</td>
<td>1.0</td>
</tr>
<tr>
<td>(P_{AE})</td>
<td>(0.025 MCR_{ME} + 250 (MCR_{ME} \geq 10000 kW))</td>
<td>(P_{AEeff}, P_{PTI}, P_{eff})</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>(0.05 MCR_{ME} ) ((MCR_{ME} &lt; 10000 kW))</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Upon substitution of magnitudes in Tab.1 into Eq. (2), the simplified EEDI formula is obtained as Eq. (3):
\[
EEDI = C_p \times \frac{142.5MCR_{ME} + 215(0.05MCR_{ME})}{\text{Capacity} \times v} = C_p \times \frac{0.5144 \times 153.25R}{\text{Capacity} \times PC}
\]

Where: \(MCR_{ME}\) is power rating of main engine, (kW); \(R\) is travelling resistance, (kN); \(PC\) is propulsive coefficient; \(Capacity\) is deadweight, (t), got by Eq. (4):

\[
\begin{align*}
\text{Capacity} &= \Delta - W_i = \rho V - W_h - W_o - W_m \\
&= \rho V - (C_h + C_o)L(B + D) - W_m
\end{align*}
\]

Where: \(\Delta\) is displacement, (t). \(W_i\) is light weight, (t), which is divided into hull weight \(W_h\), outfit weight \(W_o\) and electromechanical weight \(W_m\). \(W_h\) and \(W_o\) can be estimated by the principal dimensions: waterline length \(L\) (m), waterline width \(B\) (m) and depth \(D\) (m). Correspondingly, \(C_h\) and \(C_o\) are relevant coefficients, which are usually empirical valued. Electromechanical weight \(W_m\) can be set as a fix value referred to parent type.

A large cargo ship is taken as the example in this study, detailed descriptions of which are shown later, according to the routine statistical data of this ship type [17], constant coefficients are defined as: \(C_h=0.43\), \(C_o=0.31\), \(W_m=671(t)\) in this study.

2.2 Travelling resistance \(R\)

Numerical method based on the slender body theory such as Michell, is frequently-used in the ship resistance evaluation because of its simple assumptions and fast capabilities. However, such approach is considered out-dated and probably not accurate enough to capture the effect of small hull form modifications on the vessel resistance.

As a semi-empirical method for practical ship hull form, Holtrop method [18] with 334 towing test regressed data in ocean engineering basin in Netherlands, could get satisfactory resistance results at preliminary design stage. Because of the iterative computation in Holtrop, time cost of optimisation would increased substantially. Thus approximate model is necessary for the optimisation, which has strict requirement for the length of each step, and is also the typical and frequently-used way in ship simulation based design (SBD) process.

BP (Back Propagation) neural network is a kind of multilayer feed forward network with error back propagation algorithm [19], and becomes one of typical approximation technologies because of its excellent ability to approximate nonlinear function, Eq. (5) represents a three-layer BP neural network model which using tangent sigmoid as transfer function of neurons:

\[
BPNN_3: O_i = \sum_{j=1}^{F} W_{ij} \tan h \left( \sum_{k=1}^{K} W_{jk} \tan h \left( \sum_{n=1}^{N} W_{kn} \xi_n + b_{lk} + b_{2j} \right) \right) + b_{3i}
\]

Where: \(\xi_n\) is input variable, \(O_i\) is output variable, \(W_{kn}, W_{jk}, W_{ij}\) are the weights of the layers between neurons, \(b_{lk}, b_{2j}, b_{3i}\) are thresholds of neuron unit in each layer.

BP neural network approximate model, which is very sensitive to the internal parameters, needs mounts of simulation results as inputs, thus errors of output would appear due to some uncontrollable factors. Although this error or uncertainty has a small value in most cases, large deviation of the whole system could also be generated by continuous iterative computation. Therefore, considering the uncertainty of approximate model has an important significance.

3. Deterministic optimisation

3.1 Design variables

This study takes a large cargo ship as the example. In the optimisation design, design variables are often identified by principal dimensions, such as waterline length \(L\), waterline
width $B$, and the modification of the hull shape which can be represented by the original data points. Multiplied hull modification functions are as shown in Eqs. (6)-(7):

$$y_f(x, z) = y_f(x_0) \cdot \omega(x, z)$$

$$y_a(x, z) = y_a(x_0) \cdot \omega(x, z)$$

$$\omega(x, z) = 1 - \sum_{m} \sum_{n} A_{mn} \sin \left[ \pi \left( \frac{x-x_0}{x_{max}-x_0} \right)^{m+2} \right] \cdot \sin \left[ \pi \left( \frac{z-z_0}{z_{max}+z} \right)^{n+2} \right]$$

Where: $y_f(x, z)$ represents forward (or after) half of the hull points after changed, both in the mid ship-section of the interface; $\omega(x, z)$ is modification function of hull form; $A_{mn}$ is control factor for hull shape, in this research $m, n=1,2,3$.

### 3.2 Constraints

In order to ensure that the internal space of hull is not significantly reduced, and the travelling performance maintains acceptable, changing of molded volume and travelling resistance is taking as constraint condition:

$$\begin{align*}
\nabla &\geq \nabla_0 \\
R &\leq R_0
\end{align*}$$

Where: $\nabla$ and $\nabla_0$ are optimal and initial hull form’s molded volume, which can be calculated via “Simpson method” by the hull points; $R_0$ is initial hull form’s resistance.

### 3.3 Deterministic Optimisation Design (DOD) model and result

After integrating the constraints into the optimisation objective, minimum EEDI optimisation model is established, as in shown in Tab.2:

<table>
<thead>
<tr>
<th>Tab.2 Deterministic optimisation model for minimum EEDI hull lines</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Objective:</strong></td>
</tr>
<tr>
<td>- Minimum EEDI</td>
</tr>
<tr>
<td><strong>Design variables:</strong></td>
</tr>
<tr>
<td>- Waterline length $L$, Range: [110, 130], Initial value 120 (m)</td>
</tr>
<tr>
<td>- Waterline width $B$, Range: [21, 25], Initial value 23 (m)</td>
</tr>
<tr>
<td>- Hull control factor $A_{mn}(m,n=1,2,3)$, Range: [0, 0.15], Initial value [0.05,0.05]</td>
</tr>
<tr>
<td><strong>Constraints:</strong></td>
</tr>
<tr>
<td>- Minimum molded volume $\nabla \geq \nabla_0 = 15627$ (m$^3$)</td>
</tr>
<tr>
<td>- Maximum resistance $R \leq R_0 = 483$ (kN)</td>
</tr>
<tr>
<td><strong>Constants:</strong></td>
</tr>
<tr>
<td>- Travelling speed $v$, 14(ks)</td>
</tr>
<tr>
<td>- Carbon coefficient $C_r$, 3.5</td>
</tr>
<tr>
<td>- Propulsive coefficient $P_C$, 0.6</td>
</tr>
<tr>
<td>- Draft $T$, 7.45 (m), Depth $D$, 15.3 (m)</td>
</tr>
<tr>
<td><strong>Optimisation technique:</strong></td>
</tr>
<tr>
<td>- Adaptive Simulated Annealing (ASA)</td>
</tr>
</tbody>
</table>

BP neural network, with its great ability to improve the computational efficiency, is adopted here to establish the approximate model of $R$. When training neural networks, a large number of experimental design points (sample points) should be distributed in the design space. The accuracy of neural network is directly affected by the number of and of the distribution of sample points. Meanwhile, too much sample points cost a high computational cost. To compromise this contradiction, and to ensure the uniformity of sample distribution,
uniform design method is employed here: uniform design table of 11 factors and 16 levels \( U_{16}^m(16^1) \) is used and 16 sets of experimental program are generated. In order to scatter the sample points evenly, 50 sub-schemes are randomly generated in the range of input variables set by each experimental program, therefore, a total of 800 schemes were generated.

The parameters of BP neural network are chosen as: layer number is 3, and the number of neurons in the input layer, the middle layer and the output layer are all \((12, 6, 1)\), convergence threshold is \(1 \times 10^{-6}\). Adaptive learning rate method is used as the training method here, with an additional momentum factor, which improves speed and ensures that the network does not fall into local minima.

As an exploratory technique, Adaptive Simulated Annealing (ASA) [20] algorithm, which is very well suited for solving highly non-linear problems with short running analysis codes, when finding the global optimum is more important than a quick improvement of the design, is introduced here to do the optimisation.

The optimisation process is shown in Fig.1:

![Fig.1 Deterministic optimisation process for minimum EEDI](image)

The main internal parameters of ASA are set to: Max number of generated designs is 500, relative rate of parameter annealing is 1.0, and convergence epsilon is \(1e^{-8}\). After 520 iterations, optimal solution is obtained at the 506th step. The result is shown in Tab.3, and the objective function values curve in optimisation process is shown in Fig.2.

<table>
<thead>
<tr>
<th>Variables</th>
<th>Illustration</th>
<th>Initial scheme</th>
<th>DOD result (by ASA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(L(m))</td>
<td>Waterline length</td>
<td>120.0</td>
<td>123.6</td>
</tr>
<tr>
<td>(B(m))</td>
<td>Waterline width</td>
<td>23.0</td>
<td>24.5</td>
</tr>
<tr>
<td>(A_{mn}) ((m,n=1,2,3))</td>
<td>Hull control factors</td>
<td>[0]_3×3</td>
<td>([0.0018 0.0132 0.089 0.102 0.033 0.001 0.010 0.0072 0.0301])</td>
</tr>
<tr>
<td>(\nabla(m^3))</td>
<td>Molded volume</td>
<td>15627</td>
<td>15690</td>
</tr>
<tr>
<td>(R_t(kN))</td>
<td>Resistance</td>
<td>483</td>
<td>452</td>
</tr>
<tr>
<td>(EEDI)</td>
<td>EEDI</td>
<td>18.63</td>
<td>17.69</td>
</tr>
</tbody>
</table>
4. Sensitivity analysis

In order to analyze the influence of variable changing on outputs, it’s necessary to take the sensitivity analysis of design variables and important constants to the constraints and optimisation objective. The sensitivity analysis execution plan is shown in Fig.3.

![Fig.2](image1.png)  
**Fig.2** Objective function values curve in deterministic optimisation design process

One experiments design technique is introduced here: Latin hypercube, because its engineer has total freedom in selecting the number of designs to run. 1500 points are generated for the Latin hypercube. Pareto contributions of inputs are shown in Fig.4, which can reflect the effect degree of each input on each output.

![Fig.3](image2.png)  
**Fig.3** Sensitivity analysis execution plan

![Fig.4](image3.png)  
**Fig.4** Pareto contribution of inputs to outputs
As is shown in Fig. 4, input factors such as $L$, $B$, $T$, $v$, $C_F$, $PC$ and their mutual operation customarily have a great effect on the three outputs. Nevertheless, effect of hull control factor $A_{mn}$ ($m,n=1,2,3$) is less important than others. In order to reduce the computational complexity, it’s necessary to extrude the main factors and ignore the unimportant factors for the following reliability analysis and robustness design. Thus through sensitivity analysis, input factors: $L$, $B$, $T$, $v$, $C_F$, and $PC$ are taken into account as significant factors to the outputs response.

5. Reliability analysis

In the actual design process, input factors are often subject to interference and reflect the uncertainty, which is usually represented in the form of random variables. Because random disturbance of parameters is neglected in deterministic optimisation, the optimal solution is easy to be near or exceed the "edge position" of feasible region. Once disturbance occurs, deterministic optimal solution is likely to fall into the infeasible region, thus its reliability is hard to be guaranteed.

Reliability is the probability that the performance of result satisfies the constraint. Quality level is a comprehensive index to measure design quality, and two most typical quality levels are “3σ level” and “6σ level”. The relationship between reliability and quality level is shown in Fig. 5. For hull lines preliminary design, the 3σ level of 99.73% reliability is sufficient.

![Fig.5 Quantization of reliability and quality level](image)

To analysis the reliability and quality level of deterministic optimal result in Sec.2.3, Four typical reliability analysis methods are introduced here:

(1) Monte Carlo Simulation (MCS)

In MCS, the probability distribution of random variables is known. Through random sampling, the probability distribution of system response can be estimated and the contribution of each random variable to the response results can also be obtained. Illustration of MCS is as shown in Fig. 6.

![Fig.6 Illustration of Monte Carlo Simulation (MCS)](image)
There are two sampling techniques in MCS: simple random sampling and descriptive sampling. Compared to the former, descriptive sampling is a variance reduction technique aimed at reducing the variance of the statistical estimates derived from the population data. On the other hand, descriptive sampling can ensure the quality of statistical analysis with less sampling times and simulation, so it becomes a representative method and is used in this study.

(2) Mean Value Reliability Method (MVRM)

As a probabilistic method, MVRM utilizes the Taylor’s series expansion of failure functions at mean values of the random variables. MVRM is the most efficient of the reliability analysis methods in terms of the number of function evaluations, or simulation program executions. However, unless the failure functions are linear or quadratic with normally distributed random variables, mean-value reliability index would lose accuracy. Illustration of MVRM is shown in Fig. 7.

(3) First Order Reliability Method (FORM)

FORM takes advantage of the desirable properties of the standard normal probability distribution, and reliability index is defined as the shortest distance from the origin of the standard normal space to a point on the failure surface. Mathematically, determining the reliability index is a minimization problem with one equality constraint.

Mean value and standard deviation of respond output of FORM is as Eq. (9)

\[
\begin{align*}
\mu_y &= F(\mu_i) \\
\sigma_y &= \sqrt{\sum_{i=1}^{m} \left( \frac{\partial F}{\partial \chi_i} \right)^2 (\sigma_i)^2}
\end{align*}
\]

(9)

Where: \( x_i \) is random variable.

(4) Second Order Reliability Method (SORM)

When failure function for MPP (Probable Point Most) is nonlinear, SORM can get more accurate in the surface failure curvature of the approximate evaluation than FORM.

Mean value and standard deviation of respond output of SORM is as Eq. (10)

\[
\begin{align*}
\mu_y &= F(\mu_i) + \frac{1}{2} \sum_{i=1}^{m} \frac{d^2 F}{d\chi_i^2} (\sigma_i)^2 \\
\sigma_y &= \sqrt{\sum_{i=1}^{m} \left( \frac{\partial F}{\partial \chi_i} \right)^2 (\sigma_i)^2 + \frac{1}{2} \sum_{i=1}^{m} \sum_{j=1}^{m} \left( \frac{\partial^2 F}{\partial \chi_i \partial \chi_j} \right) (\sigma_i)^2 (\sigma_j)^2}
\end{align*}
\]

(10)

Problem model of FORM is shown in Fig. 8.
Reliability analysis model of deterministic optimal result is shown in Tab. 4.

**Objective:**
- Considering uncertainty of variables, the reliability and quality level of EEDI deterministic optimal result in Tab.3 are evaluated.

**Random design variables:**
- Waterline length $L$, Normal, $\mu=120.0$, $\sigma=1\%*\mu$
- Waterline width $B$, Normal, $\mu=23.0$, $\sigma=1\%*\mu$
- Hull control factor $A_{mn}(m,n=1,2,3)$, Constants=$A_{mnopt}$ in Tab.3

**Random Noises:**
- Travelling speed $v$, Normal, $\mu=14$, $\sigma=1\%*\mu$
- Carbon coeffic $C_F$, Normal, $\mu=3.5$, $\sigma=1\%*\mu$
- Propulsive coefficient $PC$, Normal, $\mu=0.6$, $\sigma=1\%*\mu$
- Draft $T$, Normal, $\mu=7.45$, $\sigma=1\%*\mu$

**Quality Constraints:**
- Minimum molded volume $\bigtriangledown \geq 15627$
- Maximum resistance $R_0 \leq 483$

**Analysis Method**
- MCS, MVRM, FORM, SORM

Because an accurate modeling of input distributions can be simply established or replaced from observed data, as is shown in Tab.4, normal distribution is assumed for all the random parameters (both the design variables and the noises) here. Furthermore, assessing probability density functions from observed data is outside the scopes of the present work and, therefore, no further addressed.

The optimisation objective EEDI and two constraints: molded volume and resistance are evaluated by the above four reliability analysis methods. Under the random disturbance of design variables and noises, their probability distributions are shown in Fig. 9.
Reliability analysis results of molded volume, total resistance and EEDI are shown in Tab.5. Therefore, there are slight differences between the four reliability analysis results: expectations $\mu$ of MCS, FORM and SORM are basically the same, and the standard deviations $\sigma$ of MVRM, FORM and SORM are basically the same. Because MVRM needs the linear or quadratic failure functions, its analysis accuracy is less than others consequently.

Tab.5 Reliability analysis results

<table>
<thead>
<tr>
<th></th>
<th>MCS</th>
<th>MVRM</th>
<th>FORM</th>
<th>SORM</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\nabla$</td>
<td>$\mu=15584$;</td>
<td>$\mu=15498$;</td>
<td>$\mu=15584$;</td>
<td>$\mu=15584$;</td>
</tr>
<tr>
<td></td>
<td>$\sigma=0$;</td>
<td>$\sigma=0$;</td>
<td>$\sigma=231$;</td>
<td>$\sigma=231$;</td>
</tr>
<tr>
<td></td>
<td>Reliability:</td>
<td>Reliability:</td>
<td>Reliability:</td>
<td>Reliability:</td>
</tr>
<tr>
<td></td>
<td>100%;</td>
<td>98.34%;</td>
<td>99.995%;</td>
<td>99.995%;</td>
</tr>
<tr>
<td></td>
<td>Quality Level: 8</td>
<td>Quality Level: 2.39</td>
<td>Quality Level: 4.171</td>
<td>Quality Level: 4.171</td>
</tr>
<tr>
<td>$R_t$</td>
<td>$\mu=489$;</td>
<td>$\mu=489$;</td>
<td>$\mu=489$;</td>
<td>$\mu=489$;</td>
</tr>
<tr>
<td></td>
<td>$\sigma=5.73$;</td>
<td>$\sigma=6.71$;</td>
<td>$\sigma=6.71$;</td>
<td>$\sigma=6.71$;</td>
</tr>
<tr>
<td></td>
<td>Reliability: 52.821%;</td>
<td>Reliability: 52.821%;</td>
<td>Reliability: 52.756%;</td>
<td>Reliability: 52.821%;</td>
</tr>
<tr>
<td></td>
<td>Quality Level: 0.719</td>
<td>Quality Level: 0.719</td>
<td>Quality Level: 0.718</td>
<td>Quality Level: 0.719</td>
</tr>
<tr>
<td>EEDI</td>
<td>$\mu=18.98$</td>
<td>$\mu=18.21$,</td>
<td>$\mu=18.98$,</td>
<td>$\mu=18.98$,</td>
</tr>
<tr>
<td></td>
<td>$\sigma=1.005$</td>
<td>$\sigma=1.239$</td>
<td>$\sigma=1.239$</td>
<td>$\sigma=1.239$</td>
</tr>
</tbody>
</table>

Tab.5 shows that, the reliability of $R_t$ appears to be around 50%, and quality levels are not reached 1, that demonstrates the reliability of $R_t$ is extremely low. In order to improve its reliability and quality level, it’s necessary to conduct the reliability optimisation design.
6. Reliability-based Optimisation Design

Reliability-based optimisation design (RBOD) is to find a reliability result, which is far away from the failure surface. In the case of uncertain parameters, RBOD can effectively reduce infeasible probability, thus to improve the design result’s reliability. Illustration of reliability optimisation is shown in Fig.10, and flow-process diagram of RBOD is shown in Fig.11.

![Illustration of reliability-based optimisation design](image)

**Fig. 10** Illustration of reliability-based optimisation design

![Flow-process diagram of reliability-based optimisation design (RBOD)](image)

**Fig. 11** Flow-process diagram of reliability-based optimisation design (RBOD)

Mathematical model of reliability-based optimisation problem is expressed as in Eq.(11):

\[
\begin{align*}
\text{Minimize} & \quad F(\mu(X), \sigma(X)) \\
\text{Subject to} & \quad G(\mu(X), \sigma(X)) < 0 \\
X_{\text{LSL}} + \Delta X & \leq X \leq X_{\text{USL}} - \Delta X 
\end{align*}
\]

(11)

Where: \( X \) is random variables; \( F \) and \( G \) is target function and constraints function of RBOD, which are defined by expectation \( \mu \) and standard deviation \( \sigma \); \( \pm \Delta X \) is the fluctuation region of \( X \); \( X_{\text{LSL}} \) and \( X_{\text{USL}} \) are lower bound and upper bound of design variables.

Target function \( F \) can be decomposed into two parts: “Mean on target” and “Minimize variation”: 


\[ F = \sum_{i=1}^{l} \left[ \frac{\omega_1}{S_{1i}} (\mu_i - M_i)^2 + \frac{\omega_2}{S_{2i}} \sigma_i^2 \right] \]  

(12)

Where: \( i \) is component index of performance parameter set; \( M \) is expected average performance target; \( \omega_1 \) and \( \omega_2 \) are weights of expectation \( \mu \) and standard deviation \( \sigma \), \( S_1 \) and \( S_2 \) are normalization coefficients of \( \mu \) and \( \sigma \).

Mathematical constraints \( G \) in Eq.(14) can be converted to quality constraints \( G_q \) which are expressed as “Sigma level”, that can ensure the results quality in a specific sigma level. Quality constraints include: expectation \( \mu \), standard deviation \( \sigma \), and sigma level number \( n \):

\[
\begin{align*}
G_q = \mu + n\sigma & \leq \text{Upper Limit} \\
G_q = \mu - n\sigma & \geq \text{Lower Limit}
\end{align*}
\]  

(13)

In Eq.(12), sigma level number \( n \) represents strict level of result quality requirement. When \( n=\pm 3 \), namely “3σ Design”, which reliability is 99.73% according to Fig.5, and is usually considered as the acceptable result quality. When \( n=\pm 6 \), that is “6σ Design”, which reliability is 99.9999998%, and can effectively reduce the per-million failure probability of results. As its huge time cost and high non-convergence risk, “6σ Design” is not used here.

The quality level and reliability of “3σ Design” are enough for hull lines design in the ship preliminary design stage, and is adopted in this paper for the minimum EEDI hull lines RBOD.

Considering the uncertainty of variables, deterministic optimisation result in Sec. 2.3 is taken as the initial optimisation scheme. Aiming at finding the minimum \( \mu \) and \( \sigma \) of EEDI, and also meeting the 3σ quality level of constraints, RBOD is conducting as follows. As its excellent accuracy evaluation performance, Second Order Reliability Method (SORM) is taken as the reliability analysis method. RBOD model of minimum EEDI hull lines is shown in Tab. 6.

<table>
<thead>
<tr>
<th>Tab.6 The minimum EEDI hull lines RBOD model</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Objective:</strong></td>
</tr>
<tr>
<td>- Minimum ( \mu ) and ( \sigma ) of EEDI, namely ( f_{\mu+0.01\sigma} )</td>
</tr>
<tr>
<td><strong>Random design variables:</strong></td>
</tr>
<tr>
<td>- Waterline length ( L ), [110, 130]. Initial: Normal, ( \mu=120 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td>- Waterline width ( B ), [21, 25]. Initial: Normal, ( \mu=23 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td>- Hull control factor ( A_{mn} ) ( (m,n=1,2,3) ), [0, 0.15]. Initial: Constants( A_{\text{opt}} ) in Tab.3</td>
</tr>
<tr>
<td><strong>Random noise:</strong></td>
</tr>
<tr>
<td>- Travelling speed ( v ), Normal, ( \mu=14 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td>- Carbon coefficient ( C_F ), Normal, ( \mu=3.5 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td>- Propulsive coefficient ( PC ), Normal, ( \mu=0.6 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td>- Draft ( T ), Normal, ( \mu=7.45 ), ( \sigma=1%*\mu )</td>
</tr>
<tr>
<td><strong>Constraints:</strong></td>
</tr>
<tr>
<td>- Minimum molded volume ( \geq 15627 ), Quality level( \geq 3\sigma )</td>
</tr>
<tr>
<td>- Maximum resistance ( R \leq 483 ), Quality level( \geq 3\sigma )</td>
</tr>
<tr>
<td><strong>Design Method:</strong></td>
</tr>
<tr>
<td>- ASA+SORM</td>
</tr>
</tbody>
</table>

After conducting of RBOD, comparison of deterministic optimisation result and RBOD result is shown in Tab.7. As an important parameter in hull form design, \( L_{CB} \) (length between floating center and after-perpendicular) is given to indicate the hull longitudinal deformation, as in Tab.7.
As is shown, the optimisation objective EEDI of RBOD is slightly worse than the DOD result, but the two constraints: molded volume and total resistance are more far away from the failure surface. Therefore, RBOD result has stronger immunity for the disturbance of uncertain factors.

**Tab. 7 Results comparison of two optimal design method**

<table>
<thead>
<tr>
<th>Variables</th>
<th>Initial scheme</th>
<th>DOD result</th>
<th>RBOD result</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L (m)$</td>
<td>120.0</td>
<td>123.6</td>
<td>121.9</td>
</tr>
<tr>
<td>$B (m)$</td>
<td>23.0</td>
<td>24.5</td>
<td>23.3</td>
</tr>
<tr>
<td>$A_{mn}$</td>
<td>[0, 0.01]</td>
<td>[0.0018, 0.0132, 0.089, 0.102, 0.033]</td>
<td>[0.0016, 0.0101, 0.103, 0.102, 0.045]</td>
</tr>
<tr>
<td>$L_{CB} (m)$</td>
<td>55.32</td>
<td>54.01</td>
<td>54.90</td>
</tr>
<tr>
<td>$V (m^3)$</td>
<td>15627</td>
<td>15690</td>
<td>15631</td>
</tr>
<tr>
<td>$R_t (kN)$</td>
<td>483</td>
<td>452.3</td>
<td>467.8</td>
</tr>
</tbody>
</table>

Reliability: 99.995%; Quality Level: 4.171
Reliability: 99.999%; Quality Level: 4.683
Reliability: 52.821%; Quality Level: 0.719
Reliability: 99.833%; Quality Level: 3.590

Under the normal disturbance of design variables and noises, probability distributions and quality levels comparison of DOD and RBOD are as shown in Fig. 12 and Fig. 13, which are analysis by SORM.
For hull lines comparison, section lines of the initial scheme, DOD result and RBOD result are showed in Fig.14. Apparently, the DOD result has the greatest shape changing (especially at the parallel middle body) and RBOD’s change is comparatively placed in the middle, and bulbous bow of RBOD has some greater improvements. In conclusion, the RBOD method introduced in this paper can get a more reliability hull lines result and without great changing of the conventional DOD result.

7. Conclusion

This paper considered the problems of hull lines design for the minimum EEDI in consideration of the uncertain parameters. Based on reliability-based optimisation design technology, a 3-sigma design target for quality level was conducted, which had more reliability for the actual environment disturbance. This paper researched theoretically the reliability analysis and optimisation method, and some critical assessments could be got:

(1) Both the reduction of resistance and increase of molded volume could cause deduction of EEDI, as is the goal of this paper. But the two are very contradictory, thus in finally results, as in Tab.7, the changes of molded volume are insignificant, while changes of resistance are greater. Introduction of some designer preferences to the optimisation process may be a better approach to reconcile this contradiction.
As an operation indicator, parameters of EEDI do not change during operation and EEDI is actually fixed when shipping. In order to get the best adaptability of EEDI, in the hull lines design stage, which has the important effect on EEDI, designers need to consider a variety of marine environmental uncertainty factors, not limited to this research.

RBOD method can get a more reliability hull lines and without great changing of the conventional DOD result and initial, thus the reliability-based ship hull design has the feasibility and effectiveness. Further work could conduct from more reliability constraints and higher quality level for hull lines design and may other aspects of naval architecture.

8. Acknowledgement

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REFERENCES


A STUDY OF MOTION CHARACTERISTICS LED BY CONNECTION METHODS AND POSITIONS OF A WAVE-ENERGY CONVERTER IN A REGULAR WAVE

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Original scientific paper

Summary

The potential of wave power as an alternative energy resource is being studied to address problems associated with fossil fuel exhaustion and environmental pollution. In this paper, to improve the power generation efficiency of a floating-type wave-energy converter that has an activating body, the effects of the positions of a connecting bridge and different connecting methods between a main body and the activating body of the wave-energy converter were studied. In order to research the activating body’s motion characteristics that are caused by the changes of connecting bridge’s position and connecting methods; hinged or fixed connector, the wave-energy converter was modeled and simulated by using a commercial software. The moment and angular velocity of the axis of power generation were measured from the results of simulations and then the power outputs were calculated based on the moment and angular velocity. The outputs, which were analyzed under several regular wave conditions, were compared to each other.

Key words: Wave energy converter; Regular wave; Activating body; Power generation;

1. Introduction

Wave power has recently been added to the list of new and renewable energies, such as wind, solar, tidal, and hydropower, to solve the problems of fossil fuel exhaustion and environmental pollution attributed to fossil fuels. Many researchers have tried to develop and improve technologies, such as wave energy converters with a piston or flapper, to convert wave energy [1, 2]. In general, there are three stages of conversion of wave power generation. The first is primary conversion, which converts the potential energy and kinetic energy generated by the periodic motions of waves into dynamic pressure heads or mechanical motions; the second is secondary conversion, which converts the dynamic pressure heads or mechanical motion into energy for power generation; and the third is the conversion of the
power produced by the power generator [3]. According to the conversion methods, wave-energy converters can be divided into oscillating water chamber (OWC) types [4], over-topping device (OTD) types [5], and wave-activating body (WAB) types [6]. Of these wave-power generation methods, the WAB types have the highest generation efficiency. This is because this type’s wave-power generations convert the kinetic energy and potential energy of waves into mechanical motions without any steps. In contrast, OWC types use the airflow in water chamber or water tanks caused by sea surface motion to produce kinetic energy, which is then converted to mechanical motion. OTD types use differences of the dynamic pressure head between average sea surfaces and stored water above structure that overflow the OTD structure. In WAB type wave-energy converters, relative motions between the activating body and the main body of the converter lead to power generation. Thus, the shape of the activating body and the methods used to connect the main body and the activating body are important to enhance the efficiency of power generation [7].

It is evident that the places where are deep-sea regions or have higher waves is suitable to install a wave-energy converter because these areas have much larger wave energy than other common places[8]. However, since installing a wave-energy converter in these suitable places requires awful lot of money, to reduce an installation expense, floating-type converters are usually applied rather than fixed-type converters. In this study, the floating-type was applied. Furthermore, the position of a connecting bridge and the methods used to connect the connecting bridge to an activating body were studied to improve the power generation efficiency of a floating-type wave-energy converter. The power generation efficiency does not include efficiency derived from a generator. Instead, it was focused that mechanical efficiency caused by the position and methods of connecting bridge. To be specific, as the relative motion, which is affected by hydrodynamic interaction of floating structures[9], between the activating body and the main body mainly governs the efficiency of power generation, two factors, which affect the relative motion, were investigated: the positions of connecting bridge located on activating body, and the connecting methods between the connecting bridge and activating body(i.e., hinged or fixed). Simulations were used to compare and analyze the power output, moments, and angular velocities to understand the effects of two factors. In general, many studies have studied a new energy converter design, which has a distinctive converting mechanism[10], or case study of an energy converter: arrays of energy converters, or install areas of converters but there are few studies related to the connection methods and positions. Therefore, the main purpose of this paper is not to suggest a new model that has higher generation efficiency than a previous model but to find out the facts that how the connection methods and positions influence the generation efficiency of the floating-type wave-energy converter. The facts expected to be utilized as a basic data of a wave energy converter. Thus, a simple configuration and single wave direction, which turns towards the front of the floating-type wave-energy converter, was used.

2. WAB-type wave-energy converter

2.1 Basic shape

The concept of design of the wave-energy converter used in the present study is shown in Figure 1, and the principal dimensions and mass property of the main body and activating body of the wave-energy converter are shown in Table 1. The wave-energy converter is made up of a main body which generates the power, an activating body, and a connecting bridge. The simple design of wave-energy converter was selected to focus on comparison between two methods of connecting bridge. The H-shaped body type of the main body was chosen to easily set mooring system and to get better stability. A centroid of gravity of the box type activating body was moved down to improve the stability of activating body. An optimization
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process for the shapes of both bodies was not carried out but remain for a next step of research. Although the main body and the activating body are close, there is enough space, which are 1.25 m in each breadth direction and 1.25 m in length direction, between them. When incident waves cause heave and pitch motions of the activating body, the resultant kinetic energy is transferred to Joint 1 on the main body as mechanical rotational energy through Joints 2 and 3 on the connecting bridges. This mechanical rotational energy is used to generate power.

![Conceptive design of the wave-energy converter](image)

**Fig. 1** Conceptive design of the wave-energy converter

**Table 1** Principal particulars of the floating type wave-energy converter

<table>
<thead>
<tr>
<th>Factor</th>
<th>Main body</th>
<th>Activating body</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>30.35</td>
<td>10.00</td>
<td>m</td>
</tr>
<tr>
<td>Breadth</td>
<td>40.00</td>
<td>22.50</td>
<td>m</td>
</tr>
<tr>
<td>Depth</td>
<td>8.50</td>
<td>9.00</td>
<td>m</td>
</tr>
<tr>
<td>Draft</td>
<td>8.50</td>
<td>5.00</td>
<td>m</td>
</tr>
<tr>
<td>Mass</td>
<td>3,459.30</td>
<td>1,153.10</td>
<td>ton</td>
</tr>
<tr>
<td>Radius of gyration about X axis($k_x$)</td>
<td>23.25</td>
<td>6.57</td>
<td>m</td>
</tr>
<tr>
<td>Radius of gyration about Y axis($k_y$)</td>
<td>12.77</td>
<td>3.06</td>
<td>m</td>
</tr>
<tr>
<td>Radius of gyration about Z axis($k_z$)</td>
<td>26.01</td>
<td>7.11</td>
<td>m</td>
</tr>
</tbody>
</table>
The hydrostatic results of main body and activating body are shown in Table 2, which consists of center of gravity, center of buoyancy, \( GM_T \): Transversal metacentric height, and \( GM_L \): Longitudinal metacentric height. To support these values, global axis and two local axis of simulations are shown in Figure 2. A left figure of the figure 2 shows the global axis that is a basic coordinate system for the simulation, and a right figure displays both the main body’s local axis and the activating body’s local axis. The local axis are based on a centroid of gravity of each structures.

![Fig. 2](image)

**Fig. 2** Axis of simulations

<table>
<thead>
<tr>
<th>Factor</th>
<th>Center of Gravity (x, y, z)</th>
<th>Center of Buoyancy (x, y, z)</th>
<th>( GM_T )</th>
<th>( GM_L )</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main body</td>
<td>-5.17, 0, -2.859</td>
<td>-5.17, 0, -3.52</td>
<td>36.31</td>
<td>9.48</td>
<td>m</td>
</tr>
<tr>
<td>Activating body</td>
<td>6.25, 0, -3.64</td>
<td>6.25, 0, -2.50</td>
<td>9.57</td>
<td>2.80</td>
<td>m</td>
</tr>
</tbody>
</table>

If the activating body was made up of a single material, its centroid of gravity would be too high. Thus, when the activating body faces a wave, the activating body would be overturned in simulations because its pitch motion became too high, and the activating body did not have enough strength of stability. To avoid this phenomenon, the centroid of gravity was lowered by installing a high-density mass in limited area from bottom to 40% of depth of activating body as shown in Figure 3. This made the activating body stable for the pitch motion.

![Figure 3](image)

**Figure 3** Weighted area of the activating body
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The main body and the activating body were connected by connecting bridges and hinged or fixed connections to only allow heave and pitch motion between main body and activating body. As the relative motions between the main body and the activating body also affect the efficiency of power generation, translational motions and rotational motions of the main body were restrained by using four tensioned mooring lines.

2.2 Connecting bridges

Figure 4 shows the details of connecting bridges and activating body. “A” shows the distance between the location where connecting bridge was installed on the activating body and the edge of activating body, and red circles depict Joints 1, 2 and 3, respectively. “B” and “C” explains the length of connecting bridges. The dotted bar between Joint 3 and the centroid of gravity of activating body($G_A$) is the imagined connection which will use to explain the relation between motion of the activating body and angular velocity of the joints. Joint 1 connected to the main body. By increasing the distance of “A”, the length of “C” is also increased while the length of “B” is identical. In the present study, the simulations were conducted with six configurations (Table 3). In each configuration, the figure of “A” was changed, and the two connecting methods between “B” and “C” were also changed. The shape of the activating body was identical in all the configuration. Details of initial connection of the configurations are shown in dotted box of Figure 4.

Relation between motion of the activating body and angular velocity of the joints could be expressed as follows.

$$V_{G, rel} = \dot{\theta}_1 \vec{k} \times \vec{r}_{J_2/J_1} + \dot{\theta}_2 \vec{k} \times \vec{r}_{J_3/J_2} + \dot{\theta}_3 \vec{k} \times \vec{r}_{G_A/J_3} \quad (1)$$

Where, $V_{G, rel}$ is relative velocity of the activating body to the Joint 1, $\vec{r}_{b/a}$ is relative position from the joint b to the joint a, $\vec{k}$ is a unit vector.

The connecting methods consist of “Fixed” and “Hinged”. “Fixed” restrains all kinds of motion of activating body while “Hinged” allows rotational motion of activating body, such as a pitch direction motion. The configurations of the connecting bridge used in the simulations are shown in Table 3. Therefore, the Eq.(1) can be modified as follows. When the join 3 is fixed, case 1, 2, and 3, the third term in the right hand side of eq.(1) is left out. On the other hand, when the joint 2 is fixed, case 4, 5, and 6, the second term is left out. This means that although a joint is suppressed, the other joints can be moved. As the two joints and the
activating body simultaneously rotate and move, it is difficult to find out a direct effect of the motion of the activating body on the angular velocity of the joint 1. It is assumed that the friction, stiffness, and damping effects of the joints are negligible.

Table 3 Configurations of the connecting bridge

<table>
<thead>
<tr>
<th>Case</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>Joint 2</th>
<th>Joint 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3 m</td>
<td>6.5 m</td>
<td>10.5 m</td>
<td>Hinge</td>
<td>Fixed</td>
</tr>
<tr>
<td>2</td>
<td>5 m</td>
<td>6.5 m</td>
<td>12.1 m</td>
<td>Hinge</td>
<td>Fixed</td>
</tr>
<tr>
<td>3</td>
<td>7 m</td>
<td>6.5 m</td>
<td>13.8 m</td>
<td>Hinge</td>
<td>Fixed</td>
</tr>
<tr>
<td>4</td>
<td>3 m</td>
<td>6.5 m</td>
<td>10.5 m</td>
<td>Fixed</td>
<td>Hinge</td>
</tr>
<tr>
<td>5</td>
<td>5 m</td>
<td>6.5 m</td>
<td>12.1 m</td>
<td>Fixed</td>
<td>Hinge</td>
</tr>
<tr>
<td>6</td>
<td>7 m</td>
<td>6.5 m</td>
<td>13.8 m</td>
<td>Fixed</td>
<td>Hinge</td>
</tr>
</tbody>
</table>

Remarks: Joint 1 locates at x=-3.15m, y=0m, z=3.3m from the origin of global axis

2.3 Mooring lines

Figure 5 depicts four mooring lines. The mooring system imitates the TLP (Tension Leg Platform) system and therefore the mooring lines were placed under the bottom surface of main body and moored the main body to the sea-bed. In order to prevent the main body from moving, such as translational and rotary motion, the group of mooring lines had pre-tension that accounts for about 23% of the main body’s displacement[11]. Although the mooring lines relieve the motion of main body, it does not mean absolutely suppress. Therefore, the main body has slight translation and rotation motion which affect on power output. The mooring system was considered as a linear spring system. Therefore, mooring lines’ stiffness ($K_m$) were calculated by Eq.(2)

$$K_t = K_s + K_m$$

Where $K_t$ is the moored main body’s stiffness, $K_s$ is the non-moored main body’s stiffness. $K_t$ was calculated by nature frequency equation, Eq.(3)

$$\omega_h = \frac{2\pi}{T_h} = \sqrt{\frac{K_t}{m}}$$

$$K_t = m \left( \frac{2\pi}{T_h} \right)^2$$
Where $\omega_h$ is a natural frequency of the moored main body for heave motion, $T_h$ is a natural period of the structure, and $m$ is a mass including added mass of the moored main body. In the TLP system, natural period of heave motion is considered less than 5 sec., and in this paper, natural period of heave motion was considered 2 sec[11]. $K_s$ based on water plane and was calculated by Eq.(4)

$$K_s = \rho g A_w$$ (4)

Where $\rho$ is the density of sea water, $g$ is the acceleration of gravity, and $A_w$ is the area of water plane.

Four mooring lines’ properties and physical characteristics were shown in Table 4.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moored main body’s stiffness</td>
<td>$K_t$</td>
<td>34,249,441 N/m</td>
</tr>
<tr>
<td>Non-moored main body’s stiffness</td>
<td>$K_s$</td>
<td>7,179,448 N/m</td>
</tr>
<tr>
<td>Four mooring lines’ stiffness</td>
<td>$K_m$</td>
<td>27,069,993 N/m</td>
</tr>
<tr>
<td>Natural period for heave motion</td>
<td>$T_h$</td>
<td>2 sec</td>
</tr>
<tr>
<td>Mass</td>
<td>$m$</td>
<td>3,470,194 kg</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho$</td>
<td>1025 kg/m$^3$</td>
</tr>
<tr>
<td>The acceleration of gravity</td>
<td>$g$</td>
<td>9.81 m/s$^2$</td>
</tr>
<tr>
<td>The area of water plane of main body</td>
<td>$A_w$</td>
<td>714 m$^2$</td>
</tr>
</tbody>
</table>

Fig. 5 Details of the mooring lines
3. **Analysis of the wave-energy converter in regular waves**

The commercial simulation tool AQWA was used to carry out the motion simulation of the wave-energy converter. The AQWA is a widely used numerical tool and is capable of performing fully-coupled hydrodynamic assessment for various types of floating structures. A lot of studies have utilized the AQWA for a wide range of their applications to validate numerical dynamic analyses [12]. Wang et al. used the AQWA to investigate the maximum mooring tension of FPSO [13], and Kim et al. scrutinize design of the dual-buoy wave energy converter through the AQWA [10]. The assumption of AQWA is that the incident wave acting on the body is a harmonic wave and has a small amplitude compared to its length. Furthermore, in order to use potential flow theory, the fluid is assumed to be ideal, incompressible, and irrotational [14].

![Simulation process diagram](image)

**Fig. 6** Simulation process

Figure 6 shows the simulation process and the AQWA modules used in each process. Design Modeler was used in the modeling of the wave-energy converter. AQWA Diffraction was used to distribute meshes over the wave-energy converter and to calculate wave force on the meshes and the wave-energy converter.

3.1 **Mathematical model**

The wave force is divided into Froude-Krylov force, diffraction force, and radiation force and can be written as follows [15].

\[
F = [(F_I + F_d) + \sum_{k=1}^{6} F_{rk}x_k]
\] (5)
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Where, $F_I$ is the Froude-Krylov force, $F_d$ is the diffraction force, $F_r$ is the radiation force, and $k$ is degrees of freedom, from 1 to 6, $x_k$ is translational and rotational motion of a center of gravity of the body. The Froude-Krylov force at a point $\vec{X} = [X, Y, Z]$ can be written as

$$F_I = -i\omega \rho \int_{S_0} \varphi_I(\vec{X}) n dS \quad (6)$$

Where, $\omega$ is a wave frequency, $\rho$ is density, $S_0$ is the mean wetted surface of the body, $\varphi_I$ is incident wave potential. Moreover, the diffraction force can be calculated as follows.

$$F_d = -i\omega \rho \int_{S_0} \varphi_d(\vec{X}) n dS \quad (7)$$

Where, $\varphi_d$ is diffraction wave potential. The radiation force can be explained as follows.

$$F_{rk} = -i\omega \rho \int_{S_0} \varphi_{rk}(\vec{X}) n dS \quad (8)$$

Where, $\varphi_{rk}$ is radiation wave potential induced by $k$-th motion. Therefore, the wave potential can be written as

$$\varphi(\vec{X})e^{-i\omega t} = \left( \varphi_I + \varphi_d + \sum_{j=1}^{6} \varphi_{ij} x_j \right) e^{-i\omega t} \quad (9)$$

In the multiple structures case, hydrodynamic interaction is considerable. Therefore, the wave potential can be modified to consider hydrodynamic interaction as follows[15].

$$\varphi(\vec{X})e^{-i\omega t} = \left( \varphi_I + \varphi_d + \sum_{m=1}^{M} \sum_{j=1}^{6} \varphi_{rjm} x_j \right) e^{-i\omega t} \quad (10)$$

Where, $M$ is a number of structures. $\varphi_{rjm}$ is the radiation potential due to the unit $j$-th motion of the $m$-th structure while other structures remain stationary. Furthermore, 6-degree of freedom(DOF) rigid body motion can be explained as follow.

$$(M_s + M_a)\ddot{x} + C\dot{x} + Kx = F \quad (11)$$

Where, $M_s$ is mass matrix of structure, $M_a$ is added mass matrix, $C$ is damping matrix, $K$ is stiffness matrix, $F$ is the wave force, and $x$ is response motions.

A sensitivity test was performed to determine the size of mesh. The test investigated the convergence of Response Amplitude Operator (RAO) of the activating body in the heave direction with maximum sizes of mesh with which were set 0.5m, 1m, 2m, 3m, 4m, and 5m. The
results of the sensitivity test are shown in figure 7. The figure 7 merely exhibits the peaks of RAO to see difference of RAOs well. When the maximum mesh size gets smaller, the variation among RAOs also gets smaller. Finally, the RAOs with the maximum mesh size 0.5m and 1m were almost similar. The discrepancy of areas of under RAO between the 0.5m and 1m was about 0.1%. Therefore, 1m maximum size of mesh was set to distribute mesh over the activating body for the simulation. Figure 8 shows the distribution of the meshes on the wave-energy converter.

**Fig. 7** Sensitivity test of mesh size with heave RAO of activating body

After the modeling, the motion RAO of the activating body was calculated with AQWA Diffraction, and the frequencies, which lead various heave motion respond, of regular waves for time-based simulations were selected based on the motion RAO. The mooring system for the wave-energy converter and the connection methods between the activating body and the main body were set in the AQWA Time response which was used for the time-based simulation. After the simulation, the angular velocity and the moment on Joint 1 were analyzed to calculate the power generation.

**Fig. 8** Example of the mesh distribution on the wave-energy converter
3.2 Viscous effects

As this analysis process is based on the potential theory, viscous effects are disregarded. Therefore, viscous damping elements, which are called disc and utilized to add viscous effects in AQWA[16, 17], were utilized. The disc’s centroid is a point of action of viscous drag force, and summation of each four discs’ area is equal to area of each numbered bottom surface[18]. The discs, which are displayed in the figure 9, were placed under the bottom surfaces, from one to four, and each surface had four discs. Each discs were assigned one quarter of the total area of the surface from one to four as shown in the right side of the figure 9, and a direction of all discs was vertical. The direction means a direction of viscous drag force. Finally, a viscous drag coefficient was set with 1.05, and this value is equivalent to a common cube shape’s viscous drag coefficient [19].

Fig. 9 The viscous damping elements

3.3 Wave conditions

The wave conditions for the simulations were regular waves, the amplitudes were set to 1 m in all cases, as shown in Table 5, and the wave direction was set to 180° in all cases, so that the waves faced the front of wave-energy converter. Four wave frequencies were selected based on the RAOs, which is shown in the figure 10, of the main body and activating body for the heave and pitch directions. \( \xi_a \) and \( k_w \), which were used to make non-dimension of RAOs in the Figure 10, are wave amplitude and wave number, respectively.

Table 5 Wave conditions in the simulations

<table>
<thead>
<tr>
<th>Wave</th>
<th>Frequency</th>
<th>Wave length</th>
<th>Amplitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave 1</td>
<td>0.824 rad/s</td>
<td>90.77 m</td>
<td>1 m</td>
</tr>
<tr>
<td>Wave 2</td>
<td>0.984 rad/s</td>
<td>63.65 m</td>
<td>1 m</td>
</tr>
<tr>
<td>Wave 3</td>
<td>1.144 rad/s</td>
<td>47.09 m</td>
<td>1 m</td>
</tr>
<tr>
<td>Wave 4</td>
<td>1.304 rad/s</td>
<td>36.24 m</td>
<td>1 m</td>
</tr>
</tbody>
</table>
The selected waves have comparatively large wave length, which are shown in Table 5, in comparison with the length of activating body. However, high response motions of Heave and Pitch on both bodies were considered while selecting the wave frequencies. Moreover, the selected frequencies are slightly deviated from the frequencies which have the peak of RAOs to avoid resonance phenomenon.

3.4 Calculation of the power output

The WAB type wave-energy converter designed in this study exploits relative motions between the activating body and the main body. The relative motions are transferred to the joint 1 with the form of rotational energy through the other joints on the connecting bridges, and the rotational energy is used for power generation. The rotational energy is affected by both the activating body’s surge, heave, and pitch motions and the main body’s surge, heave, and pitch motions. The power output \( P_G \) can be obtained by multiplying the moment and the angular velocity at Joint 1. Thus, the power output as a result of the simulations was calculated through Eq. (12)[20].

\[
P_G = \frac{1}{T} \int b\dot{\theta}_1 dt \text{ (kW)}
\]  

(12)

Where \( T \) is total simulation time: 150sec., \( b \) refers to the moment in Joint 1, which is hinge connected between the main body and the connecting bridge, and \( \dot{\theta}_1 \) refers to the angular velocity of the connecting bridge in Joint 1.

3.5 Capture width

In order to evaluate the performance of the wave energy converter, capture width was calculated through Eq. (13). It is a one of parameters that is used to measure the efficiency of a wave energy converter. Therefore, by using capture width, it is possible to decide that which setting is more useful and efficient[21]. As a capture width had length dimension, wave number was multiplied to change length dimension into dimensionless[22].
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\[
C_W = \frac{P_G}{P_W}
\]  

(13)

Where \( C_W \) is capture width, \( P_G \) is Power generated by the wave energy converter, and \( P_W \) is the wave energy transport and calculated through Eq. (14).

\[
P_W = \frac{1}{2} \rho g \xi_a^2 C_g
\]  

(14)

Where \( \rho \) is the density of seawater, \( g \) is the acceleration of gravity, \( \xi_a \) is the amplitude of wave, and \( C_g \) is the group velocity of wave.

4. Results of the simulations

Connecting methods can be divided into the fixed or hinged connections in the connecting bridge on the activating body. The position of the connections can also differ. Figures 11–12 are parts of the simulations which would show the motion characteristics of the activating body when Joint 2 was hinged or when Joint 3 was hinged, respectively. The thick arrow in the front of the activating body indicates the direction of the incident waves, and the main body was fixed with mooring lines, which are shown under the main body.
Figure 11 shows the hinged connection at Joint 2 and the fixed connection at Joint 3. The activating body shows pendulum movements centering on Joint 2, in addition to heave and pitch motions occurred because of the incident waves. These motions made the connecting bridge connected to the main body moved back and forth, thereby affecting the rotational motion on Joint 1. Therefore, when Joint 2 was hinge connection, the pitch motions of the activating body affected the rotational motion of Joint 1 through the connecting bridges, thereby affecting the power generation. On the other hand, figure 12 shows the fixed connection at Joint 2 and the hinged connection at Joint 3. In this case, unlike the above case where the hinge connection was made at Joint 2, pendulum movements of the activating body centering on Joint 2 did not be occurred. Instead, pitch motions of the activating body mostly occurred at Joint 3. As a result, the pitch motions transferred to the main body through the connecting bridge were insignificant. However, the heave motions of the activating body were transferred to the main body through the connecting bridge. The simulation demonstrated that the pitch motions of the activating body had almost no effect on the rotational motions of Joint 1 unlike the case with the hinged connection at Joint 2.

In order to analyze a tendency of variation efficiency as the changes of connection methods and location, the amount of power generation, moment, angular velocity was measured. Figure 13 shows graphs of the power outputs calculated by Eq.(5). In other words, displayed power outputs are the average amount of power that is generated through simulation. Figure 13 (a) results from Joint 2 hinge option and (b) results form Joint 3 hinge option. The horizontal axis shows the changes in the distance of “A” (i.e., the location of the connecting bridge). The vertical axis shows the power outputs generated at Joint 1.

As power generation was based on the movement of activating body and the wave conditions were based on the heave RAO of the activating body, it was expected that the largest amount of power was appeared under wave 2. As shown in the figure 13, the expectation was right. When joint 2 had the hinged connection(Figure 13 (a)), under all the wave conditions, the power outputs shows the tendency to descend as the values of “A”, which is distance from the main body, ascends. This trend is also found when joint 3 had the hinged connection(Figure 13(b)). Additionally, the inclinations of Figure 13 (b) are sharper than those of Figure 13 (a). This meant that when joint 3 had the hinged connection, power
outputs are more sensitive to variations of distance of A. Furthermore, under Joint 3 hinge option, the amount of generated power is larger than that of joint 2 hinge option and by using this result, the trend that in terms of power generation that joint 3 hinge option is much better than joint 2 hinge option could be found. To be specific, as shown above, if the joint 2 has a hinged connection, activating body takes pendulum motion and it affects the motion of joint 1. As a result, this effect was found as negative effect because all cases had low amount of power.

![Graph showing non-dimensional capture width](image)

**Fig. 14** Variation of the non-dimensional capture width under the different types of the connections

Figure 14 shows a non-dimensional capture width, which is calculated by multiplying $C_w$ in Eq. (6) and wave number ($k_w$) together. The non-dimensional capture width is one of parameters to estimate the efficiency of a wave energy converter as mentioned above. The lowest wave frequency, wave 1, become lower than the wave 4 because the wave 1 has the smallest wave number. This means that although the wave 4 generates the lower power, it has better efficiency than the wave 1. On the other hand, an important fact is that the same tendency of the power between joint 2 and 3 was found. Under joint 2 hinge condition, the variation of output power efficiency is low but under joint 3 hinge condition, the range of variation is relatively large. Therefore, it is appropriate to say that when joint 3 had a hinged connection, power efficiency is better and more sensitive to distance of A.

Since the power is calculated through Eq.(12), it is needed to consider moment of the joint 1 to scrutinize the tendency of power. Therefore, figure 15 explains moment that was measured at the joint 1. The vertical axis means average RMS(root mean square) moment and other values are the same those of the power graphs in figure 13. In the case of power graphs, all cases descend when the values of A ascend but in view of moment, there is a different trend that if the joint 2 has the hinged connection, moment ascends as the A’s distance ascends. In addition to the change of the A distance, the change of wave frequency also lead to a dissimilar result. If the joint 2 has the hinged connection, the wave frequency has no significant effect on the moment. On the other hand, when the joint 3 has a hinged connection, the variation tendency is similar with the variation tendency of power with the joint 3 hinged case.
Since the moment was calculated by the length of moment arm and force, and the force is calculated by mass and acceleration, the angular acceleration at joint 1 is needed to investigate these trends. Time domain graphs of, which depict absolute-values of angular acceleration of the joint 1 under the wave 3 condition, are shown at Figure 16. The graphs explain merely specific simulation time from 120 sec. to 150 sec. to show the trends clearly because if the graphs depict from zero sec., they are too broad to find out variation of data. Thus, the figures (a) and (b) are enlarged from 0.2 to 0.45 rad/s² and from 135 to 150sec. in the right side of the figure.
There are some variations of amplitude throughout the simulation time because the angular acceleration considers the relative motion between the main body and the activating body. In other words, the small amplitude explains that both the activating body and the main body move in one direction. On the other hand, the big amplitude explains that they move in different directions. According to the (a) graph that shows results when joint 2 hinge option, there are a few variation. Therefore, it is natural that the moment figure is ascended as the figure of A is ascended. This is because similar acceleration means that the figure of moment is determined by the length of moment arm. However, in the case of joint 3 hinge option, there is relatively large variation and this bring that even though the length of moment arm gets longer, the figure of moment gets smaller because of descending force.

![Fig. 17 Variation of the angular velocity under the different types of the connections](image)

As mentioned above, as angular velocity is one of the main factors to calculate the power, the angular velocity is shown in figure 17 to be investigated. The vertical axis means average RMS angular velocity and other values are same those of power graphs. The angular velocity descends as the value of the A ascends, as depicted in the figure 17 (a) and (b). There is no prominent tendency but the similar tendency of power generation. The only difference is that the contrast between the joint 2 hinge and the joint 3 hinge is pretty little. Thus, it is found out that the angular velocity would not be affected by the connecting method as much as moment received. Moreover, a figure 18 shows the relative velocities between the main body and activating body. These velocities are the factors of the angular velocity at the joint 1 as mentioned above the figure 17. Time histories of velocities are compared between the case 1 (Joint 2 hinged) and case 4 (joint 3 hinged) which have that the A is 3m and the wave condition is 2 (Freq.=0.984rad/s). There is no discrepancy of heave velocity between the two bodies, while the surge velocity of the case 4 has comparatively the larger value. This is because the case 4 has the fixed Joint 2. When the joint 2 is fixed, the sure motion of activating body can easily affect the joint 1 through the connection bridge B and C because the surge motion can push both the connection bridge B and C. However, in the case 1, joint 2 hinged and joint 3 fixed, the surge motion leads a swing based on the joint 2, and the swing reduces the effect of sure motion on the joint 1 because the kinetic energy caused by the surge motion is relieved by the swing. In other words, the surge motion of activating body of case 4 has a direct influence on the angular velocity of the joint 1 as compared with that of case 1.
This difference is one of the reasons that the cases with the hinged Joint 3 in the figure 17 have the larger angular velocities. In contrast, in the case of the pitch motion, the case 1 is faster than the case 4. The reasons are that if the joint 2 is hinged and the joint 3 is fixed, the case 1, the pitch motion of the activating body easily affect the joint 1 through the connection bridge B and C because the pitch motion activating body must move with the connection bridge B, thereby being able to push the bridge C. Additionally, if the joint 2 is fixed and the joint 3 is hinged, the case 4, the pitch motion is occurred as a swing base on the joint 3. This means that the effect of pitch motion on the joint 1 is less than that of the case 1. Nevertheless, since the variation of pitch motion between the case 1 and 4 is much smaller than the variation of surge motion, the figure 17 is reasonable.

![Relative velocities between the main body and activating body at the wave 2](image)
5. Conclusion

In the present study, to examine the motion characteristics of an activating body of a WAB-type wave-energy converter according to changes in connecting methods, wave-energy converters were modeled with a commercial software, AQWA, and simulations were conducted. The power outputs were calculated from the moments and angular velocities obtained from the results of the simulations, and the results were analyzed. The conclusions obtained from the present study were as follows:

In the case of wave-power generation systems that use the heaves and pitches of the activating body for power generation, the power generation can be improved by a type of connections, which suppress the effect of the pitch motions of an activating body (i.e. joint 3 have hinged connection.). Changing the location of the connecting bridge also has an effect on the power generation. In this study, as the power is calculated by moment and angular velocity, the longer the distance of connection location is, the less power is generated. However, in specific cases, this is not prominent. For example, if there is a generator that do not significantly consider angular velocity and it cannot suppress the effect of the pitch motions of an activating body, the longer the distance of connection location is, the more power is generated. On the other hand, if an angular velocity is a dominant factor for energy generation, the distance of connection location is pretty important because in this case, a connection type have a few effect on angular velocity.

Future work are noted as follows. As the wave conditions considered in this study are merely regular waves, the wave conditions vary from real sea site conditions. Therefore, irregular wave conditions should be considered. The effectiveness of wave-energy converter should be also investigated by considering the conditions of installation site such as irregular wave and gust wind and so on. Furthermore, the optimization of shape of the activating body and the main body should be conducted.

6. ACKNOWLEDGMENTS

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REFERENCES


A study of motion characteristics led by connection methods and positions of a wave-energy converter in a regular wave


DETERMINATION OF SEDIMENT ACCUMULATION PATTERN IN A DOUBLE BOTTOM BALLAST TANK MODEL

UDC 629.5.018.71:629.5.045.63: 629.5.062.2
Original scientific paper

Summary

Ballast tank sediments may cause several problems in a wide range changing from environmental to economical. Its contribution to biological invasion is an important concern. Thus, the sediment management is included as an integral component of the “The International Convention on the Control and Management of Ships’ Ballast Water and Sediments”. The reduction of the amount of the sediment to be removed is of great importance for management issues. Therefore, the convention underlines that ships should be designed and constructed with a view to minimize the uptake and undesirable entrapment of sediments and facilitate removal of sediments. Design and construction solutions can effectively be developed after determination of the problem areas. In this study, sediment accumulation pattern and problem areas in the ballast tank model of a longitudinally framed double bottom tanker is determined. The problem areas are found to be at the mid-section of the tank closer to the centre girder.

Key words: Ballast tank; Sediment accumulation; Tank design; Invasive species

1. Introduction

Transfer of organisms by ship’s ballast water has received widespread attention for almost decades starting from 1970’s [1]–[4]. The recognition of negative and irreversible impacts of organisms carried in the ballast tanks has triggered many national and international attempts. The International Convention on the Control and Management of Ships’ Ballast Water and Sediments at the auspices of the International Maritime Organization (IMO) brings about the measures in the international level aiming to prevent the spread of harmful aquatic organisms from one region to another. The convention entered into force on 8th September 2017. Besides the IMO convention the United States Coast Guard (USCG) published a final rule in March of 2012 to implement ballast water discharge standards. Even though it is a national legislation, it will apply to all vessels discharging in the U.S. waters that take on ballast water outside the U.S. and Canadian Exclusive Economic Zone. D-2 Standards of IMO convention along with the USCG Ballast Water Discharge Standards Final Rule are the two predominant legislations around the globe. Both legislations bring about the limitations on the allowable number of living organisms within a certain size class of organism in the ballast water [5].
There are more than 70 ballast water treatment systems which have received the Type Approval according to IMO Guidelines for Approval of Ballast Water Management Systems (resolutions MEPC.175 (58) and MEPC.228 (65)) [6]. The decision criteria for selecting ballast water treatment systems are generally based on the ship specifications or/and treatment system characteristics [7]. However, the ballast water characteristics are also effective on the system performance [8]–[11] and these systems are mostly effective on organisms which are living in the water phase. On the other hand, ballast tanks of the ships cannot be completely emptied during de-ballasting [12]. Significant amount of sediment may accumulate in some ships during their operations through the years, which give rise to a further invasion risk. The soil sediment drawn by ballast water is mostly in the form of clay (2 μm or less), silt (2-63 μm) and sand (63 μm-2 mm) [13]. On the other hand, most of the ballast water treatment systems available on the market use filters in the first stage of the treatment process. Typical backwash filters used in these systems are mostly capable to remove particles larger than 40-50 μm, which is above the particle size of the sediment drawn by the ballast water. The studies have shown that none of the ballast water treatment methods are capable of eliminating all types of organisms and each method used within the ballast water treatment systems have its own limitations [10], [14]. As a consequence, there may also be living organisms left in the ballast tanks. The size limitation for allowable living organisms is another concern as many marine organisms have the capability to form resting stages and cysts which are smaller than their vegetative cells and a lot are <10 μm in minimum dimensions [15]. There are a number of studies which draw attention to the invasion risk of organisms within the wide range of taxa and different life stages, which are hosted by the sediment of the ballast tanks [16]–[23]. Once the organisms survive a treatment method, the capability of long term germination of organisms found in ballast sediment [19] and the possibility of in situ hatching of dormant stages in the ballast tanks are important concerns in terms of invasion risk [21].

In addition to invasion risk, sediments that accumulate in the ballast tanks pose several problems in a wide array [24]. One of the concerns is about human health. If a ship has been installed with a ballast water treatment system which makes use of active substances, the sediment may play role as a repository for disinfection byproducts. This in turn increases the human exposure risk during several occasions such as the ballast water sampling, tank maintaining (sediment cleaning) and tank inspection [25]. The sediments also may be an important sink for heavy metals and compounds that are originated from the deterioration of tank plates used for the structural members and tank coating residues [13], [26], [27] which brings about potential ecological risks. On the other hand, the accumulation of the sediment at the bottom of the tanks, along with the other impurities and life forms hosted by the sediment, increases the corrosion risk through different mechanisms [28]–[31]. Besides all, the amount of the sediment accumulated within the tanks may reach up to 100s of tons depending on the vessel type and capacity [19], [32]. This may in return cause a cumulative loss of income through the years between tank cleaning which generally takes place during dry-dockings.

There are several factors affecting the sediment accumulation in a ballast tank. The flow regime in the ballast tank is one of the important factors. The structure of the ballast tanks varies from ship to ship and depends on where it is located on a single ship. However, the complex geometry due to longitudinal and transversal structural members (such as bottom longitudinals, girders, solid floors) used to maintain the ship strength in most ballast tanks is a common issue. This complexity also causes a complex flow regime during ballasting and de-ballasting operations. The tanks contain many compartments separated by longitudinal and transversal structural members and openings on these members. There are many potential dead spots in the tanks. During ballasting and de-ballasting operations, these spots with very low local flow velocities are susceptible to sediment accumulation [33] and besides, the positions of inlet and outlet directly affect the flushing efficiency [34], [35]. Sturtevant et. al. also observed that the
sedimentation pattern differs in the ballast tanks depending on both the proximity of the location to the bell-mouth (ballast intake/pump-out port) and openings on the tank members [36].

The International Convention on the Control and Management of Ships’ Ballast Water and Sediments of IMO brings obligations for ships. Along with other terms included in the Convention, Regulation B.5 states that ships should be designed and constructed with a view to minimize the uptake and undesirable entrapment of sediments. The Marine Environment Protection Committee (MEPC) of IMO also developed “Guidelines on design and construction to facilitate sediment control on ships (G12)” (resolution MEPC.209 (63)) to assist designing ships to minimize the retention of sediment [37]. The main purpose of the present study was to experimentally determine the sediment accumulation pattern in a longitudinally framed double bottom tank model. By identifying the accumulation pattern, it will be possible to identify problem areas and to develop strategies for target areas to avoid sediment accumulation.

2. Materials and Methods

2.1 Experimental System

The experimental system consisted of three components (Fig. 1). A ballast tank model was the main component of the system. A roll simulator component provided regular motion for the tank model. The third component was the sediment and water mixing tank. This tank was used to prepare a mixture (hereinafter will be referred to as ballast mixture) that would be used to fill the ballast tank instead of ballast water. The ballast mixture was filled directly into the ballast tank model with a pipe system. The tank was also emptied by the same circuit.

The ballast tank model (Fig. 2) was constructed as a 1/20 scale model of a real longitudinally framed tanker double bottom ballast tank structure. The model contained all the main components available in the real tanker double bottom ballast tank. The holes and apertures that would allow water flow through the compartments of the tank model (i.e. manholes, lightening holes on the longitudinal and transversal structural members, small apertures opened in the lower corners of the used profiles, cutouts at the intersection of the bottom longitudinal and transverse floors, scallops) were also present with the scaled dimensions. The model was separated longitudinally by a continuous centre girder. Each side of the model had an inlet / outlet at the rear wall close to the continuous centre girder. Left side of the model considered as the port side of the real ship and the right side of the model considered as the starboard side of the ship. Each side had 8 stations and 24 compartments sectioned by transversal floors and longitudinal girders (Fig. 3). The compartment rows were numbered from 1 to 8 starting from bow side to the aft side of the model; compartment columns were labelled as A, B, C from the port side towards the centre line and labelled as D, E, F from the center line to the starboard side of the model. The model was placed on the carrier stand of the roll simulator. Although ships have six degrees of freedom, only the roll motion was taken into account for simplicity.
2.1.1 Preparation of ballast mixture

Sediment directly from the sea bottom would contain different fractions depending on the time and location where the sediment was drawn. This would adversely affect the comparability of the results of replicate experiments. For this reason, it was decided to use industrially manufactured clay instead of sea sediments and commercially available bentonite clay was used to ensure homogeneity.

The ballast sediment mixture was prepared as a suspension of bentonite clay by mixing tap water with bentonite in the mixing tank. The mixing tank was installed with two mutually perpendicular blade systems. The first blade was situated at the bottom of the mixing tank while the other is situated 50 cm above the bottom. The blades had a two-way rotation with intervals to prevent sedimentation at the bottom and ensure well mixing in the water column. The ballast mixture contained 3 g/L sediment in suspension. The sediment concentration was chosen to be 10 times the suspended particulate matter of a standard harbour condition scenario [38], which was compatible with the sediment concentrations in harbours with high sediment concentrations [39]. Since the sediments contained in the ballast water were saturated with water in the real life conditions, the mixture was stirred in the mixing tank for 24 hours prior to it was taken to the tank model for the experiments.

The reason behind the choice of clay is mainly the geometric dimension of the particles. Clay minerals are clay-sized (<2µm) fraction of soils [40]. Most of the sediments that accumulate in ballast tanks are in clay and silt size (<63 µm) [13], [25]. As a result, the particle size to be selected had to be below 63 µm when scaled up to real size. On the other hand, in the ballast tank model, the bottom apertures (located where structural members meet the bottom floor) that provide water passage between the compartments were quite narrow spaces. When the real ship was scaled down to the model, the dimensions of these apertures decreased from centimetres to millimetres (i.e., the diameter of the semicircle shaped scallops decreases from
120 mm to 6 mm, and the diameter of the quarter circle shaped apertures decreases from 36 mm to 1.8 mm.). The particle size to be selected should not cause clogging more than real life condition. And lastly, the average capacity of pre-treatment technologies (filter, hydrocyclone, etc.) of the ballast water treatment systems present in the market were taken into account. It was determined that the significant number of the systems is capable of disposing particles which are sized 40 μm and above. The use of clay with a particle size <2 μm would correspond to a particle size <40 μm when scaled to real condition. Within the framework of the above criteria, clay size found to be appropriate in terms of geometric dimensions. On the other hand, different types of clay were tested preliminarily and bentonite found to be appropriate in terms of sedimentation duration in the water column within the purpose of the experiment.

2.2 Experimental plan

This work was carried out to determine the sediment accumulation pattern in a double bottom ballast tank of a conventional longitudinally framed tanker. The prepared ballast mixture was filled into the model of the ballast tank. Three ballast condition voyage of a tanker (each lasted ~4.5 days) was simulated with the roll simulator. The study was conducted assuming that each ballast voyage took place in a 3-4 sea state (Beaufort number 4-5) condition (roll angle $\phi = 5^\circ$, rolling period $T = 15.65$ s), which is the most common sea state condition [41] that the commercial ships operate at. The working duration of the system and the number of times of filling and discharging were determined depending respectively on the duration of the simulated ballast voyage and the number of ballast voyages. The experimental parameters are given in Table 1.

| Table 1 Experimental parameters |
|----------------------------------|-----------------|-----------------|
| **Parameter**                    | **Real Tank**   | **Model**       |
| Scale                            | -               | 1/20            |
| Beam [m]                         | 20              | 1               |
| Length of the ballast tank [m]   | 20              | 1               |
| Height of the ballast tank [m]   | 1.6             | 0.08            |
| Sea state                        | 3-4             | 3-4             |
| Roll angle ['']                  | 5               | 5               |
| Roll period [s]                  | 15.65           | 3.5             |
| Voyage duration [h]              | 108             | 24              |
| Voyage repetition                | 3               | 3               |

The ballast tank model was filled up to 90% of its capacity with the ballast mixture and then the roll simulator was started. After 24 hours of simulation, the ballast mixture was discharged through the outlet at the rear side of the model. As in the real discharge condition, 5° of trim to the aft side of the model was provided at the end of the process to ensure the maximum discharge volume. After the discharge process was completed, the new ballast mixture was taken on the remaining sediment and the system was restarted for 24 hours. Then the same process was repeated once again. The experimental trial was completed after simulating a total of three ballast cycles. Three trials were conducted under the above mentioned conditions.

At the end of the experiment, the sediment collected from each compartment was taken into the beakers which had been previously weighed. The sediment in the beakers was dried at a temperature of 103-105°C until a constant weight was obtained. The total solids accumulated in each compartment were then determined by analytic scale by measuring the weight gain of the evaporation residue in the sediment collected from the respective compartments.
2.3 Uncertainty Analysis

The uncertainty analysis was carried out by following the International Towing Tank Conference (ITTC) Quality Manual 7.5-02-02-02. Major experimental error resources were considered to be the model geometry, sediment collection and the weight measurement device.

Model geometry and measurement equipment:

The model should be manufactured either to be geometrically similar to the drawings or mathematical model describing the hull form. Even though great effort is given to the task of building a model, no model manufacturing process is perfect and therefore each model has an error in form. The influence of an error in the hull form affects the total volume of the model. Therefore, the bias errors due to model manufacture error both in the model dimensions and the volume were taken into account. Calculation of the bias error due to model manufacture precision allowed by the ITTC is given in Table 2. Each dimension is varied \( \pm 1 \) mm to reflect the maximum manufacture error. The precision limit of the weight measurement equipment was given as 0.0001 g by the manufacturer.

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Bias error from the model manufacture process.</th>
</tr>
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<tbody>
<tr>
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<td>Target Model Dimensions</td>
</tr>
<tr>
<td></td>
<td>Length [mm] 1000</td>
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<tr>
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<td>Beam [mm] 1000</td>
</tr>
<tr>
<td></td>
<td>Depth [mm] 70</td>
</tr>
<tr>
<td></td>
<td>Volume [L] 70,000</td>
</tr>
</tbody>
</table>

Data Evaluation:

As the port side and starboard side of the tank are symmetrical with respect to the continuous centre girder and have their separate ballast intake/pump-out ports, the results obtained from the tanks on both sides are evaluated together. First, uncertainty analysis was carried out for 6 data sets (2 data sets \( \times 3 \)Trials).

According to the uncertainty assessment method recommended in the ITTC Quality manual 7.5-02-02-02 the precision limit for a single record is calculated as,

\[ P(s) = 2 \times SDev \]  

(1)

where \( s \) stands for a single run and \( SDev \) is the standard deviation, and total uncertainty for the multiple records can be calculated as,

\[ P(M) = \frac{P(s)}{\sqrt{M}} \]  

(2)

where \( M \) is the number of records.

3. Results

Considering the weight of the accumulated sediment in the tank bottom will vary depending on the conditions, the test results were evaluated proportionally to be able to have a general perspective on the issue. The proportion of the solid material obtained from each compartment to the total solid material collected from all the compartments of the tank considered (i.e. port side and starboard side of the tank model) were taken into account. For each trial, the sediment accumulation rates (SARs) of each compartment in the port side and starboard side tanks are presented in the contour graphs given in Fig. 4, Fig. 5 and Fig. 6, respectively. The red line separating the model longitudinally represents the continuous centre
The left hand side of the figure gives the sediment accumulation rates within the port side of the model, while the right hand side of the figure gives the sediment accumulation rates within the starboard side of the model. The 100% SAR indicates the total sediment accumulation amount in each side (port and starboard sides) of the tank model.

When the graphs are examined, most of the sediment appear to accumulate intensively in the C3-C4-C5-C6 compartments on the port side and in the D3-D4-D5-D6 compartments on the starboard side. These compartments are located next to the continuous centre girder. The sediment accumulation increases towards the centre of the model, and decreases in the longitudinal and horizontal directions with increasing distance from the centre of the model.

4. Discussion

The observed sediment behavior is a result of the boundary layer formation along the compartment side walls and the bottom. During the rolling motion of a tank that is filled with water, drag will form on the tank walls due to the boundary layer formation. The water right at the sides of the compartments and all along the bottom, will be moving slower than the rest of the liquid in the tank due to the boundary layer effect. All that water pushing up against the
sides of the compartments establishes a pressure gradient, with higher pressure towards the outside of the tank. The slow-moving water is displaced and when it hits the sides of the tank, it drops downwards, and when it gets to the bottom of the tank, and meets the relatively slow-moving water there, it’s forced inwards. Once it gets to the middle section it’s forced upwards, and the cycle continues over and over again.

The time evolution of the liquid motion, including the free surface deformation at regular intervals during one period of the tank motion [42] is given in Fig. 7. In their study, [42] Jung et. al. showed that the fluid flows mainly in the same direction as the tank motion which supports the flow pattern explained above.

![Flow streamlines in a tank at =1/4T, t=2/4T, t=3/4T and t=4/4T [42]](image)

Sediments in the water are too heavy to be lifted up by the upward flow in the tank, but they get caught up in that roll and are directed towards the centre girder of the tank where they accumulate.

The results from the three trials mentioned in the results section were used to generate a general perception of the sediment distribution pattern in the ballast tank model. In this context, the mean value of the three trials was calculated and columns that are symmetrical with respect to the centre girder (starboard side and port side) were evaluated as different data sets. The assumption of different data set is based on the symmetric structure of the model. In the model, the starboard side and port side sections work as separate ballast tanks. They have their own inlet/outlet and their motion is symmetrical with respect to the centre girder, which is watertight.

In Fig. 8, the sediment accumulation in each row from 1 to 8 is given. The sediment accumulation is intensified in the C and D columns which are symmetrically located on each side of the centre girder where the amplitude of the roll motion is the smallest. This causes a slow down on the sediment motion in the compartments closer to the centre of symmetry. On the other hand, due to the boundary layer formation on the exterior watertight walls at the bow and aft sides the sediment is headed towards the centre which corresponds to the rows 4 and 5.
Considering the above mentioned approach, an uncertainty analysis was made according to the ITTCS quality manual 7.5-02-02 to obtain more refined results from the experimental study. The mean value and its rate of uncertainty variation depending on the number of records are given in Table 3 for the nearest columns (C and D) to the center girder where the maximum sediment accumulation was observed. Rate of uncertainty values were calculated by using Equation 2 in Section 2. The mean value and the standard deviation of all tests were calculated and change of uncertainty for each data set is presented in this table.

![SAR (%)](image)

**Fig. 8** The sediment accumulation rates (SAR) in each compartment row in the ballast tank model

<table>
<thead>
<tr>
<th>Row</th>
<th>Mean [g]</th>
<th>Standard Deviation [g]</th>
<th>DS-1</th>
<th>DS-2</th>
<th>DS-3</th>
<th>DS-4</th>
<th>DS-5</th>
<th>DS-6</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.8749</td>
<td>1.9023</td>
<td>0.9819</td>
<td>0.6943</td>
<td>0.5669</td>
<td>0.4909</td>
<td>0.4391</td>
<td>0.4008</td>
</tr>
<tr>
<td>2</td>
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<td>1.9502</td>
<td>0.8829</td>
<td>0.6243</td>
<td>0.5098</td>
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<td>0.3605</td>
</tr>
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<td>5.5349</td>
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<td>0.4533</td>
<td>0.3701</td>
<td>0.3205</td>
<td>0.2867</td>
<td>0.2617</td>
</tr>
<tr>
<td>4</td>
<td>40.0780</td>
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<td>5</td>
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</tr>
<tr>
<td>7</td>
<td>5.7357</td>
<td>2.0378</td>
<td>0.7106</td>
<td>0.5024</td>
<td>0.4102</td>
<td>0.3553</td>
<td>0.3178</td>
<td>0.2901</td>
</tr>
<tr>
<td>8</td>
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<td>1.9424</td>
<td>0.7527</td>
<td>0.5322</td>
<td>0.4345</td>
<td>0.3763</td>
<td>0.3366</td>
<td>0.3073</td>
</tr>
</tbody>
</table>

In Fig. 9, the variation of the uncertainty can be seen by means of the number of records. This figure shows that total uncertainty can be reduced to less than 10% of the mean value provided that 6 and more records of the sediment collection are available.

The results obtained from the uncertainty analysis shows that the general perception of the sediment distribution pattern given in Fig. 8 is in an acceptable error range. The sediment amount will naturally change due to several factors (i.e. sediment load of the port, operational conditions, duration after the last tank cleaning). However, the accumulation patterns tend to be similar.
Fig. 9 Sediment measurement error variation vs. the number of tests

Fig. 10 shows the error distribution along the rows of the subject column. The measurement error rate gets smaller with the increase of the sediment collection amount.

Fig. 10 Sediment measurement error rate vs. the place taken.

In Fig. 11, error bars were placed on the mean value of 6 sediment collection records. It is found that the total uncertainty can be as low as 10% at places with the highest sediment accumulation.

Fig. 11 Mean sediment measurement with error bars according to the uncertainty analysis.
5. Conclusion

Sediment management is an integral component of “The International Convention on the Control and Management of Ships’ Ballast Water and Sediments”. Due to the nature of the sediments, management consists of several stages, processes and procedures [43]. The best strategy for ballast management is avoiding sediment uptake with the ballast water. However, once the sediment is uptaken with the ballast water, the management starts with the collection of the sediment from the tank’s bottom, which is a tedious and demanding task. Therefore, the reduction of the amount of sediment to be collected and managed is of great importance. The regulation B.5 of the International Convention on the Control and Management of Ships’ Ballast Water and Sediments underlines that ships should be designed and constructed with a view to minimize the uptake and undesirable entrapment of sediments and facilitate removal of sediments. From this perspective, developing the design strategies that go in line with the “2012 Guidelines on Design and Construction to Facilitate Sediment Control on Ships (G12)”[37] developed by Marine Environment Protection Committee of IMO is crucial. However, effective design and construction solutions can be developed after determination of the problem areas.

In this study, sediment accumulation pattern in the ballast tank model of a longitudinally framed double bottom tanker is determined in order to find out the target area for developing effective design strategies. The highest sediment accumulation rates are determined at the compartments, which are neighboring the central continuous girder around the mid-part of the tank model. These results are also consistent with the preliminary results of the authors [44].

In their technical paper, Prange and Pereira [12], propose some simple modifications in ballast tanks that include modification in the de-ballasting system, some minor alterations in the tank design and injection of water against the bottom of tanks. Yuan et. al. [45] also proposes a flushing system to re-suspend the accumulated sediments at the bottom and circulate it through a hydrocyclone for separating. These and other strategies would be more effective if the accumulation pattern is taken into account. A combination of several strategies developed depending on the accumulation pattern can be both effective and more feasible.

The findings of this study suggest that re-suspension would be beneficial especially at the compartments neighboring the central continuous girder around mid-part of the tank model, while minor alterations alone can be sufficient at the outer surrounding compartments. These precautions will also decrease the sediment accumulation amount through the duration between two tank cleaning processes.

Although the ships have six degrees of freedom, the rotational motions (i.e. roll, pitch, yaw) are dominant for triggering the sediment motion in the tanks. In this study, only the roll motion was taken into account for simplicity. Future work that will include the pitch and yaw motions in addition to the roll motion will lead to more accurate accumulation patterns. This will benefit the strategies for avoiding sediment accumulation in the tanks.

ACKNOWLEDGEMENT

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REFERENCES
Determination of Sediment Accumulation

Pattern in a Double Bottom Ballast Tank Model


Determination of Sediment Accumulation in a Double Bottom Ballast Tank Model

C. Bilgin Güney, D. B. Danışman, Ş. N. Erteürk Bozkurtoğlu, F. Yonsel

Pattern in a Double Bottom Ballast Tank Model


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HIERARCHICAL RANKING AS BASIS FOR SHIP OUTFITTING PROCESS IMPROVEMENT

UDC 629.5.081
Original scientific paper

Summary

Within the shipbuilding process, the outfitting activities are aimed to be deployed within stages before the ship's erection and ship launching as much as possible, where the cost of the working hour is lower and work quality higher. High level of ship outfitting before launching is one of the most important goals of today’s modern shipyards. In this work, within the ship equipment process, the most important criteria will be identified, evaluated and ranked according to their impact on the level of ship outfitting before launching. Expert approaches and hierarchical ranking is going to be used along with the creation of a computer application to support the solution which can be applied for different shipyards. The result of the evaluation is the sequence of criteria relevant to their impact on the level of outfitting before launching. Based on such results, the authors are proposing improvement of the ship outfitting process, which is expected to improve the ship's equipment level before launching and thus reduce the cost of the shipbuilding process. In the end, the authors will also indicate the guidelines for continuing the research for the purpose of further improving the ship outfitting process.

Keywords: shipbuilding; outfitting; hierarchic ranking; shipbuilding costs

Nomenclature

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>AHP</td>
<td>Analytic Hierarchy process</td>
</tr>
<tr>
<td>$e_{jk}$</td>
<td>grade of k expert for the j criterion</td>
</tr>
<tr>
<td>$E_k$</td>
<td>k expert</td>
</tr>
<tr>
<td>$n$</td>
<td>number of evaluated criterions</td>
</tr>
<tr>
<td>$m$</td>
<td>number of experts</td>
</tr>
<tr>
<td>$p_j$</td>
<td>overall weight factor of j criterion</td>
</tr>
<tr>
<td>$p_{jk}$</td>
<td>weight factor of j criterion based on k expert grade</td>
</tr>
<tr>
<td>$V_j$</td>
<td>the evaluated criterion</td>
</tr>
<tr>
<td>$V_{1n}$</td>
<td>n criterion at level 1</td>
</tr>
<tr>
<td>$V_{2ni}$</td>
<td>i sub-criterion of n criterion at level 2</td>
</tr>
<tr>
<td>$V_{3nij}$</td>
<td>j sub-criterion of i sub-criterion of n criterion at level 3</td>
</tr>
</tbody>
</table>
1. Introduction

The ship outfitting process, within the ship design and building stage, involves significant financial, human and organizational resources, and as such significantly influences the time and cost of shipbuilding and is one of the key indicators of technological and technical development of a shipyard. In actual shipbuilding industry, however, the outfitting is concentrated more towards the earlier stages of shipbuilding production process, opposite to the ship outfitting on berth and in outfitting basin, [1]. Namely, the man-hours on the ship or after launching in the basin is several times more expensive than the same hour in earlier stages of production process within shipbuilding workshops. Mentioned is particularly apparent in special projects of high added value ships. Therefore, regarding the aim of modern shipbuilding industry to shift the focus of ship fitting jobs towards earlier, more efficient phases of production process authors main goal is to identify, evaluate and rank the most important criteria. Such criteria are ranked related to their influence on the level of ship's outfitting before launching, using expert approach methods and hierarchical rankings. Also a computer application to support a solution that can be applied to any other shipyard is developed. In the previous research and application of hierarchical modelling, [2,3,4] the authors have identified the applicability of the hierarchical approach to the ranking problem of the influential criteria on the ship outfitting level. Based on the results of the hierarchical ranking methodology applied, the authors propose processes whose improvement will most probably have an impact on raising the ship outfitting level and ultimately reducing shipbuilding costs. In the paper, within first chapters the basis of the manufacturing process of ship outfitting in modern shipyards is described. Furthermore, in the third chapter the general mathematical model of the hierarchical ranking method is presented. Chapter Four identifies significant influential criteria and their sub criterion by an expert approach and defines a real hierarchical model using three hierarchical levels. Such model is used as basis for the implementation of hierarchical ranking within established computer application. In addition, a hierarchical ranking method is applied on the case study for the selected shipyards of same technical level. Results, discussion, and suggestions for improvement are presented. Analysing obtained results and previous research and experience, [5, 6] the authors elaborate the planned directions for further development and research trough establishing a computer simulation model for optimization of ship outfitting process before ship launching.

2. Background

The ship outfitting process is usually divided into two mutually separate technological phases: pre-outfitting and on board outfitting. For the pre-outfitting process, it is characteristic that the timing takes place almost simultaneously with the construction of the hull and is further divided into two mutually independent technological phases: the on-block outfitting and modular outfitting, [7]. The modular outfitting concept of the ship in the assemblies, modules and block of equipment is the compilation of the ship's equipment in the assembly workshops before assembly to the site of construction, [8]. The on-board outfitting is also divided into two technologically separate phases: on berth outfitting and final outfitting in basin, [9, 10].

The primary purpose of work brake down structure is to reduce outfitting on the berth and after launching resulting in increased productivity and reduced process time and costs. Specifically, it often stated a ratio of 1: 3: 7: 11 for the allocation of working hours according to the stages of construction, [1]. This ratio speaks the following; if the outfitting is being done at an early stage of the block assembly, the cost of outfitting has factor one; for the same work performed in the closed block, the cost of the equipment is three times higher; when the outfitting is done on berth, for the same work the cost is seven times higher; while the cost of outfitting at the
final stage after launching is eleven times higher. Therefore, modern shipbuilding is constantly working to improve the shipbuilding strategy, [11]. However, the authors argue that shipyards in general, in effort to shift the outfitting work load towards the earlier stages of production process, approach the issues by comparing them with other similar shipyards and partially within production processes. In doing so, the specific characteristics of particular shipyard are not adequately involved. Furthermore, the clear identification and systematization of relevant criteria and how their particular and mutual influence is impacting the entire production process, is not adequately covered. Therefore, the authors emphasize that shipyard management, for a more efficient implementation of the outfitting concept in the earlier phases of the production process, needs a better insight on the influential factors or areas to be considered towards better results and lower risk. For this purpose, the authors in this paper define the criteria and their detailed sub-criteria, and then rank them according to their impact on the ship's outfitting process before launching. In addition, the detailed characteristics of certain criteria were further analysed and defined on the basis of the collected and systematized expert indicators that in the observed shipyards frequently led to disturbances in the design plan and the dynamics of outfitting. Furthermore, authors by using the expert approach define these significant criteria and their sub-criterions, and by applying the method of hierarchical ranking evaluate the influence on the outfitting process. The proposed methodology for outfitting process improvement based on expert approach and hierarchical ranking will be described below and further will be implemented on a real sample of five selected shipyards of same technological level, [12,13]. The authors are not familiar with the similar research that would identify and rank the selected shipbuilding process according to their impact on the outfitting process before ship launching by taking into account the interaction with the overall process.

3. Hierarchical ranking method

The authors suggest to evaluate the influential criteria on the outfitting level of before lunch using a hierarchical ranking method that identifies and classifies multi-level criteria 1, 2, ... r, as shown schematically in Figure 1, [2]. Criteria are defined based on collected and systematized expert indicators which frequently led to disturbances in the design plan and the dynamics of outfitting. Furthermore, their sub-criteria stem from the technological process of outfitting as factors directly affecting them at the second hierarchical level, and so on to the r-level.

Fig. 1 Hierarchical ranking structure in r-levels

Where is:

\( V_{kn} \) – n criterion at level 1
For the evaluation and ranking of criteria, the ranking method is applied on the basis of third-level expert assessments awarded to selected shipyards of same technological level. The ratings are based on the evaluation of the interdependence of the criteria at the third hierarchical level, depending on the impact on the outfitting process. The ranking method based on expert ratings can be mathematically described as follows, which is applicable to each hierarchical level:

\[ p_{jk} = \frac{e_{jk}}{\sum_{j=1}^{n} e_{jk}} \]  
\[ p_j = \frac{1}{m} \sum_{k=1}^{m} p_{jk} \]

where is:

\( m \)- number of experts  
\( n \)- number of evaluated criterions  
\( e_{jk} \)- grade of k expert for the j criterion  
\( p_{jk} \)- weight factor of j criterion based on k expert grade  
\( p_j \)- overall weight factor of j criterion

Based on the hierarchical model and ranking methods, authors have created a computer application that is reduced to a tabular approach to solving this problem on the basis of \( n \) criteria and \( m \) experts, as presented in table 1:

<table>
<thead>
<tr>
<th>Table 1 Criterion grades</th>
</tr>
</thead>
<tbody>
<tr>
<td>( E_k ) (k:1-m)</td>
</tr>
<tr>
<td>( V_j ) (j:1..n)</td>
</tr>
<tr>
<td>( E1 )</td>
</tr>
<tr>
<td>.</td>
</tr>
<tr>
<td>.</td>
</tr>
<tr>
<td>.</td>
</tr>
<tr>
<td>( E_m )</td>
</tr>
</tbody>
</table>

\[ \sum_{j=1}^{3} e_{jk} = 1 \]  
\[ \sum_{j=1}^{3} e_{jk} = 2 \]  
\[ \sum_{j=1}^{3} e_{jk} = 3 \]  
\[ \sum_{j=1}^{3} e_{jk} = 4 \]  
\[ \sum_{j=1}^{3} e_{jk} = 5 \]

Where is:  
\( V_j \)-graded criterion  
\( E_k \)-expert

The weighted values of each criterion (\( V_j \)), as compared to the expert grade (\( E_k \)), are further calculated according to the expression (1), and the table solver is shown in Table 2.

<table>
<thead>
<tr>
<th>Table 2 The weight factors of individual criteria in relation to expert grades</th>
</tr>
</thead>
<tbody>
<tr>
<td>( E_k ) (k:1-m)</td>
</tr>
<tr>
<td>( V_j ) (j:1..n)</td>
</tr>
<tr>
<td>( E1 )</td>
</tr>
<tr>
<td>.</td>
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<tr>
<td>.</td>
</tr>
<tr>
<td>.</td>
</tr>
<tr>
<td>( E_m )</td>
</tr>
</tbody>
</table>

\[ \sum_{k=1}^{m} p_{jk} \]
The weight factors individual criterion, \( p_{jk} \), are normalized according to expression (1), and the overall sum for each criterion. Furthermore, the total weight factors of the \( j \)-criteria are normalized and calculated according to (2) using the values in Table 2, as follows:

\[
p_1 = \frac{1}{m} \sum_{k=1}^{m} p_{1k}; \\
p_2 = \frac{1}{m} \sum_{k=1}^{m} p_{2k}; \\
\vdots \\
p_n = \frac{1}{m} \sum_{k=1}^{m} p_{nk};
\]

In such way, the hierarchical ranking of the most influential criteria is defined so to further addressed with the aim of improving the ship's outfitting process most efficiently. In the next chapter, the analysis and definition of defined criteria and their sub-criteria is presented based on selected shipyards, as the foundation for the definition of a hierarchical ranking model structure and further a computer support solution application.

### 4. Criteria and Sub-Criteria Analysis as Input for Hieratical Ranking Model Definition

There are no completely defined activities and equipment that is installed in the prefabrication phase. With the expert approach [5, 8, 9, 14], the AHP [2,12] method, the empirical method, observation method and interview methodology, the criteria for evaluating the pre-outfitting process.

In order to improve the outfitting process, the authors emphasize the necessity of defining the criteria and their sub-criteria, and to rank them regarding the influence on outfitting process. The same will enable the management of the shipyard to have a clearer overview of the critical sites and to act with higher efficiency. For this purpose, an analysis of selected shipyards has been carried out with the aim of defining influential criteria and their sub-criterions as the basis for creating a model of hierarchical ranking supported by the corresponding computer application.

#### 4.1 Impacting criteria and sub-criteria on outfitting process

<table>
<thead>
<tr>
<th>( V1 )</th>
<th>( p_{jk} = \frac{e_{jk}}{\sum_{j=1}^{n} e_{jk}} )</th>
<th>( \sum_{j=1}^{n} p_{jk} )</th>
<th>( \sum_{j=1}^{n} p_{j1} = 1 )</th>
<th>( \sum_{j=1}^{n} p_{j2} )</th>
<th>( \sum_{j=1}^{n} p_{j3} )</th>
<th>( \sum_{j=1}^{n} p_{j4} )</th>
<th>( \sum_{j=1}^{n} p_{j5} )</th>
<th>( \sum_{k=1}^{m} p_{1k} )</th>
<th>( \sum_{k=1}^{m} p_{2k} )</th>
<th>( \sum_{k=1}^{m} p_{3k} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Vn )</td>
<td>( \sum_{j=1}^{n} p_{jk} )</td>
<td>( \sum_{j=1}^{n} p_{j1} = 1 )</td>
<td>( \sum_{j=1}^{n} p_{j2} )</td>
<td>( \sum_{j=1}^{n} p_{j3} )</td>
<td>( \sum_{j=1}^{n} p_{j4} )</td>
<td>( \sum_{j=1}^{n} p_{j5} )</td>
<td>( \sum_{k=1}^{m} p_{1k} )</td>
<td>( \sum_{k=1}^{m} p_{2k} )</td>
<td>( \sum_{k=1}^{m} p_{3k} )</td>
<td></td>
</tr>
</tbody>
</table>
Impacting criteria, and related sub-criteria are defined based on the analysis of collected and systematized expert indicators. Based on such an analysis, the following nine influential criteria on outfitting process before launching were defined:

1. Criteria - Documentation adapted to pre-outfitting process
2. Criteria – Ship hull technological breakdown adapted to pre-outfitting
3. Criteria – Production planning adapted to pre-outfitting process
4. Criteria – Material supply adapted to pre-outfitting process
5. Criteria - Dimensional control
6. Criteria - The capacity and structure of the labour force adapted to pre-outfitting
7. Criteria – Workshops technological constraints for implementation of pre-outfitting
8. Criteria - Shipyard's layout
9. Criteria - Vertical and horizontal transport capacities

Further, conducted expert approach and detailed analysis of the identified criteria and their interaction with other processes in the shipyard are defined along their related sub-criteria as basis for hierarchical ranking method in three levels.

Criteria 1 - Documentation adapted to pre-outfitting; The documentation must first of all be technologically structured according to ship hull brake down structure and outfitting phases. The goal is to have a specific document with relevant information for every outfitting phase. Further, such documentation should be completed and ready for the assembly process at least one month before the start of outfitting, in order to prepare the process, material supply and work activities. In doing so, the authors for this criterion define significant sub-criteria as follows: Sub-criteria of criteria 1: Compliance with the technological process; Compliance with work phases; Information content; Deadline

Criterion 2 - Ship hull technological breakdown adapted to pre-outfitting; Geometry and volume of the ship hull blocks should be designed to accommodate the maximum ability for conducting block outfitting in such a way that boundaries of blocks follow as much as possible real space areas such as tanks, work areas on board, platform bulkheads, etc. If the outfitting is conducted within smaller block, assemblies and sub-assemblies in earlier phases of production, the similar approach applies in doing so, the authors for this criterion define significant sub-criteria as follows: Blocks; Ship hull block breakdown; Work breakdown.

Criterion 3 - Production planning adapted to pre-outfitting process; Planning and work preparation is a key prerequisite for an efficient ship outfitting process. It is important to recognize and monitor the key and interdependent activities, such as: deadlines for documentation, deadlines for ordering and delivery of materials in relation to the dynamics of ship outfitting, and planning and prediction of workload related to planning a sufficient number of labour force, own or subcontractor. In doing so, the authors for this criterion define significant sub-criteria as follows: Planning of documentation preparation; planning of material supply; Personnel planning; planning of works; financial resources planning.

Criterion 4 - Material supply adapted to pre-outfitting process; Material supply should be organized according to technological phases and outfitting process dynamics. It is important to identify equipment that requires a longer delivery time so to be ordered in time. Financial resources should be accordingly provided. For this criterion significant sub-criteria are as follows: Deadlines; Compliance with work phases; Quality.

Criterion 5 - Dimensional control; To ensure the accuracy of outfitting process, high dimensional accuracy is required in order to reduce repair works in latter stages of production, after ship assembly on berth, or on ship. In doing so, the high level of ship outfitting before
launching, generally implies a high level of accuracy [15, 16]. For this criterion significant sub-criteria are as follows: Accuracy; Quality.

Criterion 6 - The capacity and structure of the labour force adapted to pre-outfitting; In order to successfully implement the pre-outfitting concepts, it is necessary to provide adequate work force according to the degree of workload, the deadlines and the structure of the occupation in the different phases of the outfitting [17]. This should take into account peak loads, which should be anticipated in time, and also a need for possible co-operation. For this criterion, significant sub-criteria are as follows: Capacity; Structure.

Criterion 7 - Workshops technological constraints for implementation of pre-outfitting; The technological capabilities of the workshops, their size, the level of equipment with vertical and horizontal transport, and the energy supply directly affect the size of the ship blocks and the level of outfitting that can be applied. For this criterion significant sub-criteria are as follows: Size; Equipment.

Criterion 8 - Shipyard's layout; The application of the outfitting strategy in the earlier stages of production process, also requires the need for larger working surfaces for the disposal of ship blocks, assemblies, sub-assemblies, panels according to the stages of production. In this regard, the largest area should be provided for the disposal of large ship blocks waiting for final assembly prior to painting and assembly on the berth. Also, the equipment workshops should be brought closer to the earlier stage of the process [18]. For this criterion significant sub-criteria are as follows: Size; Equipment; Transport equipment.

Criterion 9 - Vertical and horizontal transport capacities; Vertical and horizontal transport capacities directly affect the technological ship breakdown structure and the level of ship outfitting through production stages. For this criterion significant sub-criteria are as follows: Load capacity; Reach; Overall capacity.

4.2 Hierarchical ranking model based on defined criteria and their sub-criteria

Based on the conducted analysis, defined criteria and their significant sub-criteria, a three-level hierarchical model is established (Figure 2). The established hierarchical model authors are using as a basis for implementing the hierarchical ranking method and basis for creating a computer application for supporting solution. In doing so, each of the identified sub-criteria for the corresponding criterion is ranked according to the criteria at the lower hierarchical level, thus evaluating the impact of the corresponding sub-criterion on the criterion at an immediately higher level.

A case study of ranking defined criteria in the three hierarchical levels of criteria and their defined sub-criteria will be presented below. In doing so, the criteria will be evaluated at the third hierarchical level, and according to their impact on the level of ship outfitting in the observed shipyard. The example is based on five selected shipyards with similar technological level.
Fig. 2 Schematic representation of hierarchical ranking model at three levels
5. Criteria hierarchical ranking for selected shipyards – case study

As already emphasized, the authors have established a computer-based application based on the hierarchical ranking method supported by expert approach. Using expert approach, the authors evaluated the impact of each of the above criteria from the third hierarchical level on the previous 2-level criteria using the grades from 1 to 10. In this case, the small impact was evaluated by grades from 1 to 5, the average impact represents the grades from 5 to 8 and a large impact represents the grades from 9 to 10. An example of a filled score sheet using such approach for the first criterion is shown in Table 3. In the same way, for other main criteria within the established application such a table was created as following the hierarchical structure from Figure 2.

Table 3 Criteria evaluation for determining ship outfitting level prior to launching

<table>
<thead>
<tr>
<th>CRITERIA</th>
<th>SUB-CRITERIA</th>
<th>IMPACT GRADES</th>
</tr>
</thead>
<tbody>
<tr>
<td>V1</td>
<td>2nd LEVEL</td>
<td>SHIPYARD 1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SHIPYARD 2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SHIPYARD 3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SHIPYARD 4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SHIPYARD 5</td>
</tr>
<tr>
<td>Deadlines</td>
<td>Technology breakdown</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Planning</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>Supply</td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>Control</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Work force</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>Workshops</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Work areas</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>Transport</td>
<td>2</td>
</tr>
<tr>
<td>Information content</td>
<td>Technology breakdown</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>Planning</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Supply</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Control</td>
<td>2</td>
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<tr>
<td></td>
<td>Work force</td>
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<td></td>
<td>Workshops</td>
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<td></td>
<td>Work areas</td>
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<tr>
<td></td>
<td>Transport</td>
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</tr>
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<td>Documentation adapted to pre-outfitting process</td>
<td>Technology breakdown</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Planning</td>
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</tr>
<tr>
<td></td>
<td>Supply</td>
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<td>Control</td>
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<td></td>
<td>Work force</td>
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<td>Work areas</td>
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<td></td>
<td>Transport</td>
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<tr>
<td>Compliance with the work phases</td>
<td>Technology breakdown</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Planning</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>Supply</td>
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<tr>
<td></td>
<td>Control</td>
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<td></td>
<td>Transport</td>
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</tbody>
</table>
Using the proposed methodology for all nine selected criteria, a comprehensible assessment of the impact of each criterion on the other criteria was calculated and consequently the ranking of their impacts on the pre-outfitting level was calculated, as shown in Table 4.

Table 4  Criteria overall ranking related to impact on outfitting process before launching

<table>
<thead>
<tr>
<th>CRITERIA</th>
<th>IMPACT STRENGTH</th>
<th>RANKING</th>
</tr>
</thead>
<tbody>
<tr>
<td>Production planning adapted to pre-outfitting process</td>
<td>16%</td>
<td>1</td>
</tr>
<tr>
<td>Ship hull technological breakdown adapted to pre-outfitting</td>
<td>14%</td>
<td>2</td>
</tr>
<tr>
<td>Workshops technological constraints for implementation of pre-outfitting</td>
<td>13%</td>
<td>3</td>
</tr>
<tr>
<td>The capacity and structure of the labour force adapted to pre-outfitting</td>
<td>12%</td>
<td>4</td>
</tr>
<tr>
<td>Shipyard's layout</td>
<td>11%</td>
<td>5</td>
</tr>
<tr>
<td>Documentation adapted to pre-outfitting</td>
<td>10%</td>
<td>6</td>
</tr>
<tr>
<td>Material supply adapted to pre-outfitting process</td>
<td>9%</td>
<td>7</td>
</tr>
<tr>
<td>Vertical and horizontal transport capacities</td>
<td>8%</td>
<td>8</td>
</tr>
<tr>
<td>Dimensional control</td>
<td>7%</td>
<td>9</td>
</tr>
</tbody>
</table>

From the obtained results, it is possible to determine the level of impact of each observed criterion on the ship's outfitting process before launching. It is evident that on the greatest part of the criteria can be influenced with a successful planning of the entire technological process of ship outfitting, which does not require major financial resources other than the well-established plan and its strict monitoring and execution.

In the second place of defined criteria there is a ship hull technological breakdown and works that directly depends on the level of workshops equipment, work surfaces, structure and capacity of the workforce, which also effects on documentation preparation, process planning and materials supply. In doing so, the level of ship outfitting before launching, regarding using suitable ship hull technological breakdown and works, can be improved in two major directions:

a) Improvement by using existing shipyard resources, which do not require special investment funds, and is achieved by investing in the improvement of the supply process, designing and drafting of documentation, and planning and organization of ship hull construction and outfitting, more tailored to the process of outfitting.

b) Improvement with intervention in existing shipyard resources by investing certain financial resources. Investments are primarily concerned with investment in improving the performance of workshops, worksites of shipyards, and investments in vertical and horizontal transport.

In the third and fourth place of the criteria there are technological possibilities of the workshops and the associated structure of the workforce, [17] whose improvement also significantly increases the level of ship outfitting. For such, a certain level of investment is required to raise the level of equipment and the structure of the workforce.

In the fifth place of criteria ranking, there is the size and capacity of the work areas and shipyard layout in general. The layout and its size directly affects the strategy and the possibility of ship blocks outfitting within those areas.

In the sixth place of criteria ranking, there is documentation tailored to the required outfitting level. The volume and content of documentation is directly dependent on the technological process of outfitting, the features of the workshops, the shipyard layout, work areas and the transportable means. Improving the design of the documentation according to the above can also be seen from two aspects:
a) Improving documentation development that does not depend on investment in the resources of the shipyard, which is achieved by creating documentation in the required planning deadlines, in accordance with the technological process of outfitting and ship hull technological breakdown.

b) Improving documentation in relation to raising the level of technological capabilities of the shipyard, which requires the inclusion of new improved features in the content of the documentation.

At seventh place is the procurement and delivery of the material in line with the technological process and the phases of ship outfitting process, for which the improvement does not require additional financial resources, but disciplined compliance with the defined shipbuilding strategy in the planned framework.

At the eighth place of criteria, there are vertical and horizontal transport options, which directly affect the technological constraints and level of ship outfitting.

At the ninth place of criteria, there is a dimensional control, whose level of accuracy significantly increases the quality of the equipment's installation and outfitting process itself. Furthermore, such high dimensional accuracy of ship blocks, initially driven by the need of outfitting process, should ultimately lead to the elimination of the use of blocks over dimensioning on berth assembly, as well as elimination of very expensive reworks in later phases of ship production process, [15], [16], [19].

Ultimately, the authors suggest that the results of this research, which is the criteria definition and their ranking according to the strength of their impact on the ship outfitting process will enable the shipyard's management important ability to identify the critical elements in the ship outfitting process related to the level of ship outfitting before launching.

For that purpose, authors used expert approach to define major criteria influencing ship outfitting process. Further, such criteria were ranked, with the proposed hierarchical ranking method in three levels, according to their identified effect on ship outfitting level before launching. In that way, the most influential areas for the most effective action, with the aim of improving the ship outfitting level, are identified. In addition, the established computing application will allow practical application for different scenarios and various shipyards of similar technological levels.

Regarding financial requirements for improvements following obtained ranking in Table 4, the author would like to emphasise that majority of criteria’s are in the domain of organisation, work discipline and adequate management. Furthermore, in the author’s opinion such investments should not require significant financial resources and could be primarily manageable from shipyards potential. Others investments which could require significant financial allocation, such as equipment, facilities or shipyard layout modifications, should be critically analysed in cost benefit manner and, in authors opinion, most probably attended following the organisational ones. Although the financial aspects was not the major scope of this research, for such analysis authors propose to be one of the significant tasks of further development of this work. For that matter, within further research authors will work on developing the proposed methodology further in a way to conduct a detailed analysis of the outfitting process of selected ship type, with the aim of defining all the activities, working hours, resources and in particular the costs and financial effects of conducted improvements. Also, an attention will be attended to the decision making process analysis as to define critical points on the particular action implementation path, for improved control of partial results and evaluation of their influence on current shipbuilding process flow which must not be interrupted. A simulation model will also be developed with major purpose to monitor process performances and further optimize the ship outfitting process for selected but also for other ship types.
6. Conclusion

In this paper, the authors analysed the ship outfitting production process for selected shipyards of the same technological level in order to identify and evaluate the criteria according to their interaction and impact on the observed process. Based on the conducted expert analysis, the authors defined the significant criteria and relevant sub-criteria with their attributes related to ship outfitting process before launching. Further, such defined criteria and their sub-criteria, were ranked using the proposed hierarchical ranking method, and that how the critical points in outfitting process have been identified. With such ranking, it is pointed out on which criteria the action would result with most effect regarding outfitting process shifting to earlier production phase’s strategy, which will ultimately result in lower shipbuilding costs. Also, the authors identified and highlighted the key prerequisites that must be met towards this goal, and they distinguish two major directions to improve the equipping process, the first one that does not involve significant financial resources and depends primarily on the organization, technology and production process planning, and other direction, which implies intervention in shipbuilding infrastructure, surfaces and equipment that requires large financial resources and should previously consist a thorough study of the feasibility and profitability of such an investment. Also, the authors have established a computer application to support the decision-making process solution, and in such way, this methodology can be applied efficiently to any shipyard of such or similar technological level. At the end of the paper, the authors also propose guidelines for future research that will primarily be reflected in a more detailed analysis of the ship's outfitting process related to the particular ship type. The aim will be to develop a relevant simulation model for tracking the performances of observed process, optimising the costs and improving the overall management capabilities regarding ship outfitting production process.

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Hierarchical ranking as basis for ship outfitting process... T. Matulja, M. Hadjina, et al


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EFFECTS OF DIFFERENT WAVE SPECTRA ON DYNAMIC POSITIONING ACCURACY

Summary

To achieve high accuracy of positioning method for ships which have good mobility and are uninfluenced by the water depth, the dynamic positioning system has come into being. More and more ships working in deepwater area tend to install dynamic positioning system. The main objective of this study is to discuss the effects of different wave spectra on the dynamic positioning accuracy. First, considering some parameters of a certain dynamic positioning ship, the Response Amplitude Operator (RAO) is calculated by AQWA. Then, time domain simulations and calculations for the ship motion have been conducted by OrcaFlex based on JONSWAP, ISSC, Ochi-Hubble, Torsethaugen and Gaussian Swell, respectively. Finally, the analysis results are compared to discuss the dynamic positioning accuracy. Some useful conclusions and recommendations are obtained to lay a theoretical foundation of actual project practice.

Key words: dynamic positioning; wave spectra; numerical simulation; time-domain analysis;

1. Introduction

With the global energy shortage, the exploitation of oil in deep seas are increasing. At present, the development trend of offshore oil and gas resources is also developed from shallow water to deep water. The development of marine resources and marine high technology has become an inevitable tendency for the utilization of ocean energy resources. Ships and other floating facilities usually need to be located at a particular position in the sea for various operations. The traditional mooring positioning method is simple and economical, but it has low positioning accuracy and poor maneuverability. Once anchored, the process of repositioning is more complicated [1]. That is to say, the traditional mooring positioning can not meet the requirements of some marine operations.

In order to meet higher requirements of ocean engineering operations and to keep their position and heading in the ocean, ships equipped with a dynamic positioning system has come into being [2]. The dynamic positioning system refers to the closed-loop control system which is used to measure the deviation of the actual course, position and target course and position of the ship; by taking into account the influence of environmental interference such
as waves, the dynamic positioning system can calculate the thrust load required to return to the target course and position to keep the ship at the target heading and position [3].

The dynamic coupling analysis is recognized as the most reliable numerical method presently and an accurate computational analysis of wave-body interaction is important in the design of marine vessels [4][5]. In order to study the effect of different wave spectra of ship dynamic positioning, the three-dimensional model is built and imported into AQWA to obtain Response Amplitude Operator (RAO), added mass and radiation damping response amplitude by frequency domain analysis. Then combined with the hydrodynamic software OrcaFlex, by changing the environmental parameters, the hydrodynamic response of the dynamic positioning ship under different wave spectra is analyzed. Finally, the influence of different wave spectra on the dynamic positioning accuracy of the DP ship has been obtained. The results have some guiding significance for the actual engineering practice.

2. Wave spectra

OrcaFlex provides regular and random waves. The regular wave theories include linear Airy and nonlinear wave theory, such as Stokes 5th order wave theory, cnoidal wave and Dean Stream. In order to describe the internal structure of random waves, standard wave frequency spectra are applied. According to Fourier analysis, the irregular wave could be decomposed into a number of waves with discrete frequencies, or a spectrum of frequencies over a continuous range. The statistical average of a certain irregular wave as analyzed in terms of its frequency content, is called its spectrum [6]. It is based on a large number of data observed by many researchers, analyzing various statistical values and spectral density of waves. Each wave spectrum is derived from the measurement, statistics and analysis in a specific condition and a specific sea area and has their own advantages and limitations. JONSWAP, ISSC, Ochi-Hubble, Torsethaugen and Gaussian Swell are used in this paper, there are brief introduction for them:

The JONSWAP spectrum in OrcaFlex is defined as:

$$S \left( f \right) = \left( \frac{3g^{2}}{16\pi^{4}} \right) f^{-5} \exp\left( -\frac{5}{4} \left( \frac{f}{f_{m}} \right)^{4} \right) \gamma^{b}$$

where $\alpha$ is a spectral energy parameter, $g$ is the gravitational constant, $f_{m}$ is the spectral peak frequency, $\gamma$ is peak enhancement factor which have detailed description in relevant literature [7]. $b=\exp\left( -\frac{5}{4} \sigma_{2}^{2} \left( 1 - 1^{f} \right) \right)$, $\sigma=\sigma_{1}$ for $f \leq f_{m}$, $\sigma=\sigma_{s}$ for $f > f_{m}$, $\sigma_{1}$ and $\sigma_{s}$ are spectral width parameters which are set at the standard values of 0.07 and 0.09 respectively. JONSWAP spectrum is mainly used for describing no-full development wave and considers finite wind pressure.

The ISSC spectrum (modified Pierson-Moskowitz) is defined as:

$$S \left( f \right) = \frac{5}{16H_{s}^{4}} f_{s}^{4} f^{-5} \exp\left( -\frac{5}{4} \left( \frac{f}{f_{m}} \right)^{4} \right)$$

$H_{s}$ is significant wave height. For the ISSC spectrum you specify $H_{s}$ and $T_{z}$ and OrcaFlex calculates the other spectral parameter. More detail information can be seen in the reference [8].

The Ochi-Hubble Spectrum allows two peaked spectra to be set up, which shows sea states that include both a remotely generated swell and a local wind generated sea. The Ochi-Hubble wave spectrum is the sum of two separate component spectra. The component spectrum with the lower frequency peak corresponds to the remotely generated swell and the one with the higher frequency peak corresponds to the local wind generated sea. This is why the Ochi-Hubble spectrum is often called a two-peaked spectrum; however in practice, the resulting total spectrum typically has only one peak (from the remotely generated swell) plus
a shoulder of energy from the local wind generated sea. More details of the Ochi-Hubble Spectrum could be found in the reference [9].

The Torsethaugen spectrum is another two-peaked spectrum, more suited to North Sea application than Ochi-Hubble. More details of the Torsethaugen and Haver could be found in the reference [10].

The Gaussian Swell spectrum is based on the normal (or Gaussian) probability density function and is defined as:

\[
S(f) = \frac{(H_s/4)^2 \sigma^{-1} (2\pi)^{-1/2}}{2\sigma^2} \exp\left(-\left[f - f_m\right]^2 / 2\sigma^2\right)
\]

\[f_m = 1/T_p\], The Gaussian Swell spectrum is typically used to model long period swell seas.

Nonlinearity of the wave has an important effect on ship motion especially in the natural frequency region [11]. During numerical simulation, the calculation of transient wave forces in irregular waves refers to literature [12].

3. Description of the dynamic positioning system

OrcaFlex includes a well-documented programming interface called OrcFxAPI which is a windows dynamic link library(DLL). The OrcFxAPI is used to produce the dynamic positioning system in our study. DLL provides facilities for setting data, calculating static positions and extracting results from those calculations or from pre-run simulation files [13]. It uses an external function in Python to keep the vessel on station by making the global applied force moment simply proportional to the difference between the actual and target position and heading of the vessel. This has the same effect as applying simple linear springs in the global X and Y-directions, and a torsional spring about the vertical. A real DP system would use multiple thrusters a more sophisticated controlling algorithm actually.

The magnitude of the heave, roll and pitch variables is relatively small, thus, the DP systems used in most of the industrial ship control systems do not consider the vertical motion [14]. In this study, only the horizontal motions are considered for the DP system. OrcFxAPI is chosen here to simulate the DP system for the ship. A global applied load is specified on the Applied Loads page of the DP Vessel data form. The load uses the external function variable data item called Thruster to calculate the components of applied force in the global X and Y directions, and the applied moment about the global Z direction (vertically upwards). On the Variable Data form, the Python module and class is specified for this Thruster variable data source. The target position and heading of the vessel, the control parameters of reaction and reaction torque, are specified on the External Functions page of the DP Vessel data form. Thruster Data can be saved in the model workspace dictionary. The governing equations of the DP system are shown as follows [15]:

\[F_X = (T_X - v_P[0]) \cdot k_f\]  \hspace{1cm} (4)

\[F_Y = (T_Y - v_P[1]) \cdot k_f\]  \hspace{1cm} (5)

\[M_Z = (T_H - v_h) \cdot k_m\]  \hspace{1cm} (6)

In the above equations, \(F_X, F_Y\) mean the thruster force in surge, sway direction, respectively; \(M_Z\) is the thruster moment in row direction; \(T_X, T_Y, T_H\) denote the target position of the vessel origin in \(X, Y\) direction, and the heading angle of the vessel; \(v_P, v_h\) represent the vessel instantaneous position in \(X,Y\) direction and vessel heading angle, which can be calculated by OrcaFlex; \(k_f, k_m\) show forces applied by thruster per unit distance away from the target position and moment per degree away from the target heading respectively.
4. **Wave spectra frequency-domain hydrodynamic analysis**

In this work, the AQWA module is used to conduct the frequency-domain hydrodynamic analysis of the ship. First, the RAOs are obtained. Then the calculated data are imported into OrcaFlex. Finally, the dynamic response of dynamic positioning ship based on different wave spectra is made by changing the environment parameter.

4.1 Establishment of ship model in AQWA

Combined with the main dimensions and other parameters of a ship, the ship model is established in AQWA and the grids are made into some elements. The basic parameters of the ship are shown in Table 1.

The global coordinate system is defined as O-XYZ and the ship coordinate system is defined as v-xyz, respectively. The motions of six degrees of freedom of the ship under the action of waves are three translational components (surge, sway, heave) along the x-axis, the y-axis and the z-axis direction of the ship coordinate system v-xyz and three rotational components (roll, pitch and yaw) round the x-axis the y-axis and the z-axis direction of the ship coordinate system v-xyz. The ship model established in AQWA has 17385 nodes and 17183 elements, the maximum element size is 1.5m, and the upper end of the range of the wave frequency is 0.457Hz. The ship is regarded as a whole in AQWA, and with a boundary element numerical method for solving the structure of wave loads and considering the 2nd order wave force. In this paper, the ship modeling in AQWA is as follows.

<table>
<thead>
<tr>
<th>Table 1 Basic parameters of the ship</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Length(m)</strong></td>
</tr>
<tr>
<td>207.21</td>
</tr>
<tr>
<td><strong>Waterplane Area (m²)</strong></td>
</tr>
<tr>
<td>6203.14</td>
</tr>
</tbody>
</table>

**Fig. 1 Ship surface element model**

4.1.1 RAOs of the ship

The ship model is imported into the AQWA and the relevant parameters are set up. The hydrodynamic parameters are shown in Table 2. The wave frequency range is 0.01-0.457Hz, which covers the vast majority of the wave energy.

<table>
<thead>
<tr>
<th>Table 2. Hydrodynamic analysis parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Water Depth /m</strong></td>
</tr>
<tr>
<td><strong>Wave Directions α /°</strong></td>
</tr>
<tr>
<td><strong>Wave Frequencies / Hz</strong></td>
</tr>
<tr>
<td><strong>Water Density /kg • m²</strong></td>
</tr>
</tbody>
</table>
The RAOs of six degrees of freedom under different wave directions (surge, sway, heave, roll, pitch, and yaw) of the ship are obtained:

![Fig. 2 Ship RAOs of different wave directions](image)

It can be obtained from Figure 2 that in the 0°-180°, when the wave frequency is very small, the surge RAOs under 0° wave direction have a maximum value and there is a minimum value under 90° wave direction for the surge RAOs; the results of the sway RAOs is contrary to that of the surge RAOs. When the frequency is in 0.01-0.05Hz, the values of the heave RAOs are basically same under each wave direction. From Figure 2, when the wave angle is 90°, the roll RAOs have the maximum value and there are two peaks around the maximum value; the pitch RAOs have the minimum value under 90° wave direction; the minimum value of yaw RAOs appears under 0° wave direction and the maximum value appears in the wave direction 45°and 135°. At the same time, Most of the RAOs peaks occur in the low frequency phase, which are less than 0.3Hz; the same frequency, the same wave direction, pitch, yaw and roll have the larger RAOs value, that’s way we focus on the three degrees of freedom ship motion response.

5. Time domain analysis of ship motion

Since the pioneering work by Cummins [16], Liapis and Beck [17] for time-domain methods, the time domain method in dealing with some of the seakeeping problems has gradually began to replace the frequency domain analysis. Thus, the nonlinear problem of the ship motions is usually solved by time-domain analysis.

Large structures on the sea surface will oscillate at a specific periodicity. These cycles do not have the influence of the first-order spectral energy, which are not for the action of the first-order wave forces. These periodic motions may cause large resonance motions in the water which is called the low-frequency motion. Although the structure will heave, roll and pitch in waves, we are more concerned about the motions in the horizontal plane: surge, sway and yaw. The equation for low-frequency motion can be expressed as [18]:

\[
\begin{bmatrix} m_s + m_a \end{bmatrix} \ddot{x}(t) = F_{sv}(t) + F_{c}(t) + F_{w}(t) + F_{h}(t) + F_{d}(t) + F_{dp}(t)
\] (7)

In the above formula, \( \ddot{x} \) is the acceleration vector; \( m_s \) is the mass and inertia of the floating body; \( m_a \) is added mass and inertia in low frequency; \( F_{sv} \) is slow drift force; \( F_c \) is...
current load; $F_w$ is wind load; $F_h$ is hydrostatic pressure; $F_d$ is damping force, $F_{DP}$ is thrust offered by propeller.

In the process of calculation, the added mass, inertia and damping are considered as constants.

5.1 Setting of model parameters in OrcaFlex

Ship model is built in OrcaFlex, and the ship-related parameters are imported from AQWA. The ship is equipped with dynamic positioning system (DP ship). In order to overcome the influence of the wind, wave, current and other environmental loads on the departure of ship from the course, the DP ship is driven by automatic control of the main propulsion, lateral thrust device, which make thrust and torque in actual construction process. The dynamic positioning effect of the ship depended on the programming module. When the environment parameters are input, the ship could get the opposite direction of the environment load force (moment) to make the ship position in the target position with small amplitude fluctuation.

For dynamic positioning vessel, only three degrees-of-freedom (sway, surge and sway) motions in the horizontal direction are considered. This is because the remaining three directions (heave, roll, pitch) have a certain restoring force (moment), can reset itself, which do not need to control.

In this paper, OrcaFlex is used to analyze the time domain of the ship equipped with dynamic positioning system. The whole process is divided into two stages: static stage and dynamic stage. The purpose of the static analysis is to determine the static equilibrium state of the whole operating system under the action of environmental load coupling. The system equilibrium achieved by static analysis provides the best initial state for dynamic analysis. The dynamic analysis is based on the static analysis results and it simulates the dynamic response of the ship under nonlinear wave loads.

The environmental parameters are set as follows: the average wave height is 6.2m, the spectrum peak period is 11.1s, the wind speed is 19.3m/s, the surface current velocity is 1.1m/s and the wind direction is 90°. In this case, we use two coordinate systems, one is global coordinate system GXYZ, another is local systems Vxyz. All the coordinate systems are right-handed, as shown in the following figure. Directions for waves, current and wind are specified by giving the direction in which the wave (or current or wind) is progressing, relative to global axes. Both of initial position and target position of this ship are set X=120m, Y=80m, $\phi = 90$ deg, which are also relative to the global coordinate system.

![Image](image_url)

Fig.4 Coordinate Systems and the environment load directions

5.2 Motion response of DP ship based on different wave spectra

In order to analyse the effects of different wave spectra on DP ship, the steady-state time series of 2000s-4000s are chosen here. The statistical result is shown in Figure.5 and Table.3.
Fig. 5 Horizontal motion of DP ship
6. Conclusion

In this paper, the effects of different wave spectra on dynamic positioning accuracy of DP ship is studied. The compared results show that different wave spectra have different effects on the calculations of DP ship’s horizontal motion amplitude. It means that the dynamic positioning accuracy calculated based on different wave spectra would be discrepant. Besides, the effects of wave spectrum on dynamic positioning accuracy in each degree of freedom could be different. The change of DP ship's transverse, longitudinal displacement and yaw angle calculated by ISSC spectrum and Ochi-Hubble spectrum are highly consistent. The displacement amplitude of DP ship calculated by JONSWAP spectrum is smallest and the dynamic positioning accuracy is best in surge direction. Dynamic positioning accuracy calculated by Gaussian Swell is poor and have sufficient safety redundancy in surge direction. In sway and yaw direction, the dynamic positioning accuracy calculated by Gaussian Swell is good. The motion amplitude calculated by Torsethaugen spectrum is large and dynamic positioning accuracy is poor, which results in the calculation have sufficient safety redundancy. Therefore, in actual project practice, the wave spectra should be selected carefully when simulation is conducted for DP ship.

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A SYSTEMATIC INVESTIGATION OF THE EFFECTS OF VARIOUS BULBOUS BOWS ON RESISTANCE OF FISHING BOATS

UDC 629.5.015.2: 629.5.024.32: 629.562
Original scientific paper

Summary

In this study, Delta (Δ), Nabla (∇) and Elliptical (O) shaped bulbs are implemented to ITU Fishing Boats Series hull forms of 148/3, 148/4, 148/8 and 148/9 in order to determine the most appropriate bulbous bow for fishing vessels. Initially, in defined Fn values, Computational Fluid Dynamics (CFD) analyses on four main forms are performed by using Realizable k-ε Model and Volume of Fluid (VOF) method. The total resistance values obtained by the CFD analyses and by the existing test results via Froude and Hughes methods are compared and examined. Thus, the method and reference values of CFD analyses are determined for the hull forms with bulbs which do not have test results. Subsequently, the CFD results of frictional resistance, pressure resistance and total resistance values for actual hulls and hulls with bulbs are compared. Following the examination of results in terms of C_B, L/B and bulb types, it is determined that the elliptical bulb type is the most suitable bulb type for the fishing vessels.

Key words: Fishing Boat; Bulbous Bow; Computational Fluid Dynamics (CFD); Resistance;

1. Introduction

According to the culture of each country, certain forms of fishing boats have been developed according to needs over time. While improvements have been made in the main forms previously, the application of bulb to fishing boats is carried out because of that efficiency of bulb is observed today. The resistance analyses, which are made in model test basins, can be done on computers with the help of CFD softwares thanks to the development of technology.

The part of head of the ships, that is underwater, is inflated like a prominence or convexity, and it is called bulb [1]. The bulb forms are classified according to the form of the crosscut area of the head. As a general definition, there are three basic types of bulb geometry, namely, Delta (Δ), Circular-Elliptic (O) and Nabla (∇) sections [2]. The sections of the bulbs are shown in Figure 1.
Taylor [3] is the first researcher to experimentally investigate the effects of bulbs on ship form. Later, Bragg [4], Inui et al. [5], Ferguson [6], and Muntjewerf [7] experimentally conducted experiments on the Δ (delta) type, cylindrical and conical bulbs, which are known as Taylor bulbs, by systematically changing the bulb parameters. Weinblum [8], Wigley [9], Inui [10] and Yim [11] studied on the theory of linearized wave resistance theoretically. Inui [10] presented a method to determine the size of the bulb by matching amplitude functions of bulb in regular waves and stem. A connection for a speed has established between the entrance angle of ship head and the size of bulb by Yim [12]. A method consisting of three main subjects for designing spherical bulb was presented by Yim [13]. Again, Yim [14] discussed the sheltering effect on spherical bulbs. Baba [15] and, Shearer and Steele [16] pointed out that the bulb has benefits like; to reduce the wave breaks on stem, to improve the flow around keel line and bilge turn, as well as preventing flow separation on ship forms. Kracht [2] developed a statistical method from the experience of propulsion tests. The method gives power reduction for the selected bulb or suitable bulb design for a selected power reduction. The Kracht method is more useful for nabla (∇) sectioned bulbs. Sharma and Sha [17] developed a method of designing a bulb by combining Kracht [2] and Yim [13] methods, which are two famous theories accepted in the bulb design. The method can do optimization of bulb parameters for design speed. The method uses a reanalysis of an approximate linear theory with sheltering effect for resistance estimation, and re-correlation with statistical analysis via a non-linear multivariate regression analysis from existing literature and tank test results available in the public domain.

There are different discretization techniques to solve various problems at CFD. The Finite Volume Method (FVM), which is derived from the Finite Difference Method (FDM) formulation, is one of the most widely used methods in CFD. Because it gives good results in non-structural solution mesh as well as in structural solutions. It is developed by Godunov [18]. With evolving technology, turbulence models have been developed that can solve the flow around complex and large geometries such as ships. The RANS (Reynolds-Averaged Navier-Stokes) turbulence modeling techniques, which are a simpler approach than other turbulence models, are used to solve the flow around the ship. Generally in CFD, k-ε models (Standard k-ε model, Realizable k-ε model, RNG k-ε model) and k-ω models are used from two-equation models at flow applications around the ship. The Realizable k-ε Model, which is the most developed version of the k-ε model, was developed by Shih et al. [19].

The CFD study of bulb optimization started in the 1990s. A comparative study of alternative bulb varieties (elliptical, conical, spoon, improved) with the aid of CFD is presented by Stromgren [20]. Kim and Jang [21] studied the effect of bulb on wave characteristics with CFD. An optimization of the pressure distribution on the surface and the around Series 60 vessels with bulb is studied with CFD by Huang et al. [22]. Lee and Sarath [23] conducted bulb designs in different forms for 12500 TEU containers ships and tried to determine optimum bulb sizes with CFD. A bulb optimization of a 36-meter-long fishing boat...
was conducted to improve it hydrodynamically with CFD by Sarasquete et al. [24], and the results of CFD analyses were compared with the data of the model resistance tests. A numerical procedure, which based on the genetic algorithm and a potential flow solver, for hydrodynamic optimization of a ship hull form with a bulbous bow has been established by Matulja and Dejhall [25]. Chrismianto and Kim [26] used a cubic Bezier curvature and curve-plane intersection methods to design a bulb for the KRISO container ship model, based on 4 design parameters. The resistance values, which were obtained by the CFD analyses, were compared with the model data, and the accuracy of the analyses was confirmed.

The aim of the study is to determine which type of bulb on the fishing boats will be more effective in reducing total resistance. For this purpose, Delta (Δ), Nabla (∇) and Elliptic (O) sectioned bulbs are added to the forms of 148/3, 148/4, 148/8 and 148/9 coded boats from ITU Fishing Boats Series. The designs of the forms with bulbs have been made. Firstly, Froude [27] and Hughes [28] method are used to calculate the total resistance of the full-scale boat from the model test data. Secondly, the results, which are obtained by CFD analyses, are compared with the test results. Then, CFD analyses of boat forms with bulbs are made. After that, the total resistance values, which are obtained from the CFD analyses, are compared with each other. Finally, the inferences, which are obtained in the study, are assessed according to ship codes, C_B, L/B ratio and bulb types. In this way, it has been determined which type of bulb is more beneficial for the fishing boats.

2. Ship, model and bulb geometry (characteristic features)

In the study conducted, the ship forms were selected from the Fishing Boats Series of ITU. Because of requirement to obtain more suitable boat forms for waters of Turkey, these fishing boat forms were produced by Kafali et al.[29]. Characteristic values of these fishing boat forms are given in Table 1. The characteristic values of the models of these fishing boats are given in Table 2. Fishing Boats Series of ITU has been concluded with the study named "Computer Aided Design of Fishing Boats Suitable for Turkish Waters" which are prepared by Aydin [30].

The effects of bulbs on resistance were examined for 148/1 coded model by Soylemez [31]. The bulbs were in the delta profile. They were named A_1, A_2 and A_3. The characteristics of the three bulbs are given in Table 3.

<table>
<thead>
<tr>
<th>Boat No.</th>
<th>L (m)</th>
<th>B (m)</th>
<th>T (m)</th>
<th>C_B</th>
<th>C_M</th>
<th>C_WP</th>
<th>C_F</th>
<th>L/B</th>
<th>B/T</th>
<th>LCB (m) (+ Aft)</th>
<th>S_W (m^2) (without app.)</th>
<th>Δs (kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>148/1</td>
<td>20.00</td>
<td>5.714</td>
<td>2.286</td>
<td>0.378</td>
<td>0.661</td>
<td>0.730</td>
<td>0.572</td>
<td>3.500</td>
<td>2.500</td>
<td>0.83</td>
<td>126.10</td>
<td>992.96</td>
</tr>
<tr>
<td>148/2</td>
<td>20.00</td>
<td>5.714</td>
<td>2.286</td>
<td>0.535</td>
<td>0.892</td>
<td>0.789</td>
<td>0.600</td>
<td>3.500</td>
<td>2.500</td>
<td>0.01</td>
<td>139.80</td>
<td>1405.38</td>
</tr>
<tr>
<td>148/3</td>
<td>20.00</td>
<td>5.714</td>
<td>2.286</td>
<td>0.406</td>
<td>0.668</td>
<td>0.727</td>
<td>0.608</td>
<td>3.500</td>
<td>2.500</td>
<td>0.80</td>
<td>125.00</td>
<td>1066.51</td>
</tr>
<tr>
<td>148/4</td>
<td>20.00</td>
<td>5.714</td>
<td>2.286</td>
<td>0.497</td>
<td>0.888</td>
<td>0.789</td>
<td>0.560</td>
<td>3.500</td>
<td>2.500</td>
<td>0.02</td>
<td>134.10</td>
<td>1305.56</td>
</tr>
<tr>
<td>148/5</td>
<td>20.00</td>
<td>5.714</td>
<td>2.286</td>
<td>0.444</td>
<td>0.720</td>
<td>0.745</td>
<td>0.617</td>
<td>3.500</td>
<td>2.500</td>
<td>0.63</td>
<td>131.00</td>
<td>1166.33</td>
</tr>
<tr>
<td>148/6</td>
<td>22.86</td>
<td>5.714</td>
<td>2.286</td>
<td>0.400</td>
<td>0.668</td>
<td>0.727</td>
<td>0.599</td>
<td>4.001</td>
<td>2.500</td>
<td>0.91</td>
<td>145.50</td>
<td>1201.01</td>
</tr>
<tr>
<td>148/7</td>
<td>22.86</td>
<td>5.714</td>
<td>2.286</td>
<td>0.491</td>
<td>0.888</td>
<td>0.789</td>
<td>0.553</td>
<td>4.001</td>
<td>2.500</td>
<td>0.02</td>
<td>152.50</td>
<td>1474.24</td>
</tr>
<tr>
<td>148/8</td>
<td>28.57</td>
<td>5.714</td>
<td>2.286</td>
<td>0.404</td>
<td>0.668</td>
<td>0.727</td>
<td>0.605</td>
<td>5.000</td>
<td>2.500</td>
<td>1.14</td>
<td>179.40</td>
<td>1516.01</td>
</tr>
<tr>
<td>148/9</td>
<td>28.57</td>
<td>5.714</td>
<td>2.286</td>
<td>0.493</td>
<td>0.888</td>
<td>0.789</td>
<td>0.555</td>
<td>5.000</td>
<td>2.500</td>
<td>0.03</td>
<td>190.80</td>
<td>1849.98</td>
</tr>
</tbody>
</table>
Table 2 Some characteristics values of the models used in the tests [32]

<table>
<thead>
<tr>
<th>Model No.</th>
<th>L</th>
<th>B</th>
<th>T</th>
<th>C_B</th>
<th>C_M</th>
<th>C_WP</th>
<th>C_P</th>
<th>L/B</th>
<th>B/T</th>
<th>LCB (m) (+ Aft)</th>
<th>S_W  (m²)</th>
<th>Δ_m (kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>148/1B</td>
<td>2.000</td>
<td>0.571</td>
<td>0.229</td>
<td>0.378</td>
<td>0.661</td>
<td>0.730</td>
<td>0.572</td>
<td>3.500</td>
<td>2.500</td>
<td>0.083</td>
<td>1.261</td>
<td>0.968</td>
</tr>
<tr>
<td>148/2B</td>
<td>2.000</td>
<td>0.571</td>
<td>0.229</td>
<td>0.535</td>
<td>0.892</td>
<td>0.789</td>
<td>0.600</td>
<td>3.500</td>
<td>2.500</td>
<td>0.001</td>
<td>1.398</td>
<td>1.370</td>
</tr>
<tr>
<td>148/3B</td>
<td>2.000</td>
<td>0.571</td>
<td>0.229</td>
<td>0.406</td>
<td>0.668</td>
<td>0.727</td>
<td>0.608</td>
<td>3.500</td>
<td>2.500</td>
<td>0.080</td>
<td>1.250</td>
<td>1.039</td>
</tr>
<tr>
<td>148/4B</td>
<td>2.000</td>
<td>0.571</td>
<td>0.229</td>
<td>0.497</td>
<td>0.888</td>
<td>0.789</td>
<td>0.560</td>
<td>3.500</td>
<td>2.500</td>
<td>0.002</td>
<td>1.341</td>
<td>1.272</td>
</tr>
<tr>
<td>148/5B</td>
<td>2.000</td>
<td>0.571</td>
<td>0.229</td>
<td>0.444</td>
<td>0.720</td>
<td>0.745</td>
<td>0.617</td>
<td>3.500</td>
<td>2.500</td>
<td>0.063</td>
<td>1.310</td>
<td>1.137</td>
</tr>
<tr>
<td>148/6B</td>
<td>2.286</td>
<td>0.571</td>
<td>0.229</td>
<td>0.400</td>
<td>0.668</td>
<td>0.727</td>
<td>0.599</td>
<td>4.001</td>
<td>2.500</td>
<td>0.091</td>
<td>1.455</td>
<td>0.171</td>
</tr>
<tr>
<td>148/7B</td>
<td>2.286</td>
<td>0.571</td>
<td>0.229</td>
<td>0.491</td>
<td>0.888</td>
<td>0.789</td>
<td>0.553</td>
<td>4.001</td>
<td>2.500</td>
<td>0.002</td>
<td>1.525</td>
<td>1.437</td>
</tr>
<tr>
<td>148/8B</td>
<td>2.857</td>
<td>0.571</td>
<td>0.229</td>
<td>0.404</td>
<td>0.668</td>
<td>0.727</td>
<td>0.605</td>
<td>5.000</td>
<td>2.500</td>
<td>0.114</td>
<td>1.794</td>
<td>1.478</td>
</tr>
<tr>
<td>148/9B</td>
<td>2.857</td>
<td>0.571</td>
<td>0.229</td>
<td>0.493</td>
<td>0.888</td>
<td>0.789</td>
<td>0.555</td>
<td>5.000</td>
<td>2.500</td>
<td>0.003</td>
<td>1.908</td>
<td>1.803</td>
</tr>
</tbody>
</table>

Table 3 Properties of A₁, A₂ and A₃ bulks [31]

<table>
<thead>
<tr>
<th>Bulb code</th>
<th>b/B ratio</th>
<th>l/b ratio</th>
<th>cross sectional area ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>A₁</td>
<td>b = C_B</td>
<td>l = 1.2</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td>B = 2.5</td>
<td>b</td>
<td></td>
</tr>
<tr>
<td>A₂</td>
<td>b = C_B</td>
<td>l = 1.2</td>
<td>0.09</td>
</tr>
<tr>
<td></td>
<td>B = 3.5</td>
<td>b</td>
<td></td>
</tr>
<tr>
<td>A₃</td>
<td>b = C_B</td>
<td>l = 1.2</td>
<td>0.07</td>
</tr>
<tr>
<td></td>
<td>B = 4.5</td>
<td>b</td>
<td></td>
</tr>
</tbody>
</table>

l: length between the endpoint of bulb and the forward perpendicular
b: width of the largest width of bulb

It was seen that the A₁ and A₂ bulbs started to become effective after the speed of 9 knots (Fn 0.331) while the A₃ bulb started to become effective after the service speed of 10 knots (Fn 0.367), and the A₂ was more effective than the A₁ [31].

At the bulb application in 148/1 coded model, Soylemez [31] stated that the most effective bulb is A₂. The maximum width of the bulb A₂ is given by the equation (1a) and the maximum length of the bulb A₂ from the fore peak by the equation (1b).

\[
\frac{b}{B} = \frac{C_B}{3.5} \quad (1a)
\]

\[
\frac{l}{b} = 1.2 \quad (1b)
\]

The maximum widths and the maximum lengths of the bulbs of 148/3, 148/4, 148/8 and 148/9 coded boats are calculated by equation (1a) and equation (1b), respectively. The maximum bulb lengths of 148/8 and 148/9 coded boats are multiplied by the length correction coefficient 1.4285 (28.57/20.00). The maximum widths and the maximum lengths of the bulbs are given in Table 4.

Table 4 Maximum widths and lengths of the bulbs

<table>
<thead>
<tr>
<th>Maximum bulb width [mm]</th>
<th>148/3</th>
<th>148/4</th>
<th>148/8</th>
<th>148/9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum bulb length [mm]</td>
<td>1</td>
<td>795</td>
<td>972</td>
<td>1131</td>
</tr>
<tr>
<td>148/3</td>
<td>663</td>
<td>810</td>
<td>660</td>
<td>805</td>
</tr>
<tr>
<td>148/4</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>148/8</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>148/9</td>
<td></td>
<td>1380</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Delta, nabla and elliptical bulb forms are designed for each fishing boat form according to the maximum width and length given in Table 4. All of the forms are modeled in the RhinoCeros program in three dimensions.

Forms of boats with bulb have been named for the purpose of making observations and comparisons easier to follow. The delta (D), nabla (N) and elliptical (E) bulb forms of 148/3, 148/4, 148/8 and 148/9 coded fishing boats are named 148/3-D, 148/3-N, 148/3-E, 148/4-D, 148/4-N, 148/4-E, 148/8-D, 148/8-N 148/8-E, 148/9-D, 148/9-N and 149/3-E, respectively.

The non-dimensional offset values of the delta, nabla and elliptical type bulbs are given in Table 5, Table 6 and Table 7, respectively. The displacements of all generated boats for use in CDF analyses are given in Table 8.

<table>
<thead>
<tr>
<th>Table 5 Non-dimensional offset values of the delta type bulbs</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>148/3-D b_k</td>
</tr>
<tr>
<td>148/4-D b_k</td>
</tr>
<tr>
<td>148/8-D b_k</td>
</tr>
<tr>
<td>148/9-D b_k</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 6 Non-dimensional offset values of the nabla type bulbs</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>148/3-N b_k</td>
</tr>
<tr>
<td>148/4-N b_k</td>
</tr>
<tr>
<td>148/8-N b_k</td>
</tr>
<tr>
<td>148/9-N b_k</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 7 Non-dimensional offset values of the elliptical type bulbs</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
</tr>
<tr>
<td>148/3-E b_k</td>
</tr>
<tr>
<td>148/4-E b_k</td>
</tr>
<tr>
<td>148/8-E b_k</td>
</tr>
<tr>
<td>148/9-E b_k</td>
</tr>
</tbody>
</table>
Table 8 The displacements of all generated boats

<table>
<thead>
<tr>
<th>Boat No.</th>
<th>$\Delta_1$ [kN]</th>
<th>Boat No.</th>
<th>$\Delta_2$ [kN]</th>
<th>Boat No.</th>
<th>$\Delta_3$ [kN]</th>
<th>Boat No.</th>
<th>$\Delta_4$ [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>148/3</td>
<td>1065.55</td>
<td>148/4</td>
<td>1304.79</td>
<td>148/8</td>
<td>1515.93</td>
<td>148/9</td>
<td>1848.77</td>
</tr>
<tr>
<td>148/3-D</td>
<td>1082.19</td>
<td>148/4-D</td>
<td>1319.21</td>
<td>148/8-D</td>
<td>1538.25</td>
<td>148/9-D</td>
<td>1870.26</td>
</tr>
<tr>
<td>148/3-N</td>
<td>1082.02</td>
<td>148/4-N</td>
<td>1319.14</td>
<td>148/8-N</td>
<td>1538.13</td>
<td>148/9-N</td>
<td>1870.18</td>
</tr>
<tr>
<td>148/3-E</td>
<td>1082.33</td>
<td>148/4-E</td>
<td>1319.46</td>
<td>148/8-E</td>
<td>1538.47</td>
<td>148/9-E</td>
<td>1870.36</td>
</tr>
</tbody>
</table>

As an example, the lines plan of 148/3 coded fishing boat is shown in Figure 2. The fore cross sections plan and bulb profile are shown in Figure 3, Figure 4 and Figure 5 for 148/3-D, 148/3-N and 148/3-E coded fishing boat, respectively. The three-dimensional models of all generated fishing boat forms are shown in Figure 6.

Fig. 2 Lines plan of the 148/3 coded fishing boat [29]
A Systematic Investigation of the Effects of Various Bulbous Bows on Resistance of Fishing Boats

Dursun Saral, Muhsin Aydin
Ercan Kose

Fig. 3 Fore cross sections plan and bulb profile of the 148/3-D code fishing boat

Fig. 4 Fore cross sections plan and bulb profile of the 148/3-N code fishing boat

Fig. 5 Fore cross sections plan and bulb profile of the 148/3-E code fishing boat

Fig. 6 Three dimensional models of fishing boats
3. Numerical modelling

3.1 Governing equations

In this study, an Unsteady Reynolds-Averaged Navier-Stokes (URANS) method is used to solve the governing equations. These mass and momentum conservation equations are solved by the commercial CFD software STAR-CCM+. The averaged continuity and momentum equations for incompressible flows are given in tensor notation and Cartesian coordinates by equation 2 and equation 3.

$$\frac{\partial (\rho \overline{u}_i)}{\partial x_i} = 0$$  \hspace{1cm} (2)

$$\frac{\partial (\rho \overline{u}_i)}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho \overline{u}_i \overline{u}_j + \rho \overline{u}_i \overline{u}_j \right) = - \frac{\partial p}{\partial x_i} + \frac{\partial \overline{\tau}_{ij}}{\partial x_j}$$  \hspace{1cm} (3)

where $\rho$ is density, $\overline{u}_i$ is the averaged Cartesian components of the velocity vector, $\rho \overline{u}_i \overline{u}_j$ is the Reynolds stresses and $p$ is the mean pressure. $\overline{\tau}_{ij}$ is the mean viscous stress tensor components, as shown in equation 4.

$$\overline{\tau}_{ij} = \mu \left( \frac{\partial \overline{u}_i}{\partial x_j} + \frac{\partial \overline{u}_j}{\partial x_i} \right)$$  \hspace{1cm} (4)

in which $\mu$ is the dynamic viscosity.

3.2 Turbulence model

The "Realizable $k-\varepsilon$ Model" developed by Shih et al. [19] is the most advanced version of the $k-\varepsilon$ model.

There are two basic differences from the standard $k-\varepsilon$ model. The first is that the model contains a new transport equation for the turbulence loss rate $\varepsilon$. Second, $C_\mu$, a critical coefficient of the model, is expressed as a function of the mean flow and turbulence properties rather than being fixed as in the standard model. The understanding of an $C_\mu$ variable is also compatible with the experimental data in boundary layer.

Shih et al. [19] developed transport equations are as follows:

$$\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho k u_j) = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu + \frac{\mu_t}{\sigma_k}}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k$$  \hspace{1cm} (5)

$$\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[ \left( \frac{\mu + \frac{\mu_t}{\sigma_\varepsilon}}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} + C_{1\varepsilon} \frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon$$  \hspace{1cm} (6)
In this equation $G_k$ is the turbulent kinetic energy production due to the average velocity gradients, $G_b$ is the production of turbulence kinetic energy depending on the density changes due to temperature differences, $Y_M$ constrictive turbulence shows the effect of the expansion in the turbulence to the whole spread. The terms $S_K$ and $S_\varepsilon$ are user-defined source terms.

3.3 Performing resistance analyses using the CFD method

Star-CCM+ software calculates the total force on the surface; Normal and tangential forces, i.e., pressure and friction (shear) forces [33].

The force on a surface is computed as:

$$f = \sum_f \left( f_{f_{\text{Pressure}}}^{f_{\text{Pressure}}} + f_{f_{\text{Shear}}}^{f_{\text{Shear}}} \right) n_f$$

(7)

where $f_{f_{\text{Pressure}}}^{f_{\text{Pressure}}}$ and $f_{f_{\text{Shear}}}^{f_{\text{Shear}}}$ are the pressure and shear force vectors on the surface face $f$, and $n_f$ is a user-specified direction vector that indicates the direction in which the force should be computed.

4. Test results

In this study, 148/3, 148/4, 148/8 and 148/9 coded models from ITU Fishing Boats Series hull forms, which carried out model tests by Kafalı et al. [29], are examined. The information of the models and test conditions are given in Table 9. The characteristics of seawater and test basin water are given in Table 10. The results of the resistance tests of the models are given in Table 11.

### Table 9 Model information and test conditions [29]

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Date of test</th>
<th>148/3B</th>
<th>148/4B</th>
<th>148/8B</th>
<th>148/9B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometrical similarity ratio</td>
<td>$\alpha$</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Length of Waterline (LWL [m])</td>
<td></td>
<td>2.000</td>
<td>2.000</td>
<td>2.857</td>
<td>2.857</td>
</tr>
<tr>
<td>Length Between Perpendiculars (LPP [m])</td>
<td></td>
<td>2.000</td>
<td>2.000</td>
<td>2.857</td>
<td>2.857</td>
</tr>
<tr>
<td>Draught (T [m])</td>
<td></td>
<td>0.229</td>
<td>0.229</td>
<td>0.229</td>
<td>0.229</td>
</tr>
<tr>
<td>Wet Surface Area (S [m²])</td>
<td></td>
<td>1.250</td>
<td>1.341</td>
<td>1.794</td>
<td>1.908</td>
</tr>
<tr>
<td>Displacement Force ($\Delta m$ [kN])</td>
<td></td>
<td>1.039</td>
<td>1.272</td>
<td>1.478</td>
<td>1.803</td>
</tr>
<tr>
<td>Basin Water Temperature (t [°C])</td>
<td></td>
<td>16.00</td>
<td>16.00</td>
<td>18.00</td>
<td>16.50</td>
</tr>
<tr>
<td>Form Factor</td>
<td>$k$</td>
<td>0.518</td>
<td>0.554</td>
<td>0.281</td>
<td>0.296</td>
</tr>
</tbody>
</table>

### Table 10 Properties of freshwater and seawater [34]

<table>
<thead>
<tr>
<th></th>
<th>Freshwater</th>
<th>Freshwater</th>
<th>Freshwater</th>
<th>Seawater</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (t)</td>
<td>16.0</td>
<td>16.5</td>
<td>18.0</td>
<td>15.0</td>
</tr>
<tr>
<td>Mass density (ρ) [kg/m³]</td>
<td>998.9461</td>
<td>998.8634</td>
<td>998.5986</td>
<td>1026.0210</td>
</tr>
<tr>
<td>Kinematic viscosity (ν) [m²/s]</td>
<td>1.1093E-06</td>
<td>1.0950E-06</td>
<td>1.0542E-06</td>
<td>1.1892E-06</td>
</tr>
<tr>
<td>Dynamic viscosity (µ) [Pa.s]</td>
<td>0.001108</td>
<td>0.001094</td>
<td>0.001053</td>
<td>0.001220</td>
</tr>
</tbody>
</table>
### Table 11 Model resistance test data

<table>
<thead>
<tr>
<th>148/3B</th>
<th>148/4B</th>
<th>148/8B</th>
<th>148/9B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_m$ [m/s]</td>
<td>$R_{Tm}$ [kgf]</td>
<td>$V_m$ [m/s]</td>
<td>$R_{Tm}$ [kgf]</td>
</tr>
<tr>
<td>0.25</td>
<td>0.0305</td>
<td>0.25</td>
<td>0.0330</td>
</tr>
<tr>
<td>0.50</td>
<td>0.1065</td>
<td>0.50</td>
<td>0.1150</td>
</tr>
<tr>
<td>0.75</td>
<td>0.2310</td>
<td>0.75</td>
<td>0.2575</td>
</tr>
<tr>
<td>1.00</td>
<td>0.4150</td>
<td>1.00</td>
<td>0.5000</td>
</tr>
<tr>
<td>1.25</td>
<td>0.6605</td>
<td>1.25</td>
<td>0.9850</td>
</tr>
<tr>
<td>1.50</td>
<td>1.1150</td>
<td>1.50</td>
<td>1.8700</td>
</tr>
<tr>
<td>1.70</td>
<td>2.2360</td>
<td>1.70</td>
<td>3.4500</td>
</tr>
</tbody>
</table>

The total resistance values of the full-scale boats are calculated by the methods of Froude [27] and Hughes [28] from the model resistance test data. The total resistance values, which are calculated by the Froude method for 148/3, 148/4, 148/8 and 148/9 coded boats, are given in Table 12. The total resistance values, which are calculated by the Hughes method, are given in Table 13.

### Table 12 Total resistance values obtained by Froude method

<table>
<thead>
<tr>
<th>148/3</th>
<th>148/4</th>
<th>148/8</th>
<th>148/9</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_n$</td>
<td>$R_{Tn}$ [kN]</td>
<td>$F_n$</td>
<td>$R_{Tn}$ [kN]</td>
</tr>
<tr>
<td>0.056</td>
<td>0.1966</td>
<td>0.056</td>
<td>0.2166</td>
</tr>
<tr>
<td>0.113</td>
<td>0.7126</td>
<td>0.113</td>
<td>0.7720</td>
</tr>
<tr>
<td>0.169</td>
<td>1.6039</td>
<td>0.169</td>
<td>1.8182</td>
</tr>
<tr>
<td>0.226</td>
<td>2.9919</td>
<td>0.226</td>
<td>3.7615</td>
</tr>
<tr>
<td>0.282</td>
<td>4.9033</td>
<td>0.282</td>
<td>8.0444</td>
</tr>
<tr>
<td>0.339</td>
<td>8.8286</td>
<td>0.339</td>
<td>16.2584</td>
</tr>
<tr>
<td>0.384</td>
<td>19.5344</td>
<td>0.384</td>
<td>31.5449</td>
</tr>
</tbody>
</table>

### Table 13 The total resistance values obtained by the Hughes method

<table>
<thead>
<tr>
<th>148/3</th>
<th>148/4</th>
<th>148/8</th>
<th>148/9</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_n$</td>
<td>$R_{Tn}$ [kN]</td>
<td>$F_n$</td>
<td>$R_{Tn}$ [kN]</td>
</tr>
<tr>
<td>0.056</td>
<td>0.1572</td>
<td>0.056</td>
<td>0.1705</td>
</tr>
<tr>
<td>0.113</td>
<td>0.5844</td>
<td>0.113</td>
<td>0.6220</td>
</tr>
<tr>
<td>0.169</td>
<td>1.3465</td>
<td>0.169</td>
<td>1.5171</td>
</tr>
<tr>
<td>0.226</td>
<td>2.5688</td>
<td>0.226</td>
<td>3.2666</td>
</tr>
<tr>
<td>0.282</td>
<td>4.2803</td>
<td>0.282</td>
<td>7.3157</td>
</tr>
<tr>
<td>0.339</td>
<td>7.9732</td>
<td>0.339</td>
<td>15.2579</td>
</tr>
<tr>
<td>0.384</td>
<td>18.4706</td>
<td>0.384</td>
<td>30.3005</td>
</tr>
</tbody>
</table>
5. Method of calculation

5.1 Boundary conditions

In this study, the dimensions of the calculation volume were determined with reference to the recommended dimensions for the flow problems around the ship, and the recommended dimensions were taken from the manual of the CFD program [33].

The computational domain is dimensioned according to the LBP (the length between the fore and the aft perpendiculars of the ship) with reference to the intersection of aft perpendicular and loaded water line of the ship as shown in Figure 7. The volume of control was selected to be of rectangular prism. The dimension of the computational domain is 100x50x75 m for 148/3 and 148/4 coded boats. 143x72x107 m is the dimension of the computational domain for 148/8 and 148/9 coded boats.

As shown in Figure 7, the surfaces of the rectangular prism, which defines boundaries in the computational domain, are named Inlet, Outlet, Top, Bottom, Symmetry and Side. The surfaces, which are represented the ship form, are named Hull.

![Diagram showing the dimensions of the computational domain and the names of the surfaces](image)

Fig. 7 The dimensions of the computational domain and the names of the surfaces

The boundary conditions of the regions called Inlet, Outlet, Top, Bottom, Symmetry, Side and Boat in Figure 7 are given in Table 14.

The velocity inlet is defined as boundary condition for Inlet, because the flow enters the computational domain on the \(-x\) direction. At the Top, Bottom and Side borders, the flow velocity is equal to the potential flow, so the boundary condition is equivalent to the velocity inlet boundary condition. The symmetry plane boundary condition is used to indicate that symmetry of the computational domain is also on the other side of the Symmetry boundary. As a result of the events occurring within the computational domain, the boundary condition
of Outlet is selected the pressure outlet, because the values such as speed, pressure are not known at the boundary. The wall is assigned to the Hull as boundary condition, and it is assumed that the flow velocity components on the ship surface are zero (no-slip boundary condition).

<table>
<thead>
<tr>
<th>Boundary Name</th>
<th>Boundary Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>Velocity Inlet</td>
</tr>
<tr>
<td>Outlet</td>
<td>Pressure Outlet</td>
</tr>
<tr>
<td>Top</td>
<td>Velocity Inlet</td>
</tr>
<tr>
<td>Bottom</td>
<td>Velocity Inlet</td>
</tr>
<tr>
<td>Symmetry</td>
<td>Symmetry Plane</td>
</tr>
<tr>
<td>Side</td>
<td>Velocity Inlet</td>
</tr>
<tr>
<td>Hull</td>
<td>Wall</td>
</tr>
</tbody>
</table>

5.2 Design of mesh structure

In this study, the rectangular prismatic mesh structure is chosen because it gives better results than other mesh structures at the free water surface flow.

An average of 600 thousand, 1.2 million, 2.8 million, 3.7 million, 4.7 million and 7.4 million cells were created in the computational domains. It has been found that the resistance values, which are obtained in mesh structures with over 2.8 million cells, have not changed or that the change has not had much effect on the solution. For this reason, the CFD calculations are made with an average of 2.8 million cells and 8.6 million surfaces for 148/3, 148/3-D, 148/3-N, 148/3-E, 148/4, 148/4-D, 148/4-N and 148/4-E coded fishing boats. The CFD calculations are made with an average of 3.8 million cells and 11.4 million surfaces for 148/8, 148/8-D, 148/8-N, 148/8-E, 148/9, 148/9-D, 148/9-N and 148/9-E coded fishing boats.

The mesh structure is produced for 112 computational domains. As an example, the mesh structure of the 148/3 coded boat is shown in Figure 8 and Figure 9.

![Fig. 8 Boat surface mesh structure](image)
5.3 Solution method

After the mesh structure was established, the physical conditions were determined. The solution model is modeled in a 2-phase fluid environment (seawater and air) as in the real environment, and the loaded waterline of fishing boat is free water surface. The physical properties of seawater and air, which is used in calculations, are given in Table 15.

| Physical properties of seawater and air |
|-----------------|--------------|
| Seawater        | Air          |
| Mass density (ρ) [kg/m³] | 1026.02100  | 1.18415       |
| Dynamic viscosity (µ) [Pa.s] | 1.220×10⁻³ | 1.855×10⁻⁵   |

The value of gravity acceleration (gravity) is entered as 9.8067 m/s² in the direction of -z axis in order to be able to create a gravitational effect on the computational domain as in the world.

The VOF (Volume of Fluid) method is chosen as the surface capture method because of the effects of free surface water. The VOF method is included the effects of free surface water in the analyses. The results of total resistance can be obtained more accurately via this method. It was developed by Hirt and Nichols [35]. It gives accurate results in high degree nonlinear free surface problems such as wave breaks. In addition, this method is preferred for calculations of ship wave because it provides flexibility and convenience in mesh production.

The implicit unsteady is chosen as the computation time in order to avoid the time-dependent interactions of the phases and increasing the accuracy rates of the results. Realizable k-ε Model is chosen as the turbulent flow model. Segregated Flow is chosen as the solution algorithm because it provides ideal results in incompressible and multiphase flows.

The implicit unsteady is selected as the calculation time. The time step of the solution for each analysis is calculated according to equation 8 [36]. In this equation U is the velocity, Δt is the time step, and Δx is the minimum mesh cell length. As a result of various assays, the value of Courant-Friedrichs-Lewy (CFL) is set at 5. The determined time steps with physical time, number of iteration and computation time, approximately, are given in Table 16.

\[
CFL = \frac{U \Delta t}{\Delta x}
\]  

(8)
Table 16  Time steps of the solver, physical time, number of iteration and computation time

<table>
<thead>
<tr>
<th>Time Step</th>
<th>Physical Time [second]</th>
<th>Number of Iteration</th>
<th>Computation Time [hour]</th>
</tr>
</thead>
<tbody>
<tr>
<td>148/3, D, N, E</td>
<td>0.056</td>
<td>150</td>
<td>10</td>
</tr>
<tr>
<td>148/4, D, N, E</td>
<td>0.113</td>
<td>200</td>
<td>18</td>
</tr>
<tr>
<td>148/8, D, N, E</td>
<td>0.169</td>
<td>350</td>
<td>120</td>
</tr>
<tr>
<td>148/9, D, N, E</td>
<td>0.226</td>
<td>300</td>
<td>240</td>
</tr>
<tr>
<td>148/10, D, N,</td>
<td>0.282</td>
<td>300</td>
<td>288</td>
</tr>
<tr>
<td>E</td>
<td>0.339</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.384</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In this study, the maximum number of internal iterations for each time step is 10. CFD analyses are made at different speeds. Implicit unsteady flow is defined in the CFD program. Therefore, it would be wrong to evaluate for convergence criteria according to the convergence of residues or a fixed physical time. Considering that the CFD analyzes are consistent with the test values, the convergence criteria for the CFD analyzes is accepted that the change in values after a certain number of iterations is below 0.01 at low speeds and below 0.1 at high speeds.

The total resistance values vary depending on the physical time due to the fact that time-dependent variable flow is defined in the CFD program. Therefore, the total resistance values, at which calculations are terminated, are not used to directly comparison. Depending on the physical time, at high speeds which the resistance value fluctuations are excessive, for more accurate result, ship length is divided by the speed at which the resistance is calculated. In this way, the duration of a flow particle to cross the length of the ship is found. And, the total physical time is divided into pieces according to this duration. The arithmetic mean of the last three total resistance values from the time of convergence of the solution is the final total resistance value for comparisons. Thus, both the accuracy of the convergence is controlled and the physical time-independent resistance values are obtained.

6. Results and discussions

Firstly, the total resistance values which are obtained by CFD analyses are compared with the total resistance values obtained from the Froude and Hughes methods. Then, the CFD results of boat forms with and without bulb are compared between themselves. Finally, it has been determined that which type of bulb is more beneficial for the fishing boats by taking into account the increase or decrease on the total resistance values of the forms.

6.1 Comparison of CFD results with test results

The CFD analyses are performed on Froude numbers which are determined for 148/3, 148/4, 148/8 and 148/9 coded boats. The compatibility of the CFD results and the test results is examined by comparing the ship total resistance values obtained from the CFD analyses with the ship total resistance values obtained from Froude and Hughes methods. The percentage difference between the CFD value and the Froude method value is found by the equation_9. The percentage difference between the CFD value and the Hughes method value is found by the equation_10.
A Systematic Investigation of the Effects of Various Bulbous Bows on Resistance of Fishing Boats

Dursun Saral, Muhsin Aydin
Ercan Kose

\[ \text{Difference Percentage} = \frac{\text{CFD Value} - \text{Froude Method Value}}{\text{Froude Method Value}} \times 100 \] (9)

\[ \text{Difference Percentage} = \frac{\text{CFD Value} - \text{Hughes Method Value}}{\text{Hughes Method Value}} \times 100 \] (10)

According to Froude numbers, total resistance values, which are obtained from Froude and Hughes methods, and the difference percentages of CFD values than Froude method values, and the difference percentages of CFD values than Hughes method values are given in Table 17, 18, 19, 20 for 148/3, 148/4, 148/8, 148/9 coded boats, respectively.

As can be seen from Table 17, 18, 19 and 20, according to Froude method, the arithmetic mean value of the difference percentages is around 13% while according to Hughes method the average is about 4%. When each analysis is evaluated in its own group, CFD results are more compatible with the total resistance values, which are obtained from the test data, at low speeds. It is also seen that the results on ships with low \( C_B \) are more consistent than the results on ships with high \( C_B \).

Table 17 The comparison between the total resistance values, which are obtained from Froude and Hughes methods, and the total resistance values, which are obtained CFD analyses, for 148/3 coded boat

<table>
<thead>
<tr>
<th>Fn</th>
<th>( V_S ) [knot]</th>
<th>Froude Method</th>
<th>Hughes Method</th>
<th>CFD</th>
<th>According to Froude Method</th>
<th>According to Hughes Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( R_T ) [kN]</td>
<td>( R_T ) [kN]</td>
<td>( R_T ) [kN]</td>
<td>%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>0.056</td>
<td>1.537</td>
<td>0.1966</td>
<td>0.1572</td>
<td>0.1594</td>
<td>-18.91</td>
<td>1.42</td>
</tr>
<tr>
<td>0.113</td>
<td>3.073</td>
<td>0.7126</td>
<td>0.5844</td>
<td>0.5760</td>
<td>-19.17</td>
<td>-1.43</td>
</tr>
<tr>
<td>0.169</td>
<td>4.610</td>
<td>1.6039</td>
<td>1.3465</td>
<td>1.3413</td>
<td>-16.37</td>
<td>-0.39</td>
</tr>
<tr>
<td>0.226</td>
<td>6.147</td>
<td>2.9919</td>
<td>2.5688</td>
<td>2.5628</td>
<td>-14.34</td>
<td>-0.23</td>
</tr>
<tr>
<td>0.282</td>
<td>7.684</td>
<td>4.9033</td>
<td>4.2803</td>
<td>4.4054</td>
<td>-10.15</td>
<td>2.92</td>
</tr>
<tr>
<td>0.339</td>
<td>9.220</td>
<td>8.8286</td>
<td>7.9732</td>
<td>7.8735</td>
<td>-10.82</td>
<td>-1.25</td>
</tr>
<tr>
<td>0.384</td>
<td>10.450</td>
<td>19.5344</td>
<td>18.4706</td>
<td>17.3691</td>
<td>-11.08</td>
<td>-5.96</td>
</tr>
</tbody>
</table>

Table 18 The comparison between the total resistance values, which are obtained from Froude and Hughes methods, and the total resistance values, which are obtained CFD analyses, for 148/4 coded boat

<table>
<thead>
<tr>
<th>Fn</th>
<th>( V_S ) [knot]</th>
<th>Froude Method</th>
<th>Hughes Method</th>
<th>CFD</th>
<th>According to Froude Method</th>
<th>According to Hughes Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( R_T ) [kN]</td>
<td>( R_T ) [kN]</td>
<td>( R_T ) [kN]</td>
<td>%</td>
<td>%</td>
<td></td>
</tr>
<tr>
<td>0.056</td>
<td>1.537</td>
<td>0.2166</td>
<td>0.1705</td>
<td>0.1636</td>
<td>-24.47</td>
<td>-4.06</td>
</tr>
<tr>
<td>0.113</td>
<td>3.073</td>
<td>0.7720</td>
<td>0.6220</td>
<td>0.6049</td>
<td>-21.64</td>
<td>-2.75</td>
</tr>
<tr>
<td>0.169</td>
<td>4.610</td>
<td>1.8182</td>
<td>1.5171</td>
<td>1.4557</td>
<td>-19.94</td>
<td>-4.05</td>
</tr>
<tr>
<td>0.226</td>
<td>6.147</td>
<td>3.7615</td>
<td>3.2666</td>
<td>3.0006</td>
<td>-20.23</td>
<td>-8.14</td>
</tr>
<tr>
<td>0.282</td>
<td>7.684</td>
<td>8.0444</td>
<td>7.3157</td>
<td>6.5601</td>
<td>-18.45</td>
<td>-10.33</td>
</tr>
<tr>
<td>0.384</td>
<td>10.450</td>
<td>31.5449</td>
<td>30.3005</td>
<td>26.0765</td>
<td>-17.34</td>
<td>-13.94</td>
</tr>
</tbody>
</table>

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Table 19 The comparison between the total resistance values, which are obtained from Froude and Hughes methods, and the total resistance values, which are obtained CFD analyses, for 148/8 coded boat

<table>
<thead>
<tr>
<th>Fn</th>
<th>V_S</th>
<th>Froude Method</th>
<th>Hughes Method</th>
<th>CFD</th>
<th>According to Froude Method</th>
<th>According to Hughes Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[knot]</td>
<td>Ft [kN]</td>
<td>Ft [kN]</td>
<td>Ft [kN]</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>0.047</td>
<td>1.537</td>
<td>0.2186</td>
<td>0.1985</td>
<td>0.1934</td>
<td>-11.52</td>
<td>-2.56</td>
</tr>
<tr>
<td>0.094</td>
<td>3.073</td>
<td>0.7627</td>
<td>0.6972</td>
<td>0.6943</td>
<td>-8.97</td>
<td>-0.37</td>
</tr>
<tr>
<td>0.142</td>
<td>4.610</td>
<td>1.6997</td>
<td>1.5669</td>
<td>1.5538</td>
<td>-8.58</td>
<td>-0.84</td>
</tr>
<tr>
<td>0.189</td>
<td>6.147</td>
<td>3.1587</td>
<td>2.9398</td>
<td>2.9331</td>
<td>-7.14</td>
<td>-0.23</td>
</tr>
<tr>
<td>0.246</td>
<td>7.991</td>
<td>5.6142</td>
<td>5.2683</td>
<td>5.3006</td>
<td>-5.59</td>
<td>0.61</td>
</tr>
<tr>
<td>0.302</td>
<td>9.835</td>
<td>9.7795</td>
<td>9.2822</td>
<td>9.2407</td>
<td>-5.51</td>
<td>-0.45</td>
</tr>
<tr>
<td>0.378</td>
<td>12.294</td>
<td>25.6370</td>
<td>24.9015</td>
<td>22.9937</td>
<td>-10.31</td>
<td>-7.66</td>
</tr>
</tbody>
</table>

Table 20 The comparison between the total resistance values, which are obtained from Froude and Hughes methods, and the total resistance values, which are obtained CFD analyses, for 148/9 coded boat

<table>
<thead>
<tr>
<th>Fn</th>
<th>V_S</th>
<th>Froude Method</th>
<th>Hughes Method</th>
<th>CFD</th>
<th>According to Froude Method</th>
<th>According to Hughes Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[knot]</td>
<td>Ft [kN]</td>
<td>Ft [kN]</td>
<td>Ft [kN]</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>0.047</td>
<td>1.537</td>
<td>0.2170</td>
<td>0.1932</td>
<td>0.1937</td>
<td>-10.74</td>
<td>0.24</td>
</tr>
<tr>
<td>0.094</td>
<td>3.073</td>
<td>0.7752</td>
<td>0.6972</td>
<td>0.7077</td>
<td>-8.70</td>
<td>1.51</td>
</tr>
<tr>
<td>0.142</td>
<td>4.610</td>
<td>1.7191</td>
<td>1.5621</td>
<td>1.6078</td>
<td>-6.48</td>
<td>2.93</td>
</tr>
<tr>
<td>0.189</td>
<td>6.147</td>
<td>3.3454</td>
<td>3.0866</td>
<td>3.1368</td>
<td>-6.24</td>
<td>1.63</td>
</tr>
<tr>
<td>0.246</td>
<td>7.991</td>
<td>7.7329</td>
<td>7.3240</td>
<td>7.2269</td>
<td>-6.54</td>
<td>-1.33</td>
</tr>
<tr>
<td>0.368</td>
<td>11.987</td>
<td>35.0224</td>
<td>34.1909</td>
<td>29.9300</td>
<td>-14.54</td>
<td>-12.46</td>
</tr>
</tbody>
</table>

When the velocity increases, it is observed that the percentage of difference between CFD results and test data rises while calculating the total resistance of the ship with CFD because of the difficulty in accurately modeling the turbulence phenomenon.

Turbulence density, turbulence velocity scale and turbulence viscosity ratio values are taken constant for each ship form and speed in the ship resistance calculation problems with CFD. In this study, the constants, which are suggested by the instruction manual of CFD program [33], are adopted for the values of turbulence intensity, turbulence velocity scale and turbulence viscosity ratio. Block coefficient of 148/3 and 148/8 coded boats are averages 0.405 while block coefficient of 148/4 and 148/9 coded boats are averages 0.495. The total resistance values of 148/4 and 148/9 coded boats, i.e., at the boats with high CB value, are calculated with a greater percentage of difference than the total resistance values of 148/3 and 148/8 coded boats because the flow around the underwater forms of 148/4 and 148/9 coded boats is more turbulent.

CFD analyses of forms with delta, nabla and elliptical bulb are carried out with the program settings and constants, which are used in CFD analyses for 148/3, 148/4, 148/8 and 148/9 coded boats.
6.2 Comparison of CFD results for the boats without bulb and with bulb

The total resistance values are calculated by performing CFD analyses for 148/3, 148/3-D, 148/3-N, 148/3-E, 148/4, 148/4-D, 148/4-N, 148/4-E, 148/8, 148/8-D, 148/8-N, 148/8-E, 148/9, 148/9-D, 148/9-N and 148/9-E coded fishing boats at the determined Fn values. The difference percentages of the total resistance values of forms with bulb according to forms without bulb are calculated according to equation (11).

\[
\text{Difference Percentage} = \frac{R_T \text{ (with bulb)} - R_T \text{ (without bulb)}}{R_T \text{ (without bulb)}} \times 100
\]

(11)

The comparison between total resistance of the forms with bulb and total resistance of the forms without bulb are done via Equation (11). Thus, it is determined that, how much increase (+) or decrease (-) are on the total resistance according to the bulb shapes.

The values of friction resistance and pressure resistance, which constitute the total resistance of the ship, are shown in Figure 10, 11, 12 and 13 for 148/3, 148/4, 148/8 and 148/9 coded boats, respectively.

As can be seen from Figure 10, Figure 11, Figure 12 and Figure 13 frictional resistance is higher than pressure resistance at low Fn values while pressure resistance is higher than frictional resistance at high Fn values. While the frictional resistance is higher at boat forms with bulb according to forms without bulb, the pressure resistance is less at forms with bulb than at forms without bulb. In general, the most significant decrease in pressure resistance is seen by elliptical type bulb and this is followed by nabla and delta type bulbs.

![Fig. 10 The total resistance values, which are obtained using CFD, for 148/3, 148/3-D, 148/3-N and 148/3-E coded boats](image-url)
Fig. 11 The total resistance values, which are obtained using CFD, for 148/4, 148/4-D, 148/4-N and 148/4-E coded boats.

Fig. 12 The total resistance values, which are obtained using CFD, for 148/8, 148/8-D, 148/8-N and 148/8-E coded boats.
The total resistance values, which are obtained using CFD, for 148/9, 148/9-D, 148/9-N and 148/9-E coded boats at the specified Fn numbers, increase (+) or decrease (-) percentages of the total resistance values of the 148/3-D, 148/3-N and 148/3-E coded boats with respect to 148/3 coded boat are shown in Figure 14.

According to Fn values, increase (+) or decrease (-) percentages of the total resistance values of 148/3-D, 148/3-N and 148/3-E in reference to 148/3.
As can be seen in the Figure 14 for 148/3 coded fishing boat, delta, nabla and elliptical bulbs start to become effective after value of \( F_n \) 0.158, \( F_n \) 0.261 and \( F_n \) 0.246 value, respectively. In other words; the delta, nabla and elliptical bulb are beginning to provide benefit after 4.3, 7.1 and 6.7 knot speeds, respectively. The three bulb types also provide the maximum benefit at value of \( F_n \) 0.339, namely, at the speed of 9.2 knots. At value of \( F_n \) 0.384, i.e., at a speed of 10.50 knots, the efficiency of the bulbs is somewhat lower than that of \( F_n \) 0.339. At 10 knot service speed and higher speeds, the elliptical type bulb provides the most benefit. While delta and nabla bulb have same benefit at the service speed, the delta bulb more useful than the nabla bulb at low speeds. It is the type of elliptical bulb that provides the most benefit at service speed.

At the specified \( F_n \) numbers, increase (+) or decrease (-) percentages of the total resistance values of the 148/4-D, 148/4-N and 148/4-E coded boats with respect to 148/4 coded boat are shown in Figure 15.

![Graph showing the difference in resistance values](image)

**Fig. 15** According to \( F_n \) values, increase (+) or decrease (-) percentages of the total resistance values of 148/4-D, 148/4-N and 148/4-E in reference to 148/4.

As can be seen in the Figure 15 for 148/4 coded fishing boat, delta, nabla and elliptical bulbs start to become effective after value of \( F_n \) 0.189, \( F_n \) 0.213 and \( F_n \) 0.200, respectively. In other words; the delta, nabla and elliptical bulb are beginning to provide benefit after 5.1, 5.8 and 5.4 knot speeds, respectively. The nabla and elliptical type bulb provide the maximum benefit at value of \( F_n \) 0.282, namely, at the speed of 7.6 knots. The delta bulb type also provides the maximum benefit at value of \( F_n \) 0.339, namely, at the speed of 9.2 knots. It is also seen that the nabla bulb form is more useful than the elliptical type bulb at the speed range of 6.7-9.2 knots. At the service speed of 10 knots and and higher speeds, the elliptical type of bulb is the most beneficial, while the nabla type bulb is more beneficial than the elliptical type bulb at speed of between 6.7 and 9.2 knots.

At the specified \( F_n \) numbers, increase (+) or decrease (-) percentages of the total resistance values of the 148/8-D, 148/8-N and 148/8-E coded boats with respect to 148/8 coded boat are shown in Figure 16.
According to Fn values, increase (+) or decrease (-) percentages of the total resistance values of 148/8-D, 148/8-N and 148/8-E in reference to 148/8.

As can be seen in the Figure 16 for 148/8 coded fishing boat, delta, nabla and elliptical bulbs start to become effective after value of Fn 0.204, Fn 0.257 and Fn 0.246, respectively. In other words; the delta, nabla and elliptical bulb are beginning to provide benefit after 6.6, 8.4 and 8.0 knot speeds, respectively. The three bulb types also provide the maximum benefit at value of Fn 0.378, namely, at the speed of 12.3 knots. In addition, the delta bulb form performs better than the elliptical type bulb at low Fn numbers. While at the 10 knot service speed and higher speeds the elliptical type bulb provides more benefits, the delta type bulb provides more benefits at lower speeds. The three types of bulbs are the same benefit at service speed, but at lower speeds it appears that the delta type bulb is more useful than the other types of bulbs.

At the specified Fn numbers, increase (+) or decrease (-) percentages of the total resistance values of the 148/9-D, 148/9-N and 148/9-E coded boats with respect to 148/9 coded boat are shown in Figure 17.

As can be seen in the Figure 17 for 148/9 coded fishing boat, delta, nabla and elliptical bulbs start to become effective after value of Fn 0.177, Fn 0.213 and Fn 0.205, respectively. In other words; the delta, nabla and elliptical bulb are beginning to provide benefit after 5.8, 6.9 and 6.7 knot speeds, respectively. The three bulb types also provide the maximum benefit at value of Fn 0.302, namely, at the speed of 9.8 knots. It is also seen that the nabla bulb form is more useful than the elliptical type bulb at the speed range of 7.6-11.5 knots. While the elliptical type bulb is more useful than the nabla type bulb at the speed range of 0.0-7.6 knots, the nabla type bulb is more useful than the elliptical type bulb at the speed range of 7.6-11.5 knots. After the speed of 11.5 knots, the elliptical type bulb is more useful.
When all of the percentages of increase and decrease in total resistance are evaluated together, it turns out that the most suitable bulb form for 148/3, 148/4, 148/8 and 148/9 coded fishing boats is the elliptical bulb type at service speed of 10 knots and at higher speeds.

According to \( C_B \), when the efficiency of the bulb is evaluated at the service speed of 10 knots:

- At the \( C_B = 0.405 \), value of the benefit of the bulbs is 10% on average.
- At the \( C_B = 0.495 \), value of the benefit of the bulbs is 13% on average.

The higher the \( C_B \) value, the greater the benefit that the bulbs have at the service speed. Also, as the \( C_B \) value increases, the speed range at which the bulbs maximum benefit is also increasing.

When the effectiveness of bulbs is evaluated according to \( L/B \) ratio:

- It has been found that the bulbs have started to benefit at lower speeds in boats with the \( L/B \) ratio of 3.5 compared to boats with the \( L/B \) ratio of 5.0.
- At 10 knots service speed, it has been determined that the bulbs benefit at average rate of 11% at boats with the \( L/B \) ratio of 3.5, and at average of 10% at boats with the \( L/B \) ratio of 5.0.

It has been found that the bulbs have started to benefit at lower speeds in boats with the \( L/B \) ratio of 3.5 compared to boats with the \( L/B \) ratio of 5.0. At the \( L/B \) ratio of 3.5, bulbs have been found to be more beneficial in service speed.

When the efficiencies are evaluated according to types of bulbs:

- It has been found that the delta type bulb is beginning to provide benefit after the speed of 5.4 knots. It has an average benefit of 8.9% at the service speed.
- It has been found that the nabla type bulb is beginning to provide benefit after the speed of 7.0 knots. It has an average benefit of 11.0% in service speed.
• It has been found that the elliptical type bulb is beginning to provide benefit after the speed of 6.7 knots. It has an average benefit of 11.2% in service speed.

At different low speeds, the delta, nabla and elliptical type bulbs cause 4%, 14% and 8% increase in total resistance, respectively.

As an example, the wave deformations in the boat symmetry plane and the wave deformations on the free water surface at Fn 0.339 of the 148/3, 148/3-D, 148/3-N and 148/3-E coded fishing boats are shown in Figure 18 and 19, respectively.

Fig. 18 Wave deformations occurring in the boat symmetry plane of 148/3, 148/3-D, 148/3-N and 148/3-E coded boats at Fn 0.339

Fig. 19 Wave deformations at the free water surface of 148/3, 148/3-D, 148/3-N and 148/3-E coded boats at Fn 0.339

7. Conclusions

The delta, nabla and elliptical type bulbs are applied to 148/3, 148/4, 148/8 and 148/9 coded boats from ITU Fishing Boats Series in order to find out which type of bulb is more effective on the fishing boats. In order to inspect the accuracy of the CFD analyses, the total resistance values, which are obtained by the Froude and Hughes methods from the test results of the 148/3, 148/4, 148/8 and 148/9 coded boats, are compared with the total resistance values which are obtained by CFD. The CFD analyses of 148/3, 148/4, 148/8 and 148/9 coded boats forms with delta, nabla and elliptic type bulb are performed. The total resistance values of the forms with bulbs and without bulbs are compared. The results are evaluated according to boat forms, C_B, L/B ratio and the efficiencies of the bulbs.

When the results are evaluated according to boat forms, it is seen that elliptic type bulb is found to be more useful than other type bulbs at range of 0-12 knots.
When the results are evaluated according to C_B, it has been found that the boats with high C_B have more benefit of bulb. Also, as the C_B increases, the ratio of the bulb efficiency increases.

When the results are evaluated according to the L/B ratio, it has been found that the bulbs have started to benefit at lower Fn values in boats with the low L/B ratio compared to boats with the high L/B ratio. Also, the efficiency rate is found to be higher.

When the results were evaluated according to the type of bulbs, it is seen that the delta type bulb starts to be benefit in lower Fn values and it is followed by elliptical and nabla type. On the contrary, around at the service speed and at higher speeds, the elliptical type bulb is more useful than other bulbs and it is followed by the nabla and the delta type.

When all the evaluations are considered, it seems that the most suitable bulb is the elliptical type bulb for fishing boats.

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RELIABILITY-BASED INSPECTION PLANNING OF THE SHIP STRUCTURE EXPOSED TO FATIGUE DAMAGES

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Original scientific paper

Summary

Reliability-based inspection planning is one of the most popular methods in determining the time of inspection and repairs in various structures. In this way, inspection and repair times are determined mainly by putting a lower limit for the reliability index. The detection and measurement of cracks is one of the possible outputs at the time of inspecting fatigue cracking. One way to use this output is to update the parameters of the fatigue reliability equation. In this study, statistical distribution of the parameters of the problem is updated and fatigue reliability is calculated for inspection planning using the Bayesian updating concept through the Markov Chain Monte Carlo (MCMC) method and the Metropolis–Hasting algorithm. The distribution of crack growth equation material parameters and the initial crack length will be updated with this method. The application of the proposed method has been shown in a structural member of a ship.

Key words: Reliability-based; inspection planning; Fatigue failure; crack growth parameters; Bayesian updating; Markov Chain-Monte Carlo.

1. Introduction

Various structures such as marine structures are aging over time. Fatigue cracking is one of the main factors that can reduce the strength of the structural members of a ship with the passage of time, thus increasing the probability of failure. To maintain the safety of ship structures, some maintenance activities over the lifetime of ships should be conducted. These activities are carried out in the form of the inspection of different parts of the ship at certain times, and, if necessary, by repairing them.

For inspection planning of structures, various methods are provided, with reliability-based inspection planning being one of the most famous methods. In this way, the reliability index is calculated over the lifetime of the structure. The degree of structural reliability reduces over time. In this planning method, usually a lower limit for reliability is defined as the target reliability index to control the structural safety a ship. The first inspection is planned for the structure after reliability reaches the limit value. This process is schematically shown in Figure 1.
No crack detection, crack detection without size measurement, and crack detection with size measurement are the three possible outputs of the inspection of the damage caused by fatigue cracking. Several studies (e.g. [1&2]) have worked on the use of the results of inspections under fatigue failure. In these studies, mostly Bayesian theory and equations, including the equations proposed by Madsen in [3], have been used to update the failure probability and its corresponding reliability index. In these research studies, random variables of fatigue reliability equations after repairing the structural components are considered equal to the amounts prior to the repair. This means that the results of the inspection are not used to update the distribution of random variables of the problem. Accordingly, using the Bayesian approach, the reliability can be calculated after the first inspection and then the second inspection can be scheduled.

An alternative approach for the above-mentioned procedure is to update the random variables of the problem based on inspection results and then to calculate the structural reliability for planning the next inspection time. Little research has been carried out in this regard so far.

Garbatov and Guedes Soares [4] proposed a Bayesian approach to update some of the parameters of the probability distributions governing the reliability assessment of the maintained floating structures. Based on the time-dependent fatigue reliability and using the information from the inspections, the description of the time to crack initiation, crack growth law, and the probability of crack detection were updated. Heredia-Zavoni and Montes-Iturrizaga [5] investigated a Bayesian framework using an analytical model for updating the initial crack size distribution at a certain point in time for the tubular joints of fixed offshore structures. The crack size measurement is used for updating as the new data. Soliman and Frangopol [6] proposed a Bayesian updating approach to find the updated distributions of fatigue crack growth model parameters based on inspection results. The updated parameters were used to find the next inspection time based on a cost-based optimization approach.

In this study, the first structural inspection time is planned by calculating the fatigue reliability. After that, by assuming that cracks can be measured with different lengths at the time of the first inspection, the distribution of the parameters of the fatigue reliability equation has been updated with the Markov Chain Monte Carlo method by using the Bayesian updating concept. Finally, the next inspection time is planned after calculating the fatigue reliability using the updated parameters.

One of the features of the proposed approach is that it can be used in minimizing aging effect of ship structure integrity. The effect of fatigue cracks on aging effect of ship structures was previously investigated in some articles. By updating the distribution of effective
parameters, a more accurate approximation of the parameters governing the problem can be studied and then the aging effects of structures can be investigated more accurately.

2. Fatigue Failure

According to linear elastic fracture mechanics, the Paris–Erdogan equation provides the crack growth as:

\[
\frac{da}{dN} = C (\Delta K)^m
\]  

(1)

Where \(a\) is the crack size, \(N\) is the number of loading cycles, and \(m\) and \(C\) are material constants. \(\Delta K\) is the stress intensity range, which is given in the following:

\[
\Delta K = \Delta \sigma K(a) \sqrt{\pi a}
\]  

(2)

Here \(\Delta \sigma\) is the applied stress range and \(K(a)\) is the geometry function. Integrating the above equation, the number of cycles of stress applying that is required for the crack growth from sizes \(a_1\) to \(a_2\) can be obtained from the following equation:

\[
N_{1-2} = \int_{a_1}^{a_2} \frac{da}{C (\Delta \sigma K(a) \sqrt{\pi a})^m}
\]  

(3)

Owing to random wave-induced loads, variable amplitude fatigue loading is inherent to ship structures, thus adding complexity to predicting the crack growth. Therefore, simplified models are preferred [7]. An equivalent constant amplitude stress range is a simplified method that renders the same degree of damage (i.e. crack growth) as the variable amplitude loading. It can be expressed as [8]:

\[
\Delta \sigma_{eq} = \left[ \int_0^{\infty} (\Delta \sigma)^\beta f_{\Delta \sigma}(\Delta \sigma) d(\Delta \sigma) \right]^{1/\beta}
\]  

(4)

Where \(\beta\) is the interaction coefficient and \(f_{\Delta \sigma}(\Delta \sigma)\) is the probability density function of the stress range.

By integrating Equation 1 and using an equivalent stress range, the following safety margin can be used for calculating the fatigue reliability as:

\[
G = \int_{a_0}^{a_c} \frac{da}{(K(a) \sqrt{\pi a})^m} - \Delta \sigma_{eq}^m C N
\]  

(5)

Where \(a_0\) and \(a_c\) are initial and critical crack lengths. The probability of failure \(P_f\) and the reliability index \(\beta\) can be obtained as:

\[
P_f = P[G < 0] = \Phi(-\beta)
\]  

(6)

Where \(\Phi(\cdot)\) is the standard normal distribution function.

Reliability evaluation can be performed with approximate methods like First Order Reliability Method (FORM), Second Order Reliability Method (SORM) or Monte Carlo simulation (MCS). In the present study, the probability of failure and the corresponding reliability indices are calculated by the MCS method.
The Weibull distribution is an adequate approximation for the long-term stress range in ship structural members. Hence, the m-th statistical moment of the long-term stress range can be expressed in terms of Scale $A$ and shape parameters $B$ of the Weibull distribution as:

$$\Delta \sigma_{eq}^m = A^m \Gamma(1+\frac{m}{B})$$  \hspace{1cm} (7)

In this study, with Bayesian updating, it has been tried to obtain the updated distribution of the model parameters, including $a_0$, $m$, and $C$, using new data of the crack size measured in the inspection time. By updating these parameters, Equation 5 is rewritten as:

$$G_{upd} = \int_{a_{0,upd}}^{a_c} \frac{da}{(K(a)\sqrt{\pi a})^{m_{upd}}} \Delta \sigma_{eq}^{m_{upd}} C_{upd} N$$  \hspace{1cm} (8)

Where $a_{0,upd}$ is the updated initial crack length, and $m_{upd}$ and $C_{upd}$ are the updated material constants.

3. Bayesian theorem

The Bayesian theory has been widely used to update the parameters of various equations. This theory can be rewritten to update the required parameters of the reliability equation. Based on this, the updated or posterior distribution of the parameters can be obtained by combining the prior distribution of the parameters with new information on the crack size obtained from the structural inspection. The Bayesian theory can be rewritten based on probability distribution functions (PDFs). Hence, in this regard, $R$ and $S$ represent the model parameter and measured parameter, respectively [9]:

$$f_R(r | S = s) = \frac{f_S(r | R = r)f_R(r)}{f_S(s)}$$  \hspace{1cm} (9)

Where $f_R(r)$ is the prior PDF of the model parameters, $f_S(s)$ is the PDF of the measurement, $f_S(s|R=r)$ is the likelihood function of obtaining the measured values conditioned on the estimated model parameters, and $f_R(r|S=s)$ is the updated or posterior PDF of the model parameters. The denominator in the above equation is the normalization constant [10] and thus it can be removed from the equation as:

$$f_R(r | S = s) \propto f_R(r)f_S(r | R = r)$$  \hspace{1cm} (10)

3.1 Bayesian updating

It is assumed that a member has been subjected to fatigue failure and the Paris–Erdogan equation governs the growth of cracks. The general model for crack growth prediction can be expressed as:

$$a_d = M(N, x; y) + e$$  \hspace{1cm} (11)

Where $a_d$ is the measured value of the crack size and $e$ is the general error term that includes the measurement error $\epsilon$ and the modeling error $\varepsilon$. $M(N, x; y)$ is the crack growth model in which $x$ is the model parameter for updating and $y$ is the model independent variable. $N$ in the above equation is a non-random variable such as the number of applied stress cycles or time.
Model parameters in the problem include $a_0$, $m$, and $\ln(C)$ (natural logarithm of the material parameter $C$) that create the model parameter vector $x=\{\ln(C), m, a_0\}$. The error terms $\varepsilon$ and $\tau$ are usually considered as normal variables with zero mean and finite variances of $s_\varepsilon^2$ and $s_\tau^2$. The likelihood function has been defined as [11]:

$$L(a_d|x) = \frac{1}{\sqrt{2\pi}\sigma_\varepsilon} \exp \left( -\frac{1}{2} \frac{(a_d - M(N,x;y))^2}{\sigma_\varepsilon^2} \right)$$

(12)

The general error term $\varepsilon \sim \text{Normal}(0,s_\varepsilon^2)$ can be substituted instead of measurement and modelling errors, and, then, the likelihood function can be defined as:

$$L(a_d|x) = \frac{1}{\sqrt{2\pi}\sigma_\varepsilon} \exp \left( -\frac{1}{2} \frac{(a_d - M(N,x;y))^2}{\sigma_\varepsilon^2} \right)$$

(13)

This function represents the probability of detecting a crack with size $a_d$ after the $N$ loading cycles provided the variable input vector $x$. If the crack size is measured "n" times in different cycles for a particular component, the likelihood function can be defined as:

$$L(a_{d,1},a_{d,2},...,a_{d,n}|x) = \frac{1}{(\sqrt{2\pi}\sigma_\varepsilon)^n} \exp \left( -\frac{1}{2} \frac{\sum_{i=1}^{n} (a_{d,i} - M(N_i,x))^2}{\sigma_\varepsilon^2} \right)$$

(14)

Here $a_{d,i}$ are measured values of crack sizes at $N_i$ corresponding loading cycles. Putting the likelihood function in Equation 10, it is possible to calculate the posterior or updated distribution of model parameters.

### 3.2 Markov Chain Monte Carlo (MCMC)

It is difficult to find an analytical solution of Equation 10 in general terms and that is why simulation-based methods, such as the MCMC method, are used. Here the cascade Metropolis–Hasting algorithm has been used. The Markov chain is formed in this method by iterative sampling. The convergence of the chain is among the features of the Markov chain regardless of the starting point. To update the parameter $x$, it starts from an initial value $x_1$ in a chain and the value $x_{i+1}$ is produced in a way that is independent of $x_1, ..., x_{i-1}, x_i$. This process is as the following [9]:

1. Each chain starts at $i=1, x_1$.
2. Generate a random value $x^*$ using the transition or proposal density function, $q(x|x_i)$.

   The proposal distribution function $q$ is usually zero mean normal or uniform density function; therefore, it is a symmetric function.

   $$q(x^*|x_i) = q(x^* - x_i) = q(x_i - x^*)$$

(15)

3. Prior acceptance of the probability evaluation is given as the following:

   $$\alpha_p = \min \left\{ 1, \frac{f_x(x^*)q(x_i|x^*)}{f_x(x_i)q(x^*|x_i)} \right\}$$

(16)

Considering the symmetry of proposal density function:
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\[
\alpha_p = \min \left\{ 1, \frac{f_x(x^*)}{f_x(x_i)} \right\} \tag{17}
\]

Where \( f_i(x) \) is the prior PDF of input parameters.

4. Calculation of \( u_p \sim \text{Uniform}[0,1] \)
   - If \( u_p > \alpha_p(x_i, x^*) \), accept and go to Step 5.
   - Otherwise, go to Step 2.

5. The likelihood acceptance of probability evaluation is given as:

\[
\alpha_L = \min \left\{ 1, \frac{L_x(x^*)}{L_x(x_j)} \right\} \tag{18}
\]

Where \( L_x \) is the likelihood given in Equations 13 and 14.

6. Calculation of \( u_L \sim \text{Uniform}[0,1] \)
   - If \( u_L > \alpha_L(x_i, x^*) \) accept and set \( x_{i+1} = x^*, i+1 = i \) and go to Step 2.
   - Otherwise go to Step 2.

This process should be repeated to achieve the desired number of elements in the Markov chain. To make the chain independent of its initial value, the initial portion of the chain, which is known as the burn-in portion, will be discarded.

The convergence of the chain is the important issue in using this method. Various methods have been proposed to evaluate the convergence of the Markov chain. Here the approach provided by [12] is used. According to this method, multiple chains should be created with different starting points. Variances within each chain are compared with the variances between chains [9]. The large difference between these two values indicates a lack of convergence. If there are \( v \) chains with \( n \) elements (which burn-in period is discarded), the variance between chains, \( B \), is obtained as:

\[
B = \frac{1}{n} \sum_{j=1}^{v} (\bar{X}_j - \bar{X})^2 \tag{19}
\]

In the above equation, \( \bar{X}_j \) is the average value of the \( j \)-th chain, and \( \bar{X} \) is the average value of all chains. The within chain variance is calculated as:

\[
W = \frac{1}{v(n-1)} \sum_{j=1}^{v} \sum_{i=1}^{n} (X_{ji} - \bar{X}_j)^2 \tag{20}
\]

Where \( X_{ji} \) is the \( i \)-th element of the \( j \)-th chain. The variance ratio \( R \) is defined as:

\[
R = \frac{\bar{V}}{W} \tag{21}
\]

\[
\bar{V} = \frac{n-1}{n} W + \frac{v+1}{vn} B \tag{22}
\]

Accordingly, if chains converge to the target distribution, \( R \) should be close to 1. According to [12], the \( R \) value less than 1.1 or 1.2 represents the convergence of the method. Here we consider the amount of 1.2 as the convergence criteria. Initial samples with the variance ratio greater than 1.2 will be discarded, while the other samples will be used to describe the posterior or updated distribution function.
4. Verification example

Virkler et al. [13] reported the results of a set of crack growth experiments on the Aluminium 2024 alloy. A total of 68 centre-through crack specimens with the same loading, geometry, and material were tested. The specimens were tested with the initial length of crack \(a_0=9\) mm, the width \(w=154.2\) mm, the thickness \(d=2.54\) mm, and under a constant tension of 48.82 MPa. The geometry function for Virkler samples is as:

\[
K\left(\frac{a}{w}\right) = \sqrt{\sec\left(\frac{\pi a}{w}\right)}
\]

Several researchers have analysed the results of these tests. In [11], a total of 15 crack growth curves (Figure 2) were selected for achieving the prior distribution of \(\{\ln(C),m\}\). Also, according to the correlation between these two variables, a bivariate normal distribution was proposed as:

\[
p_0(\ln(C),m) = \frac{1}{2\pi\sqrt{\Sigma}} \exp\left(-\frac{1}{2} \begin{bmatrix} \ln C + 26.7060 \\ m - 2.9684 \end{bmatrix}' \Sigma^{-1} \begin{bmatrix} \ln C + 26.7060 \\ m - 2.9684 \end{bmatrix} \right)
\]

\[
\Sigma = \begin{bmatrix} 0.5335 & -0.0886 \\ -0.0886 & 0.0148 \end{bmatrix}
\]

\(\Sigma\) is the covariance matrix of these two variables. According to the prior distribution of the material parameters, the crack growth curve is shown in Figure 3. On this curve, the average and the range of 95\% confidence interval (CI) have been shown.

![Fig. 2 Experimental data for parameter identification of Virkler tests [11]](image1)

![Fig. 3 Crack growth curve using prior distribution of parameters](image2)

To update the parameters, additional data on the crack size is required. This additional data can be obtained through inspection or health monitoring. Here three points of a crack growth curves provided by [11] are used as measured crack sizes in different cycles for updating. This is shown in Table 1. In the three steps, each paired data of Table 1 has been used to perform the update. As many as four chains with 30,000 elements are formed to update the parameters. Here \(s_e=0.2\) mm is considered. After updating, the distribution of parameters with respect to each of the points 1 to 3 is shown in Figure 4. The mean vector and the covariance matrix associated with prior and posterior distributions are obtained in Table 2.

Crack growth curves using posterior distributions are shown in Figure 5. Comparing Figures 3 and 5, it can be seen that using additional data on the crack size, the average value, as shown in Figure 5, is inclined towards the additional crack size information based on the
data given in Table 1, while the range of CI of 95% decreases. These two effects show the reduced uncertainty of crack growth parameters.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Additional information of crack growth size for model parameter updating [11]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Point</td>
<td>Cycles number N</td>
</tr>
<tr>
<td>1</td>
<td>38485</td>
</tr>
<tr>
<td>2</td>
<td>64064</td>
</tr>
<tr>
<td>3</td>
<td>85164</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Prior and posterior distributions of crack growth parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Point used</td>
<td>Mean vector {ln(C),m}</td>
</tr>
</tbody>
</table>
| Prior | \{-26.7060,2.9684\} | \[
| 1 | \{-27.4128,3.0767\} | \[
| 2 | \{-27.8356,3.1404\} | \[
| 3 | \{-28.0052,3.1656\} | \[

In this example, the critical crack length is considered equal to 49.8 mm. Considering the constant loading stress and using Equation 5, the failure probability curves are provided based on prior and posterior distributions in Figure 6. As is evident from this figure, the use of updated parameters based on the additional data on crack size has a very significant effect on failure probability values in various cycles.

5. Application example

To illustrate the application of model parameter updating in reliability-based inspection planning, a ship structural detail at the intersection between deck transverse and deck longitudinal is considered. This detail is selected from an article of [7] and shown in Figure 7. The critical crack depth is assumed to be the flange thickness. It is assumed that this detail is located in deck structure of a tanker with 237 m length.

The geometry function for a semi-elliptical crack shape is defined as [7]:

\[ K(a) = K_E K_S K_T K_G \]  

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\[ K_E = \text{basic crack shape factor} = \left[ 1 + 4.5945 \left( \frac{a}{2c} \right)^{1.65} \right]^{0.5} \]  

(26)

\[ \frac{a}{2c} = 1.65 \]

Fig. 5 Crack growth curve using posterior distribution of parameters related to: (a) Point 1, (b) Point 2, (c) Point 3.

\[ K_S = \text{front face factor} = 1 - 0.16 \left( \frac{a}{2c} \right) \]  

(27)

\[ K_T \left( x = a/2c ; y = a/h \right) = \text{finite thickness factor} = 1 + \left( 0.008 y x^{-2.454} + 0.0534 y^2 x^{-1.005} \right) \]  

(28)
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\[ K_G = \text{Stress gradient factor} = \frac{SCF}{1 + \left(\frac{a}{h}\right)^{0.6051}} \frac{1}{1.158} \]  

(29)

Here SCF is the stress concentration factor, which is considered to be equal to 2.1. Also, the crack aspect ratio is assumed to be \(2c=6.71+2.58a\).

All necessary data for predicting crack size and calculating reliability index is provided in Table 3. The initial crack size is assumed to follow a normal distribution with mean value of 0.5 mm and standard deviation of 0.05 mm [6]. The natural logarithm of the material parameter \(C (ln(C))\) is treated as normal distribution random variable with a mean value of -29.97 and standard deviation of 0.514 [14] while the material parameter \(m\) is assumed to follow a normal distribution with a mean value of 3.0 and standard deviation of 0.15. The correlation coefficient between \(m\) and \(ln(C)\) is assumed to be -0.9 [15].

It is assumed that the long-term stress range of the detail following the Weibull distribution. Parameters of stress range can be calculated from DNV [16]. For a detail located in deck structure, the Weibull shape parameter can be calculated as:

\[ B = 2.21 - 0.54 \log_{10}(L) \]  

(30)

The Weibull scale parameter \(A\) can be defined as:

\[ A = \frac{\Delta\sigma_0}{\left(\ln n_0\right)^{1/h}} \]  

(31)

Here, \(\Delta\sigma_0\) is the reference stress range value at the local detail, which exceeded once out of no cycles, and \(n_0\) is the total number of cycles associated with the stress range level \(\Delta\sigma_0\). In the present study, it is assumed that \(\Delta\sigma_0\) is equal to 125 MPa. Shape and scale parameters are respectively obtained as 0.9276 and 13.288. The number of cycles is defined as \(N=n_0 t\) that \(t\) is the time and \(n_0\) is the long-term average zero crossing frequency. The frequency \(n_0\) can be obtained as [16]:

\[ n_0 = \frac{1}{4.\log_{10}(L)} \]  

(32)

The frequency \(n_0\) is calculated as 0.1053 Hz and hence the annual number of cycles is 3.32e6.

![Ship structural detail and crack location](image-url)
5.1 Estimation of first inspection time

The reliability index over time is calculated by Equation 5 and shown in Figure 8. The first inspection is done at a time when the reliability index is equal to the target reliability index. Several studies have focused on the establishment of the target reliability index. Mansour [17] proposed values for the target reliability index in accordance with Table 4.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Notation (units)</th>
<th>Mean</th>
<th>Standard deviation</th>
<th>Distribution type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial crack size</td>
<td>$a_0$ (mm)</td>
<td>0.5</td>
<td>0.05</td>
<td>Normal</td>
</tr>
<tr>
<td>Material parameter</td>
<td>$Ln(C)$</td>
<td>-29.97</td>
<td>0.514</td>
<td>Normal</td>
</tr>
<tr>
<td>Annual number of cycles</td>
<td>$N_{an}$ (cycles/year)</td>
<td>$3.32 \times 10^6$</td>
<td>-</td>
<td>Deterministic</td>
</tr>
</tbody>
</table>

The first inspection times for different values of the target reliability index are given in Table 5. Also, the mean values of crack size at the first inspection time are presented in Table 5. In this study, the point 2.5 has been considered as the target reliability index. Based on prior distributions of the parameters, the structure must be inspected in 6.72 years and the mean value of crack size at this time is 2.07 mm.

<table>
<thead>
<tr>
<th>Table 4: Target reliability indices [17]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Consequences</td>
</tr>
<tr>
<td>Not serious</td>
</tr>
<tr>
<td>Serious</td>
</tr>
<tr>
<td>Very serious</td>
</tr>
</tbody>
</table>

Fig. 8 Fatigue reliability index of the selected structural detail

<table>
<thead>
<tr>
<th>Table 5: First inspection times</th>
</tr>
</thead>
<tbody>
<tr>
<td>Target reliability index</td>
</tr>
<tr>
<td>3.0</td>
</tr>
<tr>
<td>2.5</td>
</tr>
<tr>
<td>2.0</td>
</tr>
</tbody>
</table>

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5.2 Model parameter updating and next inspection time

As noted earlier, the parameters $\ln(C)$, $m$, and $a_0$ are updated with the MCMC method using the Metropolis–Hasting algorithm in this section. Prior distributions of these variables are presented in Table 3. Two Markov chains with 30,000 elements are considered. To investigate the convergence, the variances ratio is considered equal to 1.2. $s_c = 0.2$ mm is considered.

Additional information of crack size is required for parameter updating. Here it is assumed that cracks of various sizes have been detected at the first inspection time and thus the effect of the measurement of the cracks with different sizes on the posterior distribution of the parameters is investigated. It is assumed that cracks with lengths equal to 1, 1.5, 2.5, 3, and 4 mm during the first inspection time are detected.

A sample curve of the variance ratio associated with the detection of a crack with the length of 1.0 mm at the first inspection time is shown in Figure 9. Based on this figure, the initial 10,000 elements are discarded from the chain, while the others are used to describe the posterior distribution. Prior and posterior distributions of the parameters are shown in Figure 10. The mean values and standard deviations of the posterior distributions based on different crack size measurement values are shown in Table 6. In general, it can be said that with updating, the standard deviation of parameters has decreased, which implies the reduced uncertainty in the parameters. Any crack detected with a length lower than the mean value of 2.07 mm increases the average value of $\ln(C)$ and decreases the average values $a_0$ and $m$ and vice versa. The larger crack is detected at the first inspection time, where the lower standard deviation of $\ln(C)$ and $m$ are observed. Also, the correlation coefficient between the two variables is also reduced. A similar process can be observed for verification.

![Fig. 9](image-url)  
**Fig. 9** Variance ratio in the Markov Chain Monte Carlo iterations: (a) R for mean values, (b) R for standard deviation values

The curve of mean crack size has been shown in Figure 11. Based on this figure, if a crack larger than 2.07 mm is detected, the slope of the crack growth curve increases and vice versa. The mean crack size at the time equal to first inspection time, 6.72 years, has been shown in the last column of Table 7 considering the updated parameters. As can be seen, considering the updated values of parameters, the mean value of crack size is very close to the assumed measured values of the cracks. For instance, when it is assumed that a crack with 3.0 mm is detected in the first inspection time and updating the parameters, the mean value of crack size at 6.72 years will be 2.95 mm. This shows that posterior distributions can predict the crack size with more accuracy.

After updating the parameters of the problem, fatigue reliability is calculated by using Equation 8 and shown in Figure 12. Assuming a target reliability index value of 2.5, the time interval between the first and second inspections is presented in Table 7.
Table 6: Prior and posterior distributions of crack growth parameters

<table>
<thead>
<tr>
<th>Crack size measured (mm)</th>
<th>Mean Value</th>
<th>Standard deviation</th>
<th>Correlation coefficient between ( \ln(C) ) and ( m )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( \ln(C) )</td>
<td>( m )</td>
<td>( a_0 ) (mm)</td>
</tr>
<tr>
<td>Prior</td>
<td>-29.97</td>
<td>3.00</td>
<td>0.500</td>
</tr>
<tr>
<td>1.0</td>
<td>-29.57</td>
<td>2.79</td>
<td>0.479</td>
</tr>
<tr>
<td>1.5</td>
<td>-29.74</td>
<td>2.88</td>
<td>0.498</td>
</tr>
<tr>
<td>2.5</td>
<td>-29.98</td>
<td>3.01</td>
<td>0.502</td>
</tr>
<tr>
<td>3.0</td>
<td>-30.08</td>
<td>3.06</td>
<td>0.503</td>
</tr>
<tr>
<td>4.0</td>
<td>-30.20</td>
<td>3.11</td>
<td>0.508</td>
</tr>
</tbody>
</table>

Fig. 10 Posterior distribution of parameters using inspection measurements: (a) \( \ln(c) \), (b) \( m \), (c) \( a_0 \)

Fig. 11 Curve of mean crack size using posterior distribution of parameters based on inspection measurements
From these results, it can be said that the larger the crack size measured at the time of inspection, the earlier is the re-inspection of the structure. Also, the second inspection time of the structure can be predicted with more accuracy by reducing the uncertainty of the parameters. Furthermore, due to the reduction in uncertainty in model parameters, by measuring the larger crack size the updated reliability curve decreases more rapidly. As expected, the size of the detected crack has a significant impact on the estimation of the second inspection time.

**Table 7:** Time interval between second and first inspection and mean crack size at time of first inspection

<table>
<thead>
<tr>
<th>Crack size measured (mm)</th>
<th>Time interval between inspections, ( t_{\text{insp2-\text{insp1}}} ) (year)</th>
<th>Mean value of crack size at 6.72 year (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prior</td>
<td>-</td>
<td>2.07</td>
</tr>
<tr>
<td>1.0</td>
<td>24.62</td>
<td>1.11</td>
</tr>
<tr>
<td>1.5</td>
<td>19.82</td>
<td>1.48</td>
</tr>
<tr>
<td>2.5</td>
<td>14.99</td>
<td>2.45</td>
</tr>
<tr>
<td>3.0</td>
<td>13.57</td>
<td>2.95</td>
</tr>
<tr>
<td>4.0</td>
<td>11.50</td>
<td>3.95</td>
</tr>
</tbody>
</table>

**Fig. 12** Fatigue reliability index based on time after first inspection

### 6. Conclusion

Fatigue is a major cause of failure in steel structures. Fatigue reliability assessment is one of the methods for scheduling the inspection times of a structure. The prior distributions of parameters in fatigue reliability formulae can be achieved by performing various experiments or through the existing rules. In conventional methods, these distributions are also used for evaluating the reliability after the repair. One of the outputs of the structural inspection is the detection and measurement of crack length. Here the results of measuring the crack length at the time of inspection are used to update the distribution of the parameters of fatigue reliability formula. For this, the Bayesian updating concept along with the Markov Chain Monte Carlo (MCMC) method with the Metropolis–Hastings algorithm has been used. In this paper, the proposed method is used to update the material parameters in the equation of crack growth \((\ln(C) \text{ and } m \text{ in the Paris–Erdogan equation})\) and initial crack length. From the results, the following items can be concluded:
1. Any update of the parameters reduces their uncertainty and thus the behaviour of the growth of cracks in the structure can be predicted with more certainty.
2. If during the inspection time of the structure there is a possibility to measure the crack length with high accuracy, then the structure lifetime and future inspection times can be estimated with higher accuracy.
3. The proposed approach can be used for other deteriorating mechanisms of ship structures such as corrosion.
4. During the inspection time of the structure, if cracks with much higher or lower than the predicted size are detected, they must be re-examined along with other conditions relating to the problem such as loading. In general terms, this method can also be used in the update of loading conditions.

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THE MOST ECONOMICAL CONFIGURATION OF PUSHED BARGE CONVOY SYSTEM THROUGH CAIRO-ASWAN WATERWAY

UDC 629.55:629.5.015.26:629.5.076:282.263.1
Professional paper

Summary

In the recent years, as a result of the economical situation in Egypt, fuel price is rapidly increased. Consequently, the cost of cargo transport is also increased. Therefore, the aim of the present paper is to check the feasibility of a pushed barge convoy system working through Cairo-Aswan waterway as an alternative mean of cargo transport in order to encourage the transport companies to transport their cargoes through river Nile which is considered the cheapest transport mode in Egypt.

In this paper, the current situation of the river transport and the characteristics of the Egyptian inland waterways network are reviewed and investigated to identify the navigation problems and constraints which affect the navigation through Cairo-Aswan waterway. The basic concept of ship controllability is reviewed to clarify the maneuver characteristics of a pushed barge convoy system in shallow waterways. Also, different configurations for pushed barge convoy system are proposed and the required power of each configuration is calculated using a specially developed computer program.

A technical and operational measure called transport efficiency is used as a criterion to get the most economical configuration of the desired pushed barge convoy system. Finally, a comparison between the pushed barge convoy system and the existing river transport means in Egypt is made to clarify the feasibility of using pushed barge convoy system through Cairo-Aswan waterway.

Key words: Barge Convoy System; Shallow Waterways; River Nile; Transport Efficiency

1. Introduction

Transport substantially shapes the growth and development of countries specially, from economic point of view. Although, Egypt enjoys a 1530 km long stretch of the river Nile, river transport has been sorely neglected compared to the other transport modes. This sector failed to account for more than 1% of total freight services (roads possessed the lion’s share with 93% and railways followed with 6%) [1]. That is due to the following reasons:
1. Interaction in the responsibilities between the ministry of transport and the ministry of irrigation,
2. The river transport facilities have been grossly neglected for many years. Also, many vessels are not adequately equipped so they cannot travel at night,
3. Many vessels are unable to navigate year-round, since the water levels vary in accordance with the old Nile flood season.
4. Many transport companies prefer land transport due to its reliability and door to door service.

In other countries, river transport accounts for a significantly larger share of freight services. In Netherlands the figure is 34%, in Belgium it is 26% and in Luxemburg it is 14% [1]. Nowadays, to sustain and enhance the economic growth and vitality, the Egyptian government has decided to emphasize making better use of river transport through supporting the efforts of river transport authority (RTA) to complete its infrastructure projects. Moreover, transport companies intended to increase the volume of the annual transported cargo through the Nile for the following reasons:

1. River transport is considered an important solution to overcome the congestion problems on the Egyptian motorways.
2. According to the Egyptian river transport authority, river transport is considered to be an environmentally-friendly and a safer mode of transport compared to railway and road (trucks on highways) transport modes [1].
3. River transport will have a competitive edge in terms of costs, especially as oil price subsidies are gradually phased out. This is mainly due to economies of scale (one barge with a capacity of 1,500 tons can transfer more than 50 trucks).

By studying the different means of river transport worldwide, it was found that the pushed barge convoy system is the most effective and progressive system of transport with minimal environmental problems. Therefore, the aim of the present paper is finding the most economical configuration of a pushed barge convoy system working through Cairo-Aswan waterway and checking its feasibility.

2. Pushed barge convoy system

The concept of pushed barge convoy consists of a pusher boat with one or more non-propelled barges. This concept is widely used in Europe and United State of America (USA), especially in the Danube, Elbe and Mississippi rivers [2]. From the economical point of view, more barges being used at one time to transport large quantity of goods seem to be more efficient [2]. Also, it has a lower maintenance & repair and docking costs. Moreover, the possibility of disengaging the pusher boat reduces the risk of total loss in case of collision, grounding or other serious accidents and consequently reduces the insurance cost [3]. In addition to the above mentioned advantages of the pushed barge convoy system, it is characterized by a higher exploitation time since push boat does not have to wait for loading and unloading operations. Therefore, we will study the feasibility of using a pushed barge convoy system for cargo transport through Cairo-Aswan waterway.

2.1 Configurations of pushed barge convoy systems

To understand and differentiate between the different configurations of a pushed barge convoy system, a naming system in the form “nmPB” is used. Where, PB refers to pushed barge convoy system; n and m refer to the numbers of barge columns and rows, respectively. Fig.1 shows naming system of 9 pushed barge convoy systems with different configurations [2].
2.2 Controllability of pushed barge convoy systems

Controllability is a very complex and important subject and represents an essential consideration in the design of any floating structure. It covers all aspects related to a ship’s trajectory, speed and orientation at sea as well as in shallow waters where positioning and course keeping are of particular concern. It is usually divided into three areas; course keeping, maneuvering and speed changing. For conventional ships, course keeping and maneuvering may tend to work against each other [4].

Performance of pushed barge convoy systems is affected by water depth, channel restrictions and interference with nearby units or obstacles. Therefore, controllability of pushed barge convoy systems is one of the most important considerations facing the naval architect and involves compromises with the other characteristics of the convoy. Many solutions are obtained by comparison with the characteristics of the previously examined designs. In other cases, experimental techniques, theoretical analyses and rational design practices must all come into play to assure adequacy.

In 2008, K. K. King et al [2] carried out an experimental work about manoeuvring simulations of pusher barge convoy systems and model tests were performed at the Hiroshima University Towing Tank on 9 different configurations with 1:50 model scale. In the experiments two types of barges were used; leading barge was always a rake barge and box barges are between the rake barge and the pusher boat. Each configuration was tested at the corresponding full-scale speed of 12.964 km/h and full-scale draft of 2.74 m. The results showed that, for the same number of barges, pushed barge convoy systems with long configurations (12PB, 13 PB, 23PB) require larger stopping distance & time, tactical diameter, advance distance and transfer distance than those required for wide configuration (21PB, 31PB, 32PB).

K. K. King et al [5] continued his work to study the effect of shallow water on manoeuvring characteristics of pushed barge convoy systems. One pushed barge configuration (11PB) was taken as the subject of his study and the shallow water tests were conducted at Kyushu University. In the experiments, three different water depths to ship’s draught ratios were performed. The results showed that, for different rudder angles, the largest turning circle occurs in deep water, followed by medium shallow and then shallow water. Therefore, the pushed barge convoy 11PB having a turning performance similar to wide beam vessel where turning trajectory decreases from deep to medium shallow to shallow water condition [5].
2.3 Convoy resistance and powering calculation

Several formulae exist for calculating the resistance and powering of seagoing ships. However, there is little in the literature concerning the calculation of the resistance and powering in rivers, specially pushed barge convoy systems navigating in waterway of somewhat limited cross section.

Resistance of a pushed barge convoy system depends on many geometrical and operational parameters such as: speed and draught of push boat, depth and width of the waterway, length and width of the barge train, and other indirect parameters related to the barge train.

Howe [6], Wang et al [7], Kaa [8] and Michalski [9] published their empirical methods. Some are based on a model test series that is not comprehensive, and some need additional data which are not easy to obtain in advance. Consequently, these inconveniences restrict the application of such methods.

An empirical method given by Marchal et al [10] may be used to estimate the resistance of a pushed barge convoy system; it had established a generalized relation between total resistance of pushed barge convoys and the geometrical characteristics of convoys and waterways. The range of application of this methodology given in Table 1 is relatively wide. The resistance of pushed barge convoy systems is estimated using Marchal’s equation [10] after:

\[
\frac{R_T}{\Delta} = \sum_{i=0}^{n_1} \sum_{j=0}^{n_2} \sum_{k=0}^{n_3} \sum_{l=0}^{n_4} \sum_{m=0}^{n_5} \sum_{n=0}^{n_6} \beta_{ijklmn} \cdot \left(2Fn_{Rh}\right)^i \cdot \left(10Fn_L\right)^j \cdot \left(M_{\frac{10}{10}}\right)^k \cdot \left(\frac{B_x}{10h-T_x}\right)^l \cdot \left(\frac{B_x}{B_c}\right)^m \cdot \left(\frac{T_x}{h}\right)^n
\]

Where, \(\beta_{ijklmn}\) is the coefficient of each term and the maximum exponents \(n_1, n_2, n_3, n_4, n_5,\) and \(n_6\) of these six variables are 3, 3, 2, 2, 2, and 1 in order of succession [10].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range of application</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Fn_{Rh})</td>
<td>0 - 0.81</td>
</tr>
<tr>
<td>(Fn_L)</td>
<td>0 - 0.18</td>
</tr>
<tr>
<td>(L_x/V_x^3)</td>
<td>4.79 - 16.73</td>
</tr>
<tr>
<td>(B_x/T_x)</td>
<td>3.5 - 28.5</td>
</tr>
<tr>
<td>(B_x/(h-T_x))</td>
<td>0.2 - 24.4</td>
</tr>
<tr>
<td>(B_x/B_c)</td>
<td>0.063 - 0.48</td>
</tr>
<tr>
<td>(T_x/h)</td>
<td>0.047 - 0.67</td>
</tr>
<tr>
<td>(A_x/A_c)</td>
<td>0.0033 - 0.18</td>
</tr>
</tbody>
</table>

Unfortunately, no experimental work had been previously carried out to for river Nile vessels. Therefore, in this paper, the convenience of the range of application of Marchal’s formula for Cairo-Aswan waterway is examined in order to adopt the approach to estimate the resistance of pushed barge convoy systems working through this waterway.

Effective power \( (P_E) \) can be calculated based on the calculated resistance of a pushed barge convoy and the desired service speed according to Eq. 2. Also, Eq. 3 can be used to calculate the required brake power \( (P_B) \) for a push boat of pushed barge convoy system navigates at certain speed. Existing push boats have a propulsive efficiency \( (\eta_p) \) varying from 0.30 to 0.45 [11]. It is obvious that, the value of the propulsive efficiency \( (\eta_p) \) is difficult to estimate without sufficient experimental work, but for the purpose of comparison in this study, it may be adequately assumed to be of 0.4.

\[
P_E = R_T \cdot V_s
\]

\[
P_B = \frac{P_E}{\eta_p}
\]

3. Cairo-Aswan waterway

Cairo-Aswan waterway is a first class waterway with a length of 980 km and a minimum water depth of 2.5 m [12]. Fig. 2 shows Cairo-Aswan waterway. It connects all Upper Egypt Governorates from Aswan to Cairo and is used for cargoes, passengers and tourists transport. Also, it is wide enough to host traffic from South to North and vice versa. The distance from Aswan Dam to Delta Bridges is divided into four sections and contains 3 locks & 21 bridges [12]. In the past, Assiut lock extremely affects the dimensions of Nile ships which navigate through Cairo-Aswan waterway. Recently, The Egyptian government is completely finished from the final stage of replacing and renovating the old Assiut lock by another one has a length of 160 m and width of 17 m.

![Cairo-Aswan waterway](image)

The right choice of ship’s speed should be decided based on the Froude depth number to avoid the critical region of ship resistance. In this work the Froude depth number is taken equal to 0.7 [13]. Therefore, ship speed through Cairo-Aswan waterway must be less than 12.6 km/h.
4. Feasibility of pushed barge convoy in Cairo-Aswan waterway

In this section, a comparison between the most economical configuration of pushed barge convoy and the existing river transport means is made to check the feasibility of using barge convoy system in Cairo-Aswan waterway. This comparison is carried out based on transport efficiency ($ET$) which is considered one of the most important technical and operating measures by which the cost of transport can be reduced and fleet made more competitive. $ET$ can be defined as tons-kilometres per kilowatt-hour according to the following formula [14]:

$$ET = \frac{Dwt \cdot V}{P_b}$$

(4)

Eq. 4 shows that, it is not only the required power that is taken into account, but also the weight of the transported cargo and the time required to move that weight (service speed). Another comparison among three different configurations (12PB, 21PB and 22PB) of a pushed barge convoy system is made to find the most economical configuration of pushed barge convoy system working through Cairo-Aswan waterway. This procedure was carried out using a specially developed computer program, see Fig. 3.

Pushed barges which are used in this application already exist in service and used in river transport through Cairo-Aswan waterway. Pushed barges and tug boat particulars are listed as shown in Table 2.
The convenience of Marchal’s formula for Cairo-Aswan waterway is checked for three different configurations and illustrated in Table 3. The calculated values of the parameters of the configurations of the hypothetical pushed barge convoy system working through Cairo-Aswan waterways conform compatible to the range of application of Marchal’s formula as shown in Table 1.

### Table 2 Particulars of the tug boat and pushed barges

<table>
<thead>
<tr>
<th>Items</th>
<th>Pushed Barges</th>
<th>Tug Boat</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>50 m</td>
<td>24 m</td>
</tr>
<tr>
<td>Breadth</td>
<td>7.5 m</td>
<td>7.5 m</td>
</tr>
<tr>
<td>Draft</td>
<td>1.6 m</td>
<td>1.2 m</td>
</tr>
<tr>
<td>Block Coefficient</td>
<td>0.88</td>
<td>0.6</td>
</tr>
<tr>
<td>Cargo Capacity</td>
<td>425 tons</td>
<td>-----</td>
</tr>
</tbody>
</table>

### Table 3 Adequacy of Marchal’s formula for Cairo-Aswan waterway

<table>
<thead>
<tr>
<th>Parameter</th>
<th>12PB</th>
<th>21PB</th>
<th>22PB</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{nR_h}$</td>
<td>0.62</td>
<td>0.655</td>
<td>0.655</td>
</tr>
<tr>
<td>$F_{nL}$</td>
<td>0.08</td>
<td>0.103</td>
<td>0.080</td>
</tr>
<tr>
<td>$L_x/V_x^3$</td>
<td>11.72</td>
<td>6.992</td>
<td>9.475</td>
</tr>
<tr>
<td>$B_x/T_x$</td>
<td>4.69</td>
<td>9.375</td>
<td>9.375</td>
</tr>
<tr>
<td>$B_x/(h-T_x)$</td>
<td>8.33</td>
<td>16.667</td>
<td>16.667</td>
</tr>
<tr>
<td>$B_x/B_c$</td>
<td>0.08</td>
<td>0.150</td>
<td>0.150</td>
</tr>
<tr>
<td>$T_x/h$</td>
<td>0.64</td>
<td>0.640</td>
<td>0.640</td>
</tr>
<tr>
<td>$A_x/A_c$</td>
<td>0.05</td>
<td>0.096</td>
<td>0.096</td>
</tr>
</tbody>
</table>

### 5. Results

The developed computer program is used to calculate transport efficiency ($E_T$) for pushed barge convoy system with three different configurations (12PB, 21PB and 22PB). The results are shown in Figs. 4 and 5.

![Figure 4: Total resistance of pushed barge convoy](image-url)
Fig. 4 shows that, in case of two pushed barge convoy, resistance of the wide configuration (21PB) is approximately double that of the long configuration (12PB). Also, Fig. 4 shows that resistance of four pushed barges convoy (22PB) is approximately 35% higher than that of two pushed barge convoy in wide configuration (21PB). While, Fig. 5 shows that, in case of two pushed barge convoy, transport efficiency of the long configuration (12PB) is approximately double that of the wide configuration (21PB). Also, Fig. 5 shows that transport efficiency of two pushed barge convoy in long configuration (12PB) is approximately 29% higher than that of four pushed barges convoy (22PB).

The above mentioned results show that, in case of two pushed barge convoy, the long configuration (12PB) gives less resistance than the wide configuration (21PB). Although, four pushed barges convoy (22PB) gives the highest resistance, it gives higher transport efficiency than two pushed barge convoy in wide configuration (21PB). However, two pushed barges convoy in long configuration (12PB) gives the highest transport efficiency. Therefore, it can be said that 12PB is the most economical configuration for a pushed barge convoy system working through Cairo-Aswan waterway.

Also, based on transport efficiency ($E_T$), feasibility of using the pushed barge convoy system in river Nile transport is checked by making a comparison between pushed barge convoy system and the existing means of inland transport in Egypt (self-propelled inland cargo ship and pusher barge – dump barge). Particulars of these means in are listed as shown in Table 4.

<table>
<thead>
<tr>
<th>Items</th>
<th>Self-propelled cargo ship</th>
<th>Pusher barge - dump barge</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Pusher barge</td>
</tr>
<tr>
<td>Length</td>
<td>100 m</td>
<td>50 m</td>
</tr>
<tr>
<td>Breadth</td>
<td>12 m</td>
<td>7.4 m</td>
</tr>
<tr>
<td>Depth</td>
<td>2.3 m</td>
<td>2.75 m</td>
</tr>
<tr>
<td>Draft</td>
<td>1.6 m</td>
<td>1.6 m</td>
</tr>
<tr>
<td>Speed</td>
<td>12.96 km/h</td>
<td>10 km/h</td>
</tr>
<tr>
<td>Cargo Capacity</td>
<td>1450 tons</td>
<td>360 tons</td>
</tr>
<tr>
<td>Brake Power</td>
<td>600 hp</td>
<td>500 hp</td>
</tr>
</tbody>
</table>

Table 4: Particulars of the existing means of inland transport in Egypt.
Fig. 6 shows results of the comparison between the most economical configuration of a pushed barge convoy system working through Cairo-Aswan waterway and the other inland transport means in Egypt. The results show that, pushed barge convoy system with 12PB configuration gives higher transport efficiency than self-propelled vessel and pusher barge – dump barge. Therefore, it can be concluded that pushed barge convoy system is feasible for cargo transport through Cairo-Aswan waterway.

![Inland Transportation Means in Egypt](image)

**Fig. 6** Transport efficiency for inland transport means in Egypt

### 6. Conclusions

Convoy speed, cargo carrying capacity, number of barges and convoy configurations of any pushed barge convoy system working through Cairo-Aswan waterway are affected by the shallow water nature of the navigation route. Also, size of the existing locks represents another constraint on the characteristics of such system.

The pushed convoy system with long configuration gives higher tactical diameter, advanced distance, transfer distance, response time and stopping distance than the wide configuration. Also, the pushed barge convoy system with long configuration gives less resistance than the wide configuration.

The developed computer program represents a simple and accurate tool to calculate resistance and the required power for any configuration of a pushed barge convoy system works through Cairo-Aswan waterway. After some minor modifications, this program can be used for other inland waterways.

Based on the transport efficiency as a technical and operational measure, the most economical configuration of any pushed barge convoy system working through Cairo-Aswan waterway is two pushed barge convoy in the form of 12PB.

Transport efficiency of a pushed barge convoy system is higher than the other existing river transport means in Egypt. This means that, cargo transport by pushed barge convoy system working through Cairo-Aswan waterway is economically feasible.

The results of this study had been based on empirical formula, and their consistency with real cases still need to be further tested. Not only model tests may be exploited for such validation, but also numerical simulation based on computational fluid dynamics.

### REFERENCES

S. M. Shenouda, M. M. Moustafa, H. S. El-Kilani, L. B. Kamar

The Most Economical Configuration of Pushed Barge Convoy System Through Cairo-Aswan Waterway


NOMENCLATURES

\[ A_c \] Cross-section area of the waterway, m²
\[ A_s \] Cross-section area of the convoy, m²
\[ B_c \] Width of the waterway, m
\[ B_x \] Breadth of the convoy, m
\[ Dwt \] Deadweight of the convoy , tons
\[ E_T \] Transport efficiency, tons. km/(kW. h)
\[ F_{NL} \] Froude number
\[ F_{NRh} \] Froude number defined in terms of the hydraulic radius
\[ h \] Water depth, m
\[ M \] Length – displacement ratio of the convoy
\[ P_B \] Brake power, kW
\[ P_E \] Effective power, kW
\[ R_h \] Hydraulic radius, m
\[ R_T \] Total Resistance, KN
\[ T_x \] Draft of the convoy, m
\[ V_x \] Convoy speed, km/hr
The Most Economical Configuration of Pushed Barge Convoy System Through Cairo-Aswan Waterway

$\Delta$ Displacement of the convoy, tons

$\eta_p$ Propulsive efficiency

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PROPELLER FAULT DIAGNOSIS BASED ON A RANK PARTICLE FILTER FOR AUTONOMOUS UNDERWATER VEHICLES

UDC 629.5.035:629.58
Original scientific paper

Summary

Rank particle filtering was applied to fault diagnosis technology. Control force and yaw moment losses that occurred in the corresponding degrees of freedom were estimated, and the trends of change were calculated by the multi-step-ahead prediction process in the rank particle filter. Based on the estimated results in the normal state, the situations involving failure were detected. To achieve this target, a mathematical model of the normal and faulty motion states of an underwater vehicle was first developed. Subsequently, the rank sample method combined with the particle filter was used, to obtain the importance probability density function of the autonomous underwater vehicle status. The rank particle filter obtained above realized the real-time state estimation and trend prediction of the motion state. A modified Bayesian algorithm was used to process the estimated control force and yaw moment losses in a given length of time. Based on the calculation results in the normal situations, a back propagation neural network was trained to obtain the diagnostic values. The condition of the propellers was determined based on the diagnostic values. Fault diagnosis simulation experiments were carried out using both the data obtained from the semi-physical simulation system with a hypothetical propeller failure, and the real sea trial data to verify the performance of the proposed algorithm. The results showed that the rank particle filter method could be applied to the propeller fault diagnosis of autonomous underwater vehicles, and solved the problem that arises when a single degree of freedom of the loss is utilized.

Key words: Autonomous Underwater Vehicle; Rank Particle Filter; Fault diagnosis; Propeller

1. Introduction

Deprived of a tether that would connect them to ships, the working state of autonomous underwater vehicles (AUVs) is difficult to observe in the underwater environment; therefore, AUVs must possess enough autonomy to accomplish missions [1]. For such vehicles to be capable of accomplishing missions, many advanced control systems are being widely applied to the guidance and control of these unmanned vehicles [2][3]. Without proper steering, a ship
Propeller fault diagnosis based on a rank particle filter for autonomous underwater vehicles

J. He, Y. Li

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may lose its control under harsh conditions [4]. The underwater environment is even more complicated and variable, and frequently has unpredictable effects on AUVs. Such effects include minor and severe effects, where a minor effect may lead to the failure of a task, and a severe effect may lead to the damage, and even the loss of the AUV. For example, the loss accidents of KAIKO and Nereus [5][6] was regrettable, which brought significant losses to the research development of the two study teams. As described above, improvements in the AUV’s operating ability increased the complexity of the AUV system [7][8], and the AUV’s safety and security have received increasing attention. Therefore, the fault diagnosis of propulsion, sensors, and other equipment subsystems of the AUVs is required [9].

According to different classification criteria, fault types can be divided into different types and regions. As the manipulating operation of an AUV depends on its propellers, a fault in a propeller may lead to a failure in the underwater task, or even the eventual loss of the vehicle. For example, an AUV controlled by double stern propellers may encounter a situation where the propeller can output the desired thrust at low speed, but cannot output the desired thrust that exceeds a certain value. This failure affects the yaw primarily, such that the AUV will always be subject to an uncontrollable yaw moment, resulting in the failure of path following. Therefore, it is necessary to carry out the fault detection of the propellers independently [10].

When the motion state of an AUV is represented by the state equation, we can utilize the observations to estimate the unknown state variables. Therefore, a model-based fault diagnosis method was considered: based on the states observed, we could detect the fault and estimate the fault degree [11]. Fault diagnosis based on the analytic model is the oldest and most systematic method; for example, Matko [12] applied the sigma-point unscented Kalman filter method to two types of AUVs in an open-water navigation task simulation, and Filaretov et al. [13] solved the problem of fault detection and localization using the kinematic model of the AUV, with special data fusion from its navigation sensors. Chu et al. [14] presented a propeller fault reconstruction method based on a terminal sliding observer. Sun et al. [15] proposed the fault diagnosis method based on a Gaussian particle filter, introducing the yaw moment loss parameter to estimate the fault. The idea of using the modified Bayesian algorithm to detect faults is worth learning from; however, as only a certain degree of freedom in the control force and yaw moment losses is used, the method may misdiagnose fault-free situations. In addition, because the method lacks forecasts for future trends, the fault is detected relatively late.

According to the principle of rank statistics [16], we can conclude that the sampling points obtained by the rank sampling method are reasonable and can simulate the probability distribution of the system state effectively. Further, the particle filtering method has high filtering precision, and can track system state changes with time. In the literature [17], to solve the problem of particle diversity and effectiveness lost due to particle degeneracy and resampling, the rank filter and particle filter methods were combined. First, the importance density function was obtained by the rank filtering method. As the importance density function contains the latest observation information, it is closer to the true state probability density; the particle filter method is used subsequently to estimate the system state. This allows the rank particle filter to have high filtering accuracy, while eliminating the complexity of calculations in the particle resampling method.

In view of the theories above, this study applied the rank particle filter method to the fault diagnosis technology. First, the mathematical model of the AUV was modelled by hydrodynamic parameters. The probability density function of the motion state of the AUV was obtained by the rank filtering method, and its trend was predicted at this time. Subsequently, the particle filter was combined to construct the rank particle filter, and the
control force and yaw moment losses in the AUV were estimated. Further, the modified Bayes (MB) algorithm was used to analyse the estimated losses over a time period. Finally, the back propagation (BP) neural network was used for propeller fault diagnosis. With the simulation data and the sea trail data with the propeller fault, the fault diagnosis simulation experiments were carried out to test the performance of the proposed method.

2. AUV Mathematical Model and Fault Model

2.1 AUV Platform and Hydrodynamic Model

Because the main target of the fault diagnosis is the propeller of the AUV, a certain type of AUV was selected as the research object. Propellers were the only actuators of this AUV, and no fin was used in its motion control.

To simplify the complexity of the fault diagnosis algorithm, the mathematical model of the AUV system was decomposed in the horizontal and vertical planes. Only the planar motion of the AUV was observed, and it was assumed that the fault occurred in the two stern propellers in the horizontal plane. The AUV stern propellers’ configuration in the horizontal plane is shown in Fig. 1. The efficiency of the channel propellers will reduce at high velocities, therefore the turning motion is primarily controlled by these two stern propellers. The two propellers adopt catheter propellers, in which the longitudinal axis of each propeller and that of the AUV form a 13° angle. The main parameters of the AUV are listed in Table 1.

Table 1 Main hydrodynamic parameters of the AUV

<table>
<thead>
<tr>
<th></th>
<th>2.15×10⁻³ kg</th>
<th>5.6 m</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>2 m</td>
<td>h</td>
</tr>
<tr>
<td>b</td>
<td>1 m</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>(0.09, 0, -0.022) m</td>
<td></td>
</tr>
<tr>
<td>Iₓ</td>
<td>5.42×10² kg</td>
<td></td>
</tr>
<tr>
<td>Iᵧ</td>
<td>7.62×10³ kg</td>
<td></td>
</tr>
<tr>
<td>I自媒体</td>
<td>-1.58×10⁻³</td>
<td>-8.154×10⁻²</td>
</tr>
<tr>
<td>Z自媒体</td>
<td>-7.63×10⁻²</td>
<td>-3.88×10⁻³</td>
</tr>
<tr>
<td>Y自媒体</td>
<td>-8.09×10⁻³</td>
<td>3.80×10⁻³</td>
</tr>
</tbody>
</table>

The established space coordinate system is defined as recommended by the International Ship Model Towel Pool Recall (ITTC) and the Shipbuilding and Engineering Society (SNAME) terminology bulletin system, as shown in Fig. 2.

The E–ξηζ geodetic coordinate system is fixed on the earth at any fixed point, and the origin of the body coordinate system O–xyz is fixed at the AUV’s centre of gravity.
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Furthermore, the geodetic coordinate system and the body system can reach the corresponding relations through the conversion:

\[
\begin{bmatrix}
    x \\
y \\
    z
\end{bmatrix} = \mathbf{T}^{-1} \begin{bmatrix}
    \xi \\
    \eta \\
    \zeta
\end{bmatrix}
\]

(1)

In this equation,

\[
\mathbf{T}^{-1} = \begin{bmatrix}
    \cos \psi \cos \theta & \sin \psi \cos \theta & -\sin \theta \\
    \cos \psi \sin \theta - \sin \psi \cos \phi & \sin \psi \sin \theta + \cos \psi \cos \phi & \cos \theta \sin \phi \\
    \cos \psi \sin \theta \cos \phi + \sin \psi \sin \phi & \sin \psi \sin \theta \sin \phi - \cos \psi \cos \phi & \cos \theta \cos \phi
\end{bmatrix},
\]

(2)

where \( \phi \) is the roll angle, and the AUV tilting to the right is defined as positive; \( \theta \) is the pitch angle, and the AUV tilting to the stern is defined as positive; \( \psi \) is the yaw angle, and the AUV turning to the right is defined as positive.

We assumed that the AUV operated in a deep and still environment; thus, the control force was produced by the main body hydrodynamic force and the propeller thrust, according to equation (3):

\[
F = f(V, \dot{V}, \Omega, \dot{\Omega}, n)
\]

(3)

where \( V \) and \( \dot{V} \) are the velocity and acceleration of the AUV, respectively; \( \Omega \) and \( \dot{\Omega} \) are the angular velocity and angular acceleration of the AUV, respectively; \( n \) is the propeller’s rotating speed.

Through a Taylor expansion at the reference point, the hydrodynamic coefficients in the formula above were obtained, while ignoring the second-order coefficients above. Based on the previous assumptions, the inertial hydrodynamic terms were attributed to the terms on the left side of the equation, and the non-inertial hydrodynamic terms were attributed to the terms on the right side of the equation. By setting \( \mathbf{X} = [u \ v \ w \ p \ q \ r]^T \), we obtained

\[
D\ddot{\mathbf{X}} = F_{\text{vis}} + F_{\text{propeller}}
\]

(4)

In the equation,

\[
D = \begin{bmatrix}
    m - X_\dot{\eta} & -X_v & -X_w & 0 & m_zG - X_\dot{\eta} & -X_f \\
    -Y_\dot{\eta} & m - Y_v & 0 & -m_zG - Y_\dot{\eta} & 0 & m_xG - Y_f \\
    -Z_\dot{\eta} & 0 & m - Z_w & -Z_p & -m_xG - Z_\dot{\eta} & 0 \\
    0 & -m_zG - K_v & -K_w & I_x - K_p & -K_q & -K_f \\
    m_zG - M_\dot{\eta} & 0 & -m_xG - M_w & -M_p & I_y - M_q & 0 \\
    -N_\dot{\eta} & m_xG - N_v & -N_w & -N_p & 0 & I_z - N_f
\end{bmatrix}
\]

\[
F_{\text{vis}} = \begin{bmatrix}
    X_{\text{vis}} & Y_{\text{vis}} & Z_{\text{vis}} & K_{\text{vis}} & M_{\text{vis}} & N_{\text{vis}}
\end{bmatrix}^T
\]

\[
F_{\text{vis}} \text{ is the non-inertial hydrodynamic term; } F_{\text{propeller}} \text{ is the thrust generated by the propeller.}
\]

The motion system of the AUV is highly nonlinear and can be described by the following equation of state:
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\[
\begin{align*}
\dot{x} &= Ax + Bu \\
\text{Zobser} &= Cx,
\end{align*}
\]

(5)

where \( A \) is the system matrix, \( x \) is the vector of the motion state variables, \( u \) is the control input, \( B \) is the input matrix of the system, \( \text{Zobser} \) is the measured value, and \( C \) is the observation matrix of the system.

2.2 AUV Platform Fault Model

In Section 2.1, the state equation of the AUV was obtained, which was a continuous equation. However, in the actual operation, the status of the AUV is periodically sampled. Similarly, the corresponding control command is also periodically sent to the propellers. Therefore, the discretization for the continuous equation above is needed:

\[
\begin{align*}
x_{k+1} &= G_k \cdot x_k + H_k \cdot u_k + Q_k \\
\text{Zobser}_{k+1} &= C_k \cdot x_k + R_k,
\end{align*}
\]

(6)

where \( G_k \) is the discretized system matrix, \( H_k \) is the discretized input matrix of the system, \( Q_k \) and \( R_k \) are the system process noise and measurement noise, respectively.

The failure in the propeller will weaken the output thrust; therefore, the actual axial control force and the yaw moment of the AUV will change. The actual control force and yaw moment will deviate from \( u_k \) to \( u_k + \Delta u_k \), where \( \Delta u_k \) in the normal state should be zero. To diagnose the AUV’s state, the actual control force and yaw moment, which include the fault condition are introduced into equation (6), and the motion model including the failure of the AUV can be obtained as shown in equation (7):

\[
\begin{align*}
x_{k+1} &= G_k \cdot x_k + H_k \cdot (u_k + \Delta u_k) + Q_k \\
\text{Zobser}_{k+1} &= C_k \cdot x_k + R_k
\end{align*}
\]

(7)

3. Rank Particle Filtering and Improved Algorithm

3.1 Rank Filtering Method

The rank sampling method is a deterministic sampling method. First, we determined the state vector \( X \) with a mean value \( \bar{X} \) and the covariance matrix \( \text{Cov}_X \), and then calculated the mean and mean square error of the nonlinear change \( f(X) \).

According to \( \bar{X} \) and \( \text{Cov}_X \), we obtained the initial sample points \( x_{\text{SigmaPts}} \): the \( i^{th} \) sample point was

\[
x_{\text{SigmaPts}}(i) = \bar{X} + \mu_j \cdot \lambda_{pj} (\sqrt{\text{Cov}_X})_l
\]

(8)

In this case, \( l = 1, 2, ..., R_n \) ( \( R_n \) is the dimension of \( X \)), and when \( j = 1, 2, ..., R_m \) ( \( R_m \) represents the layers of rank sampling, and the more the layers, the higher is the precision of the sampling), \( i = (j-1)R_n + 1 \). \( \lambda_{pj} \) is the lower bound of the confidence interval with probability \( p_j \) in the \( X \) distribution. \( \mu_j \) is the correction factor of the sampling point, subjected to
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\[ \sum_{j=1, j \neq R_n + 1}^{2R_n + 1} \mu_j \hat{A}_{pj} = 0 \]  

The initial sampling points were subject to non-linear changes to obtain the set of output variables \( x_{Pred} \):

\[ x_{Pred}(i) = f(x_{SigmaPts}(i)) \]  

Next, the mean and covariance were calculated:

\[ \bar{x}_{Pred} = \frac{1}{n_{Pts}} \sum_{i=1}^{n_{Pts}} x_{Pred}(i) \]  

\[ Cov_{x_{Pred}} = \frac{1}{\omega} \sum_{i=1}^{n_{Pts}} \mu_i (x_{Pred}(i) - \bar{x}_{Pred})(x_{Pred}(i) - \bar{x}_{Pred})^T \]  

Here, the weight coefficient of the covariance is \( \omega \)

\[ \omega = \sum_{j=1, j \neq R_n + 1}^{2R_n + 1} \mu_j \hat{A}_{pj}^2 \]  

3.2 Rank Particle Filter with Multi-Step-Ahead Prediction

The rank particle filter method was used subsequently to estimate the future state of the AUV. The importance density function was obtained by the rank filtering method in Section 3.1 above, and then the particle filter method was combined to estimate the future state.

We knew that \( x_{SigmaPts} \) was the initial sample points from the posterior probability distribution \( N(\bar{x}, Cov_x) \), and it was updated to generate the \( x_{Pred} \) and corresponding weights via the rank filter method above. Further, we assumed that \( x_{Pred}(k,i) \) was one of the sample points in \( x_{Pred}(k) \) at time \( k \) obtained by the particle filter sampling process.

We defined a discrete distribution that contained the sample points and weights \( \omega(k,i) \):

\[ \hat{p}(x_{Pred}(k) \mid Zobser_{\emptyset k}) = \sum_{i=1}^{N} \omega(k,i) \delta(x_{Pred}(k) - x_{Pred}(k \mid k-1,i)) \]  

where \( \delta(x) \) is the Dirac delta function,

and the p-step-ahead prediction probability density function could be obtained based on the following distribution:

\[ p(x_{Pred}(k+p) \mid Zobser_{\emptyset k}) = \int p(x_{Pred}(k) \mid Zobser_{\emptyset k}) \prod_{j=k+1}^{k+p} p(x_{Pred}(j) \mid x_{Pred}(j-1)) dx_{k:k+p-1} \]

Subsequently, \( p(x_{Pred}(k) \mid Zobser_{\emptyset k}) \) in this equation was an alternative by the particles obtained by the rank sampling method:

\[ \sum_{i=1}^{N} \omega(k,i) \int p(x_{Pred}(k+1) \mid x_{Pred}(k,i)) \prod_{j=k+2}^{k+p} p(x_{Pred}(j) \mid x_{Pred}(j-1)) dx_{k+1:k+p-1} \]
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If \( p \geq 2 \in \mathbb{N}^* \), by calculating the integrals, \( x_{\text{Pred}}(k,i) \) in \( x_{\text{Pred}}(k) \) could now be extended. The \( p \)-step prediction algorithm for the rank particle filter was obtained as follows:

\[
\begin{align*}
&\text{For } j = 1, 2, \ldots, p \\
&\text{For } i = 1, 2, \ldots, N \\
&\quad \text{sample } x_{\text{Pred}}(k + j \mid k + j - 1, i) \text{ from } p(x_{\text{Pred}}(k + j) \mid x_{\text{Pred}}(k + j - 1)) \\
&\text{end} \\
&\text{end}
\end{align*}
\]

The multi-step-ahead prediction probability density function was then obtained, and the weights \( \omega(k,i) \) were maintained as those at time \( k \):

\[
\hat{p}(x_{\text{Pred}}(k + p) \mid \text{Zobser}_{0:k}) = \sum_{i=1}^{N} \omega(k,i) \delta(x_{\text{Pred}}(k + p) - x_{\text{Pred}}(k + p \mid k + p - 1, i)) \tag{16}
\]

Finally, we could resample the prediction particles from this distribution and repeated the prediction process in the next iteration.

4. AUV Propeller Fault Diagnosis Method

4.1 Constructing the Rank Particle Filter

Using the fault diagnosis model obtained in Section 2.2, the following rank particle filter was constructed:

\[
\begin{align*}
\dot{x}_{k+1} &= G_k \cdot \dot{x}_k + H_k \cdot (u_k + \Delta u_k) + Q_k \\
\dot{Z}_{\text{obser},k+1} &= C_k \cdot \dot{x}_k + R_k 
\end{align*}
\tag{17}
\]

The goal of the fault diagnosis is to analyse the value \( \Delta u_k \) and its trend, of which the mean value should be zero in the normal state. When the propeller fails, the thrust loss and additional yaw moment lead to changes in \( \Delta u_k \). The amplitude of the changes is determined by the gap between the desired and the actual control force and yaw moment. Therefore, we considered \( \Delta u_k = \{\Delta X, \Delta Y, \Delta Z, \Delta K, \Delta M, \Delta N\} \) as the estimated term of the AUV state.

4.2 Fault Diagnosis Method

The MB algorithm was then used to analyse the time series of the AUV states. The mean value \( \bar{\Delta \hat{u}}_k \), the variance \( S_2(k) \), the relative normal mean value \( \Delta \bar{u}_k \), and the variance \( S_1(k) \) of the N-point state estimation values were obtained by the following operations, based on the \( \Delta \hat{u}_k \) estimated at N time points respectively before time \( k \):

\[
\begin{align*}
\bar{\Delta \hat{u}}_k(k) &= \frac{1}{N} \sum_{j=1}^{N} \Delta \hat{u}_k(k - j) \\
S_1(k) &= \frac{1}{N - 1} \sum_{j=1}^{N} [\Delta \hat{u}_k(k - j) - \Delta \bar{u}_k]^2
\end{align*}
\]
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\[ S_2(k) = \frac{1}{N-1} \sum_{j=1}^{N} [\Delta \hat{u}_k(k-j) - \Delta \tilde{u}_k(k)]^2 \]

\[ MB(k) = \frac{S_1(k)}{Q'} - \ln\left(\frac{S_2(k)}{Q'}\right) - 1 \] (18)

If the fault was diagnosed directly by the MB value \( MB(k) \) above using the threshold method, owing to the use of only one degree of freedom, problems would occur. For example, when considering the state-estimated value in the X-axis direction alone, thrust losses would also exist except in fault conditions, as a result of the accelerating process or the presence of ocean current interference. This might lead to the misdiagnosis of a normal condition; therefore, a BP neural network algorithm was combined, considering its pattern recognition ability.

As shown in Fig. 3, the control force and yaw moment losses, and the MB values were taken as the input, and the output was the diagnostic value indicating whether the fault existed. The thrust loss and diagnostic values in normal conditions were selected as training samples to train the neural network.

![Fig. 3 Fault diagnosis algorithm process](image)

### 4.3 Rank Particle Filter Propeller Fault Diagnosis Method

Based on the theories above, the fault diagnosis process is described below:

Step 1: Initialize the system state values, and give the initial state \( x_0 = \{ u, v, r, \Delta X, \Delta Y, \Delta N \} \) first; subsequently, obtain the particle set \( \{ xSigmaPts(i), 1/N \} (i = 1, 2, ..., N) \) from the initial distribution.

Step 2: Obtain a new set of particles \( \{ \hat{xPredSigmaPts(k, i)} \} (i = 1, 2, ..., N) \) by the rank sampling method.

Step 3: Use the rank particle filter method to obtain the posterior probability density function, which is then considered to be the proposed distribution.

Step 4: Use the multi-step-ahead prediction method, described in Section 3.2, to update the sample distribution.

Step 5: Obtain the new particle set \( \{ xSigmaPts(k, i) \} (i = 1, 2, ..., N) \) from the proposed distribution. The important weights are taken as the likelihood probability density function, and are normalized using the weighted set of particles, such that the mean and covariance of the probability distribution at \( k \) can be calculated.
Step 6: Estimate the AUV state $\Delta \hat{u}$ based on the particle set and the normalized weights, and use the MB algorithm to manage $\Delta \hat{u}$.

Step 7: Take the AUV state estimation $\Delta \hat{u}$, and the MB value obtained by the MB algorithm as the input to the BP neural network. If the diagnostic value is beyond a certain value, a fault is considered to have occurred.

5. Fault Diagnosis Simulation

5.1 Simulation Environment Configuration

To simplify the complexity of the fault diagnosis algorithm, the mathematical model of the AUV system was decomposed in the horizontal and vertical planes as described in Section 2. Assuming that one of the stern propellers in the horizontal plane was faulty, the fault only affects three degrees of freedom in the horizontal plane. Thus, we primarily discuss the fault situation simulation in the horizontal plane, and use the fault diagnosis algorithm in this paper to diagnose the fault.

During the actual sea trial with this AUV, we found that a faulty propeller might not output enough thrust to meet the desired thrust, owing to the fault in the blade. In addition, another fault state existed: when the desired thrust was small, the propeller could output the required thrust; however, when the desired thrust was greater than a certain value, the propeller could not output the desired output thrust, instead, a lower thrust was output.

Therefore, the failure condition of the AUV's left stern propeller was set as follows: when the output thrust of the propeller was less than 60 N, the propeller was considered to be rotating at a low speed, and the thrust of the propeller could be output normally; when the desired thrust of the propeller was greater than 60 N, this faulty propeller could only output 60 N thrust. In this case, the fault diagnosis result was analysed. Considering that the desired output thrust of the propeller was related to the desired velocity and yaw angle of the AUV, the simulation parameters were set based on the different desired velocities and yaw angles of the AUV.

The accelerating process, or the turning of the bow during acceleration will cause thrust loss, which may be misdiagnosed by the algorithm as a fault. In determining the threshold, we carried out a set of simulations under different desired velocities and yaw angles, where no fault occurred in the propeller model, and obtained the MB values caused by the motion changes in the AUV. In Table 2, the maximum MB values in fault-free conditions, diagnosed by the MB algorithm under different AUV motion states, are enumerated:

<table>
<thead>
<tr>
<th>AUV Status</th>
<th>X-axis</th>
<th>Y-axis</th>
<th>Z-axis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accelerate</td>
<td>26</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Accelerate (Turn)</td>
<td>13</td>
<td>14</td>
<td>155</td>
</tr>
<tr>
<td>Turn</td>
<td>9</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Cruise</td>
<td>9</td>
<td>8</td>
<td>8</td>
</tr>
</tbody>
</table>

The maximum operating velocity of the AUV in the simulation was 2.0 m/s; therefore, a series of simulations were carried out, where the desired velocities range from 0 to 2.0 m/s and different yaw angles were selected. The MB values of the AUV in different motion states characterized the AUV states during a time period. The threshold should be set to improve the sensitivity of the algorithm, and provide the basis for the comparison of different algorithms.
We considered that if the MB value of the fault diagnosis algorithm was greater than the maximum MB value at a certain degree of freedom in a fault-free state, the propeller was considered to be possibly faulty. In this case, the maximum MB values obtained with different degrees of freedom respectively in Table 2 were set as the threshold.

5.2 Simulation and Results Analysis

First, the case of an AUV cruising in a straight line was simulated: the desired velocity was set as 0.5 m/s during the first 50 s; from 50 s to 100 s, and the desired velocity was 1.0 m/s. The desired velocity of the AUV was set as 2.0 m/s starting from 100 s.

Figures 4–6 show the simulation results, describing the use of the rank particle filter method. The control command and the current motion state of the AUV were taken as the input, and the control force and yaw moment losses and their multi-step-ahead predictions in different degree of freedom were shown. The solid line is the estimated result, and the dash-dot line is the result of the forward prediction. The results show that the multi-step-ahead prediction will amplify the original control force and yaw moment losses.

![Fig. 4 X-axis force loss estimation and prediction](image)

![Fig. 5 Y-axis force loss estimation and prediction](image)

![Fig. 6 Yaw moment loss estimation and prediction](image)

Based on the MB algorithm, the control force and yaw moment losses are used to calculate the trend in a certain time period using the sliding time window, and the threshold method is used to judge whether a fault had occurred. Figures 7–10 show the results. Fig. 7 shows that the calculation result in the X-axis direction is greater than the set threshold value of 26 at 110.5 s, and the fault diagnosis algorithm considers that a fault has occurred at this time.
The control force and the yaw moment losses prediction were also analysed by the MB algorithm. The threshold method was also used to judge whether a fault has occurred. Figure 14 shows that the fault diagnosis algorithm with prediction diagnosed the potential failure earlier, at 105 s. However, between 7 s to 23 s and between 57 s to 73 s, the AUV was accelerating and the thrust loss in the X-axis direction was amplified by the multi-step-ahead prediction, resulting in a larger estimated thrust loss. Although the MB values in the Y-axis and Z-axis directions showed that the AUV was in a normal state, a fault was error detected, owing to the X-axis value that is beyond the threshold set previously. Therefore, it would not be accurate if only the loss in a single degree of freedom is considered to determine the fault.
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The comparison above shows that the control force and yaw force losses prediction can reflect the trend, and can detect an imminent fault sooner. However, only a certain degree of freedom of loss was used, and was not enough to eliminate the misdiagnosis. Considering the propeller fault configuration again: the left main propeller could output a thrust below 60 N normally, but could not provide enough thrust beyond 60 N. When the AUV was no longer at a standstill, a flow had caused resistance during acceleration. Furthermore, a delay response occurred between the control command and the execution of the actuator in the actual sea trial. This caused the estimated thrust output to deviate from the expected thrust. This deviation was detected, and also amplified in the multi-step-ahead prediction process, which resulted in an increase of the MB values. Faults were error determined to occur.

Therefore, the control force and yaw moment losses with only one degree of freedom should not be analysed. All the control force losses in the X-axis and Y-axis, the yaw moment in the Z-axis, and the corresponding MB values should be taken into account. Subsequently, the normalized values were used as the input of the BP neural network, which was trained by the values in the normal state, and the fault situation could be diagnosed.

The BP neural network is capable of pattern recognition, and we expected the diagnostic value to be approximately zero in the normal states. During the training process, the control forces and the yaw moments, the MB values, and the expected diagnostic values in the normal states were taken as samples. According to the previous simulations, a thrust loss due to acceleration was estimated. Thus, we expected that when the AUV was accelerating when cruising in a straight line or when turning the bow, the output should be 0.2 and 0.3, respectively. Some examples for training the neural network are given in Table 3.

The state vectors included the control force, yaw moment losses, and the MB values; the BP neural network was trained by those values. Under normal circumstances, the diagnostic value of the neural network is up to 0.3. If the diagnostic value was close to 0, it indicated that the AUV was in a normal state. However, if the diagnostic value was farther away from 0, it indicated that the AUV might be faulty. In this study, the system was considered to be faulty if the diagnostic value was greater than 0.3.

<table>
<thead>
<tr>
<th>AUV Status</th>
<th>Example of Samples</th>
<th>Expected Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accelerate</td>
<td>[-4.35,0.04,0.13, -87.55,0.0]</td>
<td>0.2</td>
</tr>
<tr>
<td>Accelerate (Turn)</td>
<td>[3.26,5.54,7.84,1.44,2.1,11.02]</td>
<td>0.3</td>
</tr>
<tr>
<td>Turn</td>
<td>[-75.3,36.7, -135, -79.3,35.9, -136.2]</td>
<td>0</td>
</tr>
<tr>
<td>Cruise</td>
<td>[0.68,0.0, -3.76,0.0]</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 3 BP neural network training samples
As shown in Fig. 15, when the diagnostic value deviates from 0 during the previous two accelerating processes, the output value of the fault diagnosis result is always below the threshold value of 0.3. At 108 s, the diagnostic value of the BP neural network is greater than the threshold value of 0.3 for the first time, which means that the AUV experienced an imminent fault. The comparison between Fig. 15 and Fig. 14 show that the fault diagnosis using the BP neural network can avoid the misdiagnosis when the AUV is accelerating. Compared with the fault diagnosis results obtained by the fault diagnosis algorithm without prediction in Fig. 10, we observed that the original method detected the fault in the propeller at 110.5 s, and the proposed method showed that the AUV was faulty at 108 s. The proposed method can detect the AUV failure at the soonest possible.

Next, the case of an AUV turning the bow during acceleration was simulated: from 0 s to 100 s, the AUV was released and accelerated to 0.5 m/s; from 100 s to 200 s, the AUV was turning 45° to the left; subsequently, the AUV was accelerating from 0.5 m/s to 1.0 m/s while turning its bow 90° to the left starting from 200 s. Fault diagnosis algorithms with and without prediction were applied to judge whether a fault has occurred.

Figures 16–19 show the simulation results. Fig. 17 shows that the MB value in the Z-axis was greater than the set threshold value of 155 at 212 s, and the fault diagnosis algorithm considered that a fault had occurred at this time. In addition, from Fig. 18 and Fig. 19, we could conclude that the diagnosis algorithm with prediction detected the fault sooner, and the diagnostic value of the BP neural network exceeded the threshold at 202.5 s.
We considered that the AUV was primarily controlled by only one propeller in the process of turning the bow; for instance, the AUV’s right propeller provided the primary part of the thrust, while the left propeller was less involved in the turning left process. Therefore, another simulation test was configured: from 0 s to 50 s, the AUV was accelerating to 1.0 m/s, and no fault occurred in the propeller during this process; from 50 s to 100 s, the AUV was turning 45° to the left; and the AUV started turning to the right from 100 s. In the simulation, the AUV’s left propeller was fault-free at first, but it became faulty starting from 50 s: the maximum output thrust was 60 N.

Fig. 20 to Fig. 23 show that the fault diagnosis algorithm without prediction diagnosed the existence of a fault at 103.5 s, from the MB value in the Z-axis exceeding the threshold.
whereas the algorithm with prediction diagnosed a fault at 75 s. However, owing to the prediction, from 4 s to 26 s, a misdiagnosis occurred. The BP neural network diagnosed the existence of a fault at 78.5 s instead, and a misdiagnosis did not occur.

When the AUV was turning to the left, the yaw moment was primarily provided by the right propeller, and the left one was less involved at the beginning of the process. If the left propeller failed, it was very difficult to accurately judge the problem during this process. The fault diagnosis algorithm did not have the correct estimation of the propeller thrust loss; on the contrary, when the AUV was close to the desired yaw angle, the required thrust of the left propeller increased, the impact of the fault on the AUV's motion state became more obvious, and the presence of the fault was quickly detected.

Finally, we tried to use the real sea trail data to test the performance of the fault diagnosis algorithms. Figures 24 and 25 display a part of the sea trail process with a fault situation. The AUV was configured to sail straight at the velocity of 1.0 m/s, and then turn the bow to 30° to continue sailing. From the velocity curve and the yaw angle curve, we observed that the AUV experienced a velocity loss and an unstable change in the yaw angle during the last section of the trail, which corresponded to the thrust loss in the relevant degree of freedom.

Figures 26 and 27 compare the results analysed from the sea trail data, using the Gaussian particle filter method [11] and the rank particle filter method. Sea trail data was used by these two kinds of filter methods to estimate the control force and yaw moment losses. Subsequently, the MB algorithm was used to analyse the estimated values during a certain time period. The dash-dot line is the MB value of the Gaussian particle filter and the solid line is the result of the rank particle filter.

---

**Fig. 24** AUV velocity data in sea trail

**Fig. 25** AUV yaw angle data in sea trail

**Fig. 26** X-axis force loss with MB algorithm

**Fig. 27** Yaw moment loss with MB algorithm
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Here, we observed that in the initial stage of releasing the AUV, the Gaussian particle filter method along the X-axis direction has a larger estimation of the control force loss than the rank particle filter method, and the MB value estimated during this stage is close to that at the end of the trail, which indicates that the AUV may be faulty. In contrast, the rank particle filter method estimated the current state of the AUV sooner and more accurately in the initial stage, and also estimated the control force loss quickly at the end of the trail, reducing the possibility of a misdiagnosis.

Environmental interference in the real sea trail and a delayed response in the mechanical devices have also occurred. To determine the threshold value of a fault, several groups of the normal-state data with desired velocities from 0 m/s to 1.0 m/s during the sea trial process were selected. We obtained the MB values in these various states, and the maximum values were selected to determine the threshold. The AUV had a maximum value of 539 in the X-axis, 356 in the Y-axis, and 2518 in the Z-axis. At the same time, in the training process of the BP neural network, the expected diagnostic value of the turning process with acceleration was set as 0.3 and the expected diagnostic value of the accelerating process was set as 0.2. Using this method, the trained BP neural network could be used to detect the existence of a fault; the fault diagnosis process is shown in Figure 28.

Fig. 28 Fault diagnosis simulation using the actual sea trail data

Figure 29 shows the comparison between the fault diagnosis results with and without prediction. It is known that a certain deviation exists between the actual model of an AUV in the sea trail and the hydrodynamic model in the simulation. Over the course of the experiment, the fault diagnosis algorithm without prediction diagnosed a fault at 180.5 s. Further, the algorithm with prediction detected a fault at 178.5 s, but misdiagnoses still occurred. This was also due to the amplified loss that caused the MB value to be greater than the threshold, and the fault threshold was kept as before. In Figure 30, from the diagnostic result by the BP neural network, we observed some delays between the control command and the response of the actuator when the AUV moves from the initial position at around 50 s. During this stage, a relatively greater diagnostic value compared with the rest of the normal conditions existed. From 125 s to 178 s, the average absolute value of the diagnostic results is greater than the average absolute value of the previous 50 s, because the AUV was turning its bow, and there was a larger change in the Y-axis direction. A certain degree of increase in the force loss would occur in the Y-axis, causing the diagnostic value of the BP neural network to vibrate. Finally, the diagnostic value at 179 s exceeds the threshold of 0.3, implying that the AUV's propeller was faulty.
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6. Conclusions and Future Work

1. The simulation results show that the rank particle filtering method can be applied to fault diagnosis, and the control force (moment) loss can be analysed through the MB algorithm with the sliding time window.

2. The proposed algorithm can detect the fault in advance through the multi-step-ahead prediction process, and the misdiagnosis caused by using only the loss in a single degree of freedom can be reduced using the BP neural network.

3. The noise that exists in the sea trail environment creates a mostly non-Gaussian case. The rank particle filter method is suitable for the actual state estimation of an AUV and helps to improve the accuracy of the diagnosis.

4. The next step is to increase the real-time ability of the fault diagnosis algorithm, and complete the real-time diagnosis of failure of an AUV in open water, or under conditions where currents exist.

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<td>WAVE INDUCED COUPLED MOTIONS AND STRUCTURAL LOADS BETWEEN TWO OFFSHORE FLOATING STRUCTURES IN WAVES (str.149-173)</td>
<td>Mun Sung Kim, Kwang Hyo Jung, Sung Boo Park</td>
<td>Original scientific paper</td>
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SELF-ORGANIZING COOPERATION MODEL FOR SHIPS
NAVIGATING IN RESTRICTED ONE-WAY WATERWAY

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Original scientific paper

Summary

An effective ship scheduling strategy is critical for the efficiency of waterway transportation, especially in restricted one-way waterways. By treating ship traffic as a distributed system, a novel Ship Self-Organizing Cooperation algorithm (SSOCA) is proposed to evaluate the effects of self-organizing cooperation between ships. An assumption is made that overtaking is not allowed under the given safety requirement. Observing that traffic efficiency is influenced by speed differences, two types of delay times, wait time and navigation time as increased by speed reductions, are applied to evaluate traffic efficiency. The mathematical model of delay time is inferred in different entry sequences subsequently. Taking advantage of the delay model, each ship makes a decision regarding its own sequence to find the local optimum iteratively. An Arena-based ship traffic model is constructed. The simulation results indicate that average delay time for ships is decreased in comparison with the First Come First Served (FCFS) model. That advantage exists for different combinations of traffic flow parameters. Moreover, a balance between efficiency and computation is also achieved by distributing the computational burden to each ship.

Key words: self-organizing; cooperation mechanism; restricted one-way waterway; traffic efficiency; delay time; speed difference
1. Introduction

Maritime transportation plays an important role in cargo shipping. As the numbers and sizes of ships increase [1], waterways are becoming one of the most important resources, especially in world-class ports and busy inland waterways. Some of them are becoming increasingly congestion. Congestion brings high navigational risks and also reduces overall traffic efficiency, with many delays. Waterway improvement projects are supposed to be among the most effective ways to mitigate such problems. However, huge infrastructure investments for waterways are not always available in practice. Another idea is to re-organize waterway traffic to improve efficiency by enhancing the overall utilization of the waterway resource and by reducing the overall delays of arriving ships. Ship scheduling [2] is among the most effective ways to do that. First Come First Serve (FCFS) is one of the most popular scheduling models for transportation. However, FCFS is not the most efficient because it prioritizes equality over efficiency.

In general, ship scheduling can be performed in centralized or distributed ways. A large amount of existing research has concentrated on centralized modes, including queuing models [3, 4], waterway-berth coordination [5, 6] and sequential scheduling models [7]. Among their premises, the information on all arriving ships should be available and a top-level agent assigns the arrival time for each ship. All the ships need to navigate under the requirements from the top-level agent. However, it should be noted that a ship is usually controlled by crewmembers who are responsible for route planning. Operations such as ship scheduling and collision avoidance are resolved individually and locally. Although Vessel Traffic Services (VTSs) are responsible for ship traffic from a macro-perspective [8], it is usually arguable how VTSs play an importance role in maritime supervision. In fact, VTSs are only in charge of information services and receiving reports on intentions and constraints from ships. That is to say, VTSs do not control ship traffic directly. Therefore, VTSs are not top-level agents. In addition, an assumption is often made in centralized models that all the ships are navigating under identical and stable speeds. Nevertheless, the speeds of ships usually vary by large amounts within certain ranges in many real cases. For instance, ship speed in the Pelagos Sanctuary is high and dependent on vessel type, with specific spatial distributions [9]. A similar result was also found for the Port of Rotterdam, in which a normal distribution was fit to a speed histogram with a mean value of 10.7 knots and a standard deviation of 1.2 knots [10]. Variances in speed exist not only at different locations at a given time but also at different times at a given location, which triggers changes in course or speed. In order to analyze such problems, several characteristic parameters [11] are proposed for traffic in restricted one-way waterways, and simulation results show that the interactions of speed differences are closely related to congestion degree. A sensitivity analysis [12] then shows that the interactions owing to speed variances also largely influence traffic efficiency. Overall, unstable traffic derived from those interactions reduces navigation efficiency. Because ships interact with each other in a parallelized way, it is difficult to predict the future state [13] of a ship traffic system in centralized scheduling mode.

Distributed scheduling provides an alternative that focuses on the interactions between individuals, which enables the ships to make decisions individually by negotiating with adjacent ships [14]. Distributed scheduling can be accomplished using a self-organization approach [15].
Individuals can respond to local stimulations, realizing the division of work and accomplishing complex goals [16]. The individuals are equals, manifest self-control and make decisions based on local information that conforms to the ship traffic situation. In this sense, ship traffic should be treated as a self-organizing system, and distributed scheduling based on self-organization is more suitable for ship traffic than central scheduling. Self-organization, which is a process in which some form of overall order arises from local interactions between parts of an initially disordered system, was introduced and formulated by W. Ross Ashby [17]. The main idea is to build the components of a system in such a way that they will find solutions to problems by themselves [15]. Because trends in traffic systems are far from balanced [18], self-organizing theory is suitable for studying the behaviours and characteristics of such systems. The self-organizing behaviours of drivers, which is commonly identified in vehicle traffic [19], decides nearly all the spatial-temporal behaviours of a traffic system [20], including route choice [21] and traffic light coordination [22]. From statistics of the Automatic Identification System (AIS), ships navigating in restricted waterways cluster, similar to results obtained in car-following models [23, 24]. However, there are some differences. First, vehicles often choose routes according to degree of congestion in road networks, whereas ships usually navigate over regular or recommended routes because waterway networks seldom exist. Even if a ship discovers a bottleneck emerging in the planned route, the ship’s pilot would prefer to wait near the entrance rather than find an alternative. Second, vehicles are largely affected by traffic signal lights at intersections. Ships, however, act autonomously in restricted areas according to rules. In summary, the self-organizing models used to study vehicle traffic are not suitable for ship traffic due to the latter’s particular characteristics.

A novel self-organizing cooperation model for ships navigating in restricted waterways is proposed in this paper. The remainder of the paper is organized as follows. In Section 2, the reasons and preconditions of ship deceleration are ascertained by analyzing the process of ships navigating in restricted waterways. Considering safety distance, a mathematical model of delay time is inferred in Section 3. A ship self-organizing cooperation method is offered in Section 4. Using an Arena-based ship traffic simulation model, self-organizing cooperation is verified and discussed in Section 5. Finally, the conclusions are presented in Section 6.

2. Problem statement

The restricted waterway discussed in this paper is a one-way waterway with one entrance and one exit. Overtaking is prohibited to all ships navigating in the waterway. To avoid collisions in the waterway, ships are required to maintain safe distances between each other. Safe distance is usually described as ship domain [25]. In general, a ship will need a larger safe distance if it has a greater size or speed.

Assume that two ships are arriving sequentially at a restricted waterway of length of $L_{\text{wat}}$. The first arriving ship is $ship\ i$, and the later-arriving ship is $ship\ j$. As shown in Fig. 1, the whole process of the ships is divided into 4 phrases. In phrase 1 in Fig. 1(a), $ship\ i$ arrives at the approach of the waterway at $t_{i}^{\text{arr}}$ while $ship\ j$ is traveling to the approach. If there are no ships ahead, $ship\ i$ would immediately enter the waterway at $t_{i}^{\text{in}}$. Under that circumstance, $t_{i}^{\text{in}}$ is equal to $t_{i}^{\text{arr}}$. $Ship\ j$ then arrives at the approach of the waterway at $t_{j}^{\text{arr}}$ and enters the waterway at $t_{j}^{\text{in}}$. In phrase 2 in Fig. 1(b), $ship\ i$ and $ship\ j$ are sailing in the waterway. The original distance $d$
between them exceeds safe distance $d_j^s$. Ship $i$ and ship $j$ could maintain their initial speeds $v_i$ and $v_j$, respectively. If $v_j > v_i$, the distance between them would gradually decrease. At phase 3 in Fig. 1(c), ship $j$ arrives a point $A$ from the entrance point $O$, which are separated by a distance $l_j$. Ship $j$ would then slow to $v_i$ to maintain a safe distance because $d = d_j^s$ and because overtaking is not allowed in the restricted waterway. At phase 4 in Fig. 1(d), ship $i$ and ship $j$ depart the waterway at $v_j$. It can be seen that ship $j$ is delayed owing to its speed reduction in the section of $L_{wat} - l_j$ (from $A$ to $B$ at the exit), and hence traffic efficiency is sacrificed to some degree.

![Ship navigation process](image)

According to the description in Fig. 1, the deceleration has two preconditions. The first precondition is that $v_j$ must be greater than $v_i$, as shown in Eq. (1); that is,

\[ v_i < v_j \]  

(1)

If the speeds of the two ships are same, their distance will be maintained until they depart the waterway. The distance between the two ships will gradually increase if $v_j < v_i$. Ship $j$ will catch up with ship $i$ only if $v_j > v_i$. The second precondition is that the difference in the arrival times is sufficiently small, which is expressed as Eq. (2). $\Delta T_{ij}^{arr}$ is the difference between $t_{i}^{arr}$ and $t_{j}^{arr}$. The extreme is denoted as $\Delta t_{ij}^{arr}$ such that ship $j$ will not catch up with ship $i$ until ship $i$ leaves the waterway. $\Delta t_{ij}^{arr}$ can be derived as Eq. (3).

\[ \Delta T_{ij}^{arr} = t_{j}^{arr} - t_{i}^{arr} < \Delta t_{ij}^{arr} \]  

(2)
Self-organizing cooperation model for ships navigating in restricted one-way waterway

Hongbo Wang, Jingxian Liu
Jinfen Zhang, Kezhong Liu, Xugang Yang, Qing Yu

\[ \Delta t_{ij}^{arr} = \frac{L_{\text{wat},i} - L_{\text{wat},j} - d_j^s}{v_i} \]  

(3)

If Eq. (2) is satisfied, ship \( j \) will slow down owing to the obstruction caused by ship \( i \). If Eq. (2) is not satisfied, ship \( j \) will maintain its initial speed until leaving the waterway because ship \( i \) is sufficiently distant from ship \( j \) throughout the process. Thus, the precondition that ship \( j \) decelerates due to ship \( i \) is summarized in Eq. (4).

\[
\begin{align*}
  t_i^{arr} &< t_j^{arr} \\
  v_i &< v_j \\
  \Delta t_i^{arr} &< \Delta t_j^{arr}
\end{align*}
\]  

(4)

3. Traffic efficiency for different sequences

To reflect the influence of deceleration, delay time is applied to evaluate traffic efficiency, being defined as the sum of wait time and the increased navigation time caused by deceleration, which is denoted deceleration time. Obviously, smaller delay times are more desirable. Therefore, the objective function for traffic efficiency in a restricted one-way waterway can be described as

\[
\text{Min} \quad \frac{1}{m} \sum_{i=1}^{m} T_i^{\text{delay}} = \frac{1}{m} \sum_{i=1}^{m} (T_i^{\text{wait}} + T_i^{\text{dec}}) 
\]  

(5)

In Eq. (5), \( m \) is the number of arriving ships, \( T_i^{\text{delay}} \) is the delay time for ship \( i \), \( T_i^{\text{wait}} \) is the wait time for ship \( i \), and \( T_i^{\text{dec}} \) is the deceleration time.

The assumption is still that only two ships will pass through the waterway. The distance between the two ships always exceeds \( d_j^s \) when Eq. (4) is not satisfied. In that case, neither waiting nor deceleration occurs. Thus, the overall delay time \( T_{ij}^{\text{delay}} \) is zero, as expressed in Eq. (6), and traffic efficiency reaches its optimum.

\[
T_{ij}^{\text{delay}} = T_i^{\text{delay}} + T_j^{\text{delay}} = 0 + 0 = 0
\]  

(6)

However, when Eq. (4) is satisfied, ship \( j \) will slow down owing to the obstruction caused by ship \( i \). If the sequence is changed, ship \( j \) will enter the waterway immediately, and the deceleration is eliminated. At the same time, the wait time for ship \( i \) is increased, although the deceleration time remains zero. Whether the sequence should be changed to improve traffic efficiency remains uncertain, and to answer that question, the mathematical model is analyzed in the following subsections.

3.1 Traffic efficiency for the original sequence

If ship \( i \) arrived and entered the waterway earlier, it passed through the waterway without restrictions throughout the entire process. That is to say, the wait time \( (T_i^{\text{wait}}) \) and deceleration
time \((T_{i}^{\text{dec}})\) are both zero, as shown in Eq. (7) and Eq. (8). Hence, \(T_{i}^{\text{delay}}\) is described in Eq. (9).

\[
T_{i}^{\text{waiting}} = 0 \quad (7)
\]

\[
T_{i}^{\text{dec}} = 0 \quad (8)
\]

\[
T_{i}^{\text{delay}} = T_{i}^{\text{wait}} + T_{i}^{\text{dec}} = 0 \quad (9)
\]

Ship \(j\) enters the waterway after ship \(i\) and may be waiting at the entrance due to the safe distance constraint. When ship \(j\) arrives at the entrance, ship \(i\) has travelled some distance in the waterway. If the distance exceeds \(d_{j}^{s}\), ship \(j\) can enter the waterway directly like ship \(i\), and wait time \(T_{j}^{\text{wait}}\) is equal to zero, as shown in Eq. (10).

\[
T_{j}^{\text{wait}} = 0 \quad (10)
\]

Otherwise, ship \(j\) has to wait until the distance between ship \(j\) and ship \(i\) reaches \(d_{j}^{s}\). In that case, \(T_{j}^{\text{wait}}\) can be expressed as Eq. (11).

\[
T_{j}^{\text{wait}} = d_{j}^{s} / v_{i} - (t_{j}^{\text{arr}} - t_{i}^{\text{arr}}) \quad (11)
\]

It can be determined that the result of Eq. (11) is negative when ship \(j\) does not wait at the entrance. \(T_{j}^{\text{wait}}\) as shown in Eq. (12), which is the wait time caused by the constraint of safe distance:

\[
T_{j}^{0} = \text{Max}\left(0, d_{j}^{s} / v_{i} - (t_{j}^{\text{arr}} - t_{i}^{\text{arr}})\right) \quad (12)
\]

As shown in Fig. 1(c), ship \(j\) navigates with speed \(v_{j}\) from \(O\) to \(A\) and reduces speed to \(v_{i}\) from \(A\) to \(B\). Therefore, \(T_{j}^{\text{dec}}\) as expressed as Eq. (13), which is the deceleration time of ship \(j\) and is related to the speed change and deceleration distance (i.e. \(L_{\text{wat}} - l_{ij}\)).

\[
T_{j}^{\text{dec}} = \frac{L_{\text{wat}} - l_{ij}}{v_{i}} - \frac{L_{\text{wat}} - l_{ij}}{v_{j}} \quad (13)
\]

Here, \(l_{ij}\) is calculated based on Newton's first law, which is closely associated with arrival time and speed difference, as shown in Eq. (14).

\[
l_{ij} = \frac{v_{i} \times (t_{j}^{\text{arr}} - t_{i}^{\text{arr}} + T_{j}^{0}) - d_{j}^{s} \times v_{j}}{v_{j} - v_{i}} \quad (14)
\]

Therefore, the sum of wait time and deceleration time for ship \(j\) is \(T_{j}^{\text{delay}}\) in Eq. (15).

\[
T_{j}^{\text{delay}} = T_{j}^{\text{wait}} + T_{j}^{\text{dec}} = T_{j}^{0} + \frac{L_{\text{wat}} - l_{ij}}{v_{i}} - \frac{L_{\text{wat}} - l_{ij}}{v_{j}} \quad (15)
\]
Furthermore, the overall delay time for the two ships, \( T_{ij}^{\text{delay}} \), is expressed in Eq. (16) when the FCFS model is followed and Eq. (4) is satisfied.

\[
T_{ij}^{\text{delay}} = T_i^{\text{delay}} + T_j^{\text{delay}} = T_i^0 + \frac{L_{\text{wat}} - l_{ij}}{v_i} - \frac{L_{\text{wat}} - l_{ij}}{v_j}
\]  

(16)

3.2 Traffic efficiency for the changed sequence

The sequence of ship \( i \) and ship \( j \) is changed in this section, presuming that Eq. (4) is still satisfied. That is to say, ship \( i \) enters behind ship \( j \), although ship \( i \) arrives the approach earlier. In general, anchorages lie near approaches. When ship \( i \) arrives, the ship would wait for ship \( j \) at the anchorage in Fig. 2(a) and then follow ship \( j \) as shown in Fig. 2(b).

![Fig. 2](image)

\( \text{ship } i \) and \( \text{ship } j \) enter the waterway in a different sequence. (a) \( \text{ship } j \) is entering the waterway, and \( \text{ship } i \) is waiting at the anchorage, (b) \( \text{ship } i \) enters the waterway behind \( \text{ship } j \).

As shown in Fig. 2(a), ship \( i \) does not enter the waterway until ship \( j \) arrives and steers for \( d_i^s \) under the constraint of safe distance. As a result, the wait time of ship \( i \) is the sum of the difference of arrival time and the period of ship \( j \) navigating for \( d_i^s \) as shown in Eq. (17). After entering the waterway, ship \( i \) does not slow down, as shown in Eq. (18), because of the increasing distance from ship \( j \).

\[
T_i^{\text{wait}} = t_j^{\text{arr}} - t_i^{\text{arr}} + d_i^s / v_j
\]  

(17)

\[
T_i^{\text{dec}} = 0
\]  

(18)

Therefore, the delay time of ship \( i \) is expressed as Eq. (19). In the meantime, ship \( j \) does not wait and has the deceleration time shown in Eq. (20).

\[
T_i^{\text{delay}} = T_i^{\text{wait}} + T_i^{\text{dec}} = t_j^{\text{arr}} - t_i^{\text{arr}} + d_i^s / v_j
\]  

(19)

\[
T_j^{\text{delay}} = T_j^{\text{wait}} + T_j^{\text{dec}} = 0
\]  

(20)

Furthermore, \( T_{ij}^{\text{delay}} \), which is the overall delay time when ship \( j \) enters the waterway first and ship \( i \) enters the waterway afterwards, is expressed in Eq. (21).
4. Ship self-organizing cooperation model

With regard to the transportation problem, it is usually time and space consuming [26] to calculate the solution using a deterministic algorithm. An algorithm that calculates all of the permutations is possible for traffic efficiency when many ships arrive in succession. However, the computations will grow exponentially according to the permutation combination formula, especially when it involves too many ships. Heuristic algorithms are usually used to solve such problems, as in Zhang et al. [5]. Nevertheless, it is usually challenging to obtain a mathematical expression when many ships are navigating with different speeds. In this section, a self-organizing cooperation model is proposed to solve the problem. The fundamental idea of this model is that a slower ship yields priority to another faster ship to improve traffic efficiency employing a cooperation mechanism [27]. Cooperation usually refers to the process of groups of organisms working or acting together for joint or mutual benefit [28]. In this paper, it mainly denotes that ships follow simple local rules to closely approach optimal global coordination [29] in a distributed mode.

4.1 Ship self-organizing cooperation algorithm

The assumption is that ship i, ship j and ship k will arrive at the entrance in rapid succession and navigate through the waterway as shown in Fig. 2. To reduce the overall delay time, the sequence between ship i and ship j should be changed when Eq. (22) is matched besides Eq. (4). Otherwise, the original sequence is retained for the two ships.

\[ T_{ji}^{\text{delay}} < T_{ij}^{\text{delay}} \]  

(22)

If ship i and ship j follow the initial sequence of entering the waterway when Eq. (22) is not satisfied, the same method employed in section 3 is applied to calculate the delay between ship j and ship k. In the case that ship j would enter the waterway before ship i after calculation, ship k is faced with whether rearranging ship i and ship k is reasonable. In that situation, it is determined that ship i will wait for ship j in anchorage so that \( t_i^{\text{in}} = t_j^{\text{arr}} \) is no longer satisfied, and \( t_i^{\text{in}} \) should theoretically be reassigned as expressed in Eq. (23). Eq. (23) indicates that ship i does not enter the waterway until ship j navigates a distance \( d_i^s \) in the waterway, as shown in Fig. 2(a).

\[ t_i^{\text{in}} = t_j^{\text{arr}} + (t_j^{\text{arr}} - t_i^{\text{arr}} + d_i^s / v_j) = t_j^{\text{arr}} + d_i^s / v_j \]  

(23)

The deceleration criteria between ship i and ship k is shown in Eq. (24) for \( t_i^{\text{in}} < t_k^{\text{arr}} \) and omitting the influence of ship j.
Once Eq. (24) is satisfied, the criteria for a sequence change to improve traffic efficiency is presented in Eq. (25), which is similar to Eq. (22).

\[
\begin{align*}
T_{ki}^{\text{delay}} &< T_{ik}^{\text{delay}} \\
T_{ik}^{\text{delay}} &= T_k^0 + \frac{L_{\text{wat}} - l_{ik}}{v_i} - \frac{L_{\text{wat}} - l_{ik}}{v_k} \\
T_{ki}^{\text{delay}} &= t_k^{\text{arr}} - t_i^{\text{in}} + \frac{d_k^s}{v_k} \times v_k \\
l_{ik} &= v_i \times \left( t_k^{\text{arr}} - t_i^{\text{in}} + T_k^0 \right) - \frac{d_k^s}{v_k} \times v_k \\
T_k^0 &= \text{Max}\left( 0, \frac{d_k^s}{v_i} - \left( t_k^{\text{arr}} - t_i^{\text{in}} \right) \right)
\end{align*}
\]

It is also possible that \( t_i^{\text{in}} \geq t_k^{\text{arr}} \). Supposing the safe distances for ship \( i \) and ship \( k \) are same for simplicity, ship \( i \) would wait for at least \( t_i^{\text{in}} \) to satisfy the safe distance requirement, whereas the earliest time that ship \( k \) could enter the waterway is also \( t_i^{\text{in}} \). That is to say, the two ships will wait at the anchorage, and the earliest times at which they could enter the waterway are both \( t_i^{\text{in}} \). In that case, the faster ship should enter first to increase overall efficiency, as shown in Eq. (26).

\[
\begin{align*}
\text{ship } k \text{ should enter ahead of ship } i, \text{ when } t_k^{\text{arr}} \leq t_i^{\text{in}} \text{ and } v_i < v_k \\
\text{ship } i \text{ should enter ahead of ship } k, \text{ when } t_k^{\text{arr}} \leq t_i^{\text{in}} \text{ and } v_k \leq v_i
\end{align*}
\]

The problem here is to obtain details on arrivals of other ships. As a matter of fact, ship reporting systems [30] have been established in many restricted waterways and can be used to obtain position and dynamic information from passing ships. When a ship arrives at a report line, it should report its intention, arrival time and other information to the VTS centre for information sharing. Using an information inquiry and display system, all the ships could easily obtain other ships’ information, including the arrival list.

According to local optimization, a ship only considers those ships that affect its navigation and puts others aside [31]. Under the assumption that every ship arrives at the approach at the time coinciding with its report, ships whose arrival times are earlier are the most likely obstacles in the waterway. Therefore, each ship only needs to find the target ship that will arrive at the waterway earlier for self-organization. Using the arrival list offered by VTS, every ship could seek out the target ship, even if the target ships are not detected by common instruments such as radar or AIS. Own ship could then ask for the target ship to wait at the
anchorage temporarily so that own ship is not hampered in the waterway. Own ship should not make a request to give way for local interests if Eq. (22) or Eq. (25) is not satisfied.

The top level of the ship self-organizing cooperation algorithm (SSOCA) is presented in Tab. 1. According to Tab. 1, every ship reports the details of its navigational information to the VTS. The VTS then stores the information to ArrivingList. From ArrivingList, a ship can acquire the information of the last arriving ships before it. Once a ship has determined an entry sequence after calculating the delay times under different sequences, the decided sequence is reported and stored in SequenceList. The next ship behind in ArrivingList then undertakes further calculations. The procedure will proceed until all the ships are cleared from ArrivingList, which means that all the ships have determined their own sequences, and the self-organization is realized.

Tab. 1  Pseudo code of SSOCA

<table>
<thead>
<tr>
<th>Line</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Input:</strong> ArrivingList</td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>SequenceList ← the first two ships from ArrivingList;</td>
</tr>
<tr>
<td>2.</td>
<td>Remove the first two ships from ArrivingList;</td>
</tr>
<tr>
<td>3.</td>
<td>while ArrivingList is not empty do</td>
</tr>
<tr>
<td>4.</td>
<td>ShipTemp ← Collect the last two ships (ship i and ship j, ( t^{arr}_i &lt; t^{arr}_j )) in SequenceList and the first ship (ship k) in ArrivingList;</td>
</tr>
<tr>
<td>5.</td>
<td>if the entry sequence between ship i and ship j has not been changed in SequenceList then</td>
</tr>
<tr>
<td>6.</td>
<td>( T_{jk}^{delay} ) ← Calculate the delay time of ship j and ship k when ship j enters waterway before ship k;</td>
</tr>
<tr>
<td>7.</td>
<td>( T_{kj}^{delay} ) ← Calculate the delay time of ship k and ship j when ship k enters waterway before ship j;</td>
</tr>
<tr>
<td>8.</td>
<td>if ( T_{jk}^{delay} \leq T_{kj}^{delay} ) then</td>
</tr>
<tr>
<td>9.</td>
<td>ShipTemp ← ship i, j and k in sequence;</td>
</tr>
<tr>
<td>10.</td>
<td>else</td>
</tr>
<tr>
<td>11.</td>
<td>ShipTemp ← ship i, k and j in sequence;</td>
</tr>
<tr>
<td>12.</td>
<td>endif</td>
</tr>
<tr>
<td>13.</td>
<td>else</td>
</tr>
<tr>
<td>14.</td>
<td>( T_{ik}^{delay} ) ← Calculate the delay time of ship i and ship k when ship i enters waterway before ship k;</td>
</tr>
<tr>
<td>15.</td>
<td>( T_{ki}^{delay} ) ← Calculate the delay time of ship k and ship i when ship k enters waterway before ship i;</td>
</tr>
<tr>
<td>16.</td>
<td>if ( T_{ik}^{delay} \leq T_{ki}^{delay} ) then</td>
</tr>
<tr>
<td>17.</td>
<td>ShipTemp ← ship j, i and k in sequence;</td>
</tr>
<tr>
<td>18.</td>
<td>else</td>
</tr>
<tr>
<td>19.</td>
<td>ShipTemp ← ship j, k and i in sequence;</td>
</tr>
<tr>
<td>20.</td>
<td>endif</td>
</tr>
<tr>
<td>21.</td>
<td>endif</td>
</tr>
<tr>
<td>22.</td>
<td>SequenceList ← ShipTemp;</td>
</tr>
<tr>
<td>23.</td>
<td>Remove ship k from ArrivingList;</td>
</tr>
<tr>
<td>24.</td>
<td>Clear ( T_{jk}^{delay} ), ( T_{kj}^{delay} ), ( T_{ik}^{delay} ), ( T_{ki}^{delay} ), ShipTemp;</td>
</tr>
<tr>
<td>25.</td>
<td>endwhile</td>
</tr>
<tr>
<td>26.</td>
<td>return SequenceList</td>
</tr>
</tbody>
</table>
The delay times of ships in different sequences are calculated according to the ships’ arrival times and speeds, which can be obtained from ArrivingList. Tab. 2 presents pseudo code for ship self-organizing cooperation. Unlike previous algorithms such as in [5] and [7], the main purpose of the algorithm is to decide the entry sequence in SequenceList rather than the entry time. In consideration of computation efficiency, every ship can obtain the sequence in SequenceList as long as it undertakes computation. Meanwhile, in order to be fair to all, the ships arriving first will be served first as much as possible. It also should be noted that it is assumed that all the ships are willing to make a sacrifice to promote local traffic efficiency. That is to say, there is a trade-off between equity and efficiency in the scheduling model. Furthermore, deceleration is deemed to accomplished instantly, and the safe distances for all the ships are the same for simplification.

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Fig. 3  ship i, ship j and ship k in different sequences of entering the waterway. (a) ship i is entering the waterway, ship j and k are approaching the entrance, (b) ship i is navigating in the waterway, ship k is entering the waterway, and ship j is waiting at the anchorage, (c) ship i is navigating in the waterway, ship j is entering the waterway, and ship k is approaching the entrance, (d) ship j is entering the waterway, ship i is waiting at the anchorage, and ship k is approaching the entrance, (e) ship j and ship i are navigating in the waterway, and ship k is approaching the entrance, (f) ship j is entering the waterway, and ship i and ship k are waiting at the anchorage.
The ship self-organizing cooperation in each round is described as follows. First, own ship looks for the two front ships in ArrivingList and then judges whether the two ships change the sequence in SequenceList. If the sequence is unchanged, own ship (for example, ship \(k\) in Fig. 3(a)) makes a further judgement of whether own ship will be influenced by the target ship ahead (for example, ship \(j\) in Fig. 3(a)) according to Eq. (4), omitting the other hampers. If so, Eq. (22) is the second criterion for changing the sequence. In other words, the better strategy is that the target ship awaits own ship when Eq. (4) and Eq. (22) are both satisfied, as displayed in Fig. 3(b). Otherwise, when either of Eq. (4) or Eq. (22) is not satisfied, it is more effective that own ship enters the waterway afterwards, as indicated in Fig. 3(c).

Comparing the arrival times of two ships ahead in ArrivingList, own ship may find that they have already changed the sequence in SequenceList. Eq. (25) and Eq. (26) are applied in that circumstance. When \(t_{j}^{m} < t_{i}^{arr}\), own ship (for example, ship \(k\) in Fig. 3(d)) confirms whether it will switch the order with the waiting target ship (for example, ship \(i\) in Fig. 3(d)) according to Eq. (24) and Eq. (25), leaving the others (for example, ship \(j\) in Fig. 3(d) and Fig. 3(e)) out. Otherwise, the more efficient sequence could be decided in accordance with Eq. (26), as indicated in Fig. 3(f).

When all the ships are cleared from ArrivingList and confirm the sequence in SequenceList, the final entry sequence of all the ships can be obtained.

<table>
<thead>
<tr>
<th>Function Round Self-organization();</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input: (L_{wat}); % length of waterway</td>
</tr>
<tr>
<td>Input: (d^{k}); % safe distance of ship</td>
</tr>
<tr>
<td>Input: ShipTemp; % ship (i) and (j) ((t_{i}^{arr} &lt; t_{j}^{arr})) are the last two ships in succession in SequenceList and ship (k) is the first ship in ArrivingList;</td>
</tr>
<tr>
<td>Input: (t_{i}^{arr}); % arrival time of ship (i)</td>
</tr>
<tr>
<td>Input: (v_{i}); % speed of ship (i)</td>
</tr>
<tr>
<td>Input: (t_{j}^{arr}); % arrival time of ship (j)</td>
</tr>
<tr>
<td>Input: (v_{j}); % speed of ship (j)</td>
</tr>
<tr>
<td>Input: (t_{k}^{arr}); % arrival time of ship (k)</td>
</tr>
<tr>
<td>Input: (v_{k}); % speed of ship (k)</td>
</tr>
<tr>
<td>1. for each ship (k) in ShipTemp do</td>
</tr>
<tr>
<td>2. if the sequence between ship (i) and (j) in SequenceList is ship (i) and ship (j) in sequence then % the sequence is not updated</td>
</tr>
<tr>
<td>3. if (v_{j} &lt; v_{k}) and (t_{k}^{arr} - t_{j}^{arr} &lt; L_{wat} / v_{j} - (L_{wat} - d^{v}) / v_{k}) then</td>
</tr>
<tr>
<td>4. (T_{k}^{i} ← \text{Max}(0, d^{v} / v_{j} - (t_{k}^{arr} - t_{j}^{arr})));</td>
</tr>
<tr>
<td>5. (l_{jk} ← (v_{j} × (t_{k}^{arr} - t_{j}^{arr} + T_{k}^{i}) - d^{v}) × v_{k} / (v_{k} - v_{j}));</td>
</tr>
<tr>
<td>6. (T_{jk}^{delay} ← T_{k}^{i} + (L_{wat} - l_{jk}) / v_{j} - (L_{wat} - l_{jk}) / v_{k});</td>
</tr>
<tr>
<td>7. (T_{kj}^{delay} ← t_{k}^{arr} - t_{j}^{arr} + d^{v} / v_{k});</td>
</tr>
<tr>
<td>8. if (T_{k}^{delay} &lt; T_{j}^{delay}) then</td>
</tr>
<tr>
<td>9. ShipTemp ← ship (i, k) and (j) in sequence;</td>
</tr>
<tr>
<td>10. else</td>
</tr>
<tr>
<td>11. ShipTemp ← ship (i, j) and (k) in sequence;</td>
</tr>
</tbody>
</table>
12. endif
13. else
14. ShipTemp ← ship i, j and k in sequence;
15. endif
16. else % sequence has been changed
17. \( t_{i}^{in} \leftarrow t_{i}^{arr} + d_{i} / v_{j} \);
18. if \( t_{i}^{in} < t_{k}^{arr} \) then
19. if \( v_{i} < v_{k} \) and \( t_{k}^{arr} - t_{i}^{in} < L_{wat} / v_{i} - (L_{wat} - d_{i}) / v_{k} \) then
20. \( T_{k}^{0} \leftarrow \text{Max}(0, d_{i} / v_{i} - (t_{k}^{arr} - t_{i}^{in})) \);
21. \( l_{ik} \leftarrow (v_{i} \times (t_{k}^{arr} - t_{i}^{in} + T_{k}^{0}) - d_{i}) \times v_{k} / (v_{k} - v_{i}) \);
22. \( T_{k_{i}}^{delay} \leftarrow T_{k_{i}}^{0} + (L_{wat} - l_{ik}) / v_{i} - (L_{wat} - l_{ik}) / v_{k} \);
23. \( T_{k_{i}}^{delay} \leftarrow t_{k}^{arr} - t_{i}^{in} + d_{i} / v_{k} \);
24. if \( T_{k_{i}}^{delay} < T_{k_{i}}^{delay} \) then
25. ShipTemp ← ship j, k and i in sequence;
26. else
27. ShipTemp ← ship j, i and k in sequence;
28. endif
29. else 
30. ShipTemp ← ship j, i and k in sequence;
31. endif
32. else % ship i and k are both waiting at the anchorage;
33. if \( v_{i} < v_{k} \) then
34. ShipTemp ← ship j, k and i in sequence;
35. else
36. ShipTemp ← ship j, i and k in sequence;
37. endif
38. endif
39. endif
40. endfor
41. return ShipTemp

4.2 Simple SSOCA and secondary SSOCA

A simple SSOCA and secondary SSOCA are offered in addition to SSOCA. In view of the weaker computational capabilities of individual ships, the simple SSOCA model is proposed for easy calculation in Tab. 3. This model only refers to the arrival times of ships as expressed in Eq. (4) and ignores the calculation when the ship ahead could enter the waterway earliest, even if the ship ahead has changed the sequence, such as \( t_{i}^{in} \) in Eq. (23). Hence, the computational process is simplified to a large degree.
Tab. 3 Pseudo code for simple SSOCA

<table>
<thead>
<tr>
<th>Input: ArrivingList</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. SequenceList ← the first ships in ArrivingList;</td>
</tr>
<tr>
<td>2. Remove the first ships from ArrivingList;</td>
</tr>
<tr>
<td>3. while ArrivingList is not empty do</td>
</tr>
<tr>
<td>4. ShipTemp ← Collect the last ships (ship i) in SequenceList and the first ship (j) in ArrivingList;</td>
</tr>
<tr>
<td>5. if  ( v_i &lt; v_j ) and ( t^{\text{arr}}<em>i - t^{\text{arr}}<em>j &lt; L</em>{\text{wat}} / v_i - (L</em>{\text{wat}} - d) / v_j ) then</td>
</tr>
<tr>
<td>6. ( T^{\text{delay}}_{ij} \leftarrow \text{Max}(0, d / v_i - (t^{\text{arr}}_i - t^{\text{arr}}_j)) );</td>
</tr>
<tr>
<td>7. ( L_{ij} \leftarrow (v_i \times (t^{\text{arr}}_i - t^{\text{arr}}<em>j + T^{\text{delay}}</em>{ij}) - d) \times v_j / (v_j - v_i) );</td>
</tr>
<tr>
<td>8. ( T^{\text{delay}}_{ij} \leftarrow t^{\text{arr}}_i - t^{\text{arr}}_j + d / v_i );</td>
</tr>
<tr>
<td>9. ( T^{\text{delay}}_{ji} \leftarrow t^{\text{arr}}_j - t^{\text{arr}}_i + d / v_j );</td>
</tr>
<tr>
<td>10. if ( T^{\text{delay}}<em>{ij} &lt; T^{\text{delay}}</em>{ji} ) then</td>
</tr>
<tr>
<td>11. ShipTemp ← ship j and i in sequence;</td>
</tr>
<tr>
<td>12. else</td>
</tr>
<tr>
<td>13. ShipTemp ← ship i and j in sequence;</td>
</tr>
<tr>
<td>14. endif</td>
</tr>
<tr>
<td>15. else</td>
</tr>
<tr>
<td>16. ShipTemp ← ship i and j in sequence;</td>
</tr>
<tr>
<td>17. endif</td>
</tr>
<tr>
<td>18. SequenceList ← ShipTemp;</td>
</tr>
<tr>
<td>19. Remove ship j from ArrivingList;</td>
</tr>
<tr>
<td>20. Clear ( T^{\text{delay}}<em>{ij} ), ( T^{\text{delay}}</em>{ji} ), ShipTemp;</td>
</tr>
<tr>
<td>21. endwhile</td>
</tr>
<tr>
<td>22. return SequenceList</td>
</tr>
</tbody>
</table>

In addition, it is noted that the sequence of a ship is just moved one step forward in the SSOCA model. Nevertheless, it is possible that the ship ahead still matches the requirements of both Eq. (4) and Eq. (22) after own ship has moved forward once in SequenceList. The further experiment is very necessary to inspect if own ship could continue to move forward for better results. Hence, an algorithm herein named the secondary SSOCA model, the pseudocode for which is shown in Tab. 4, is introduced to test that idea. The algorithm is performed with SSOCA twice, taking the SequenceList as the new ArrivingList in the secondary SSOCA model.

Tab. 4 Pseudo code for secondary SSOCA

<table>
<thead>
<tr>
<th>Input: ArrivingList</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Obtain SequenceList according to SSOCA;</td>
</tr>
<tr>
<td>2. ArrivingList ← SequenceList;</td>
</tr>
<tr>
<td>3. Obtain the second SequenceList according to SSOCA;</td>
</tr>
<tr>
<td>4. return the second SequenceList</td>
</tr>
</tbody>
</table>

5. Experiments and analysis

5.1 Simulation model

The experiments performed for a restricted one-way waterway are presented in this section. Monte Carlo simulations, which are widely accepted in ship traffic studies, are used to
make an approximate evaluation by the law of large numbers, [32]. Arena software (version 14.0) [33] was used to construct a ship traffic simulation model that includes four sub-models: arriving, self-organizing cooperation, navigation and departure. A top-level view of the model is shown in Fig. 4(a). Taking the process as an example, the arriving ships follow some kind of defined probability distribution. A ship’s attributes (including arrival time and speed) are assigned randomly when it is created by the ship arrival module. In the ship self-organizing cooperation module in Fig. 4(b), each ship then adjusts the entry sequence using SSOCA. After determining the entry sequence, the ship enters the waterway and navigates under the constraints of safe distance. Finally, the ship leaves the waterway through the ship departure module.

Fig. 4  An Arena-based ship traffic simulation model. (a) Top-level view of the model, (b) Modules used for self-organizing cooperation.
In Fig. 4(b), the ship has priority to enter the channel in two cases. In case 1, when the arrival time of the ship is later than the entry time of the previous ship, three requirements must be satisfied: (1) its speed is larger than the previous ship, (2) the time interval between its arrival time and the entry time of previous ship is sufficiently small, and (3) the delay time can be shortened. In case 2, when the arrival time of own ship is earlier than the entry time of the previous ship, the ship has higher priority only when the first requirement in case 1 is satisfied.

5.2 Parameter settings

In accordance with Eqs. (16), (21) and (25), the delay time of a ship is related to the time interval between ships’ arrival times, speeds, channel length and safe distances. These simulation parameters and the ranges of the fixed steps over which they can updated (in the last column) are summarized in Tab. 5. According to a statistical analysis of traffic flow data for the Xiashimen Waterway in China, the ship arrival rule was subject to a Poisson distribution with a rate of arrival of 3 ships per hour, and the ship speeds obeyed a normal distribution. The base case values of mean speed and standard deviation of speed were designed as 10 knots and 2 knots, respectively. The speeds were confined in the range between 4 and 40 knots. 99.73% of speed values fell within [4, 16] in accordance with three-sigma rule of thumb [34], so that ignoring values outside that range would pose very little influence on the normal probability distribution. Waterway lengths from 8 to 12 nautical miles and safe distances from 1000 to 1400 meters were used in the model. Meanwhile, the safe distances for all ships were the same in a simulation. The above presumptions can be adjusted to real cases via statistical analyses of historical data on such parameters. Five inputs are therefore introduced: rate of arrival, mean speed, standard variation of speed, waterway length and safe distance.

<table>
<thead>
<tr>
<th>Factor</th>
<th>Base case value</th>
<th>Fixed step</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rate of arrival (h⁻¹)</td>
<td>3</td>
<td>0.5</td>
<td>2.0-4.0</td>
</tr>
<tr>
<td>Mean speed (kn)</td>
<td>10</td>
<td>0.5</td>
<td>9-11</td>
</tr>
<tr>
<td>Standard deviation of speed (kn)</td>
<td>2</td>
<td>0.1</td>
<td>1.8-2.2</td>
</tr>
<tr>
<td>Waterway length (nm)</td>
<td>10</td>
<td>1</td>
<td>8-12</td>
</tr>
<tr>
<td>Safety distance (m)</td>
<td>1200</td>
<td>100</td>
<td>1000-1400</td>
</tr>
</tbody>
</table>

Considering that waiting and ship speed reductions would influence traffic efficiency, the average of delay time is used to undertake a quantitative analysis of those influences. The average delay time is calculated as

\[
\frac{1}{m} \sum_{i=1}^{m} T_{i}^{\text{delay}} = \frac{1}{m} \sum_{i=1}^{m} (T_{i}^{\text{wait}} + T_{i}^{\text{dec}}) = \frac{1}{m} \sum_{i=1}^{m} \left[ (t_{i}^{\text{out}} - t_{i}^{\text{arr}}) - \frac{L_{\text{wat}}}{v_{i}} \right]
\]  

where \( t_{i}^{\text{out}} \) is the departure time of ship \( i \), which is obtained through simulation. On the one hand, \( (t_{i}^{\text{out}} - t_{i}^{\text{arr}}) \) displays the actual time consumed for ship \( i \) owing to waiting or deceleration. On the other hand, \( L_{\text{wat}} / v_{i} \) is the shortest time consumed for ship \( i \) without waiting and deceleration. The difference between the two comprises the delay for ship \( i \).

Furthermore, the number of ships in all simulations was set to 2000, and each simulation was repeated 200 times. To ascertain the warm-up period, a scatter plot was used to observe
when the system reached a stable state. The horizontal axis of Fig. 5 represents simulation running time, and the vertical axis represents average delay time. As can be seen, 72 hours (approximately $260 \times 10^3$ seconds) of warm-up was sufficient.

![Fig. 5 Scatterplot from the ship traffic simulation model for the base case](image)

5.3 Results and analysis

To undertake a comprehensive analysis, the following five simulation scenarios were used.

1. Rate of arrival is a variable varied by fixed steps over its range, while the other parameters remain unchanged.
2. Mean speed is a variable varied by fixed steps over its range, while the other parameters remain unchanged.
3. Standard deviation of speed is a variable varied by fixed steps over its range, while the other parameters remain unchanged.
4. Waterway length is a variable varied by fixed steps over its range, while the other parameters remain unchanged.
5. Safety distance is a variable varied by fixed steps over its range, while the other parameters remain unchanged.

By removing the data from the warm-up period, Arena generates mean values of the outputs at a 95% confidence level. The average delay times with rate of arrival, mean speed, standard deviation of speed, waterway length and safe distance under the different scenarios are presented in Fig. 6. The performance is analyzed by comparing the ship traffic simulation results determined using the SSOCA model with those of the other models in Fig. 6. In the first place, it can be seen that the factors had significant positive correlations with average delay time, with the exception of mean speed. The average delay time tended to increase with rate of arrival, standard deviation of speed, waterway length and safe distance but experienced a decreasing trend with mean speed. The tendency proves that traffic efficiency would decrease for more frequently arriving ships, smaller mean speeds, greater speed differences, greater waterway lengths and larger safe distances.
Fig. 6  Average delay times for different (a) rates of arrival, (b) mean speed, (c) standard deviations of speed, (d) waterway lengths and (e) safe distances.
It can be seen that the four lines are almost parallel and that the delay time for the FCFS model was always the highest, which shows that delay time was reduced with the variation of each factor in some of the methods. For the base case, the average delay time was reduced from 550 seconds for FCFS to 418 seconds for SSOCA, i.e., 24.0% of the average delay time was eliminated. This indicates that the adjustments in the sequences of traffic flow in the waterways were greatly improved and the phenomenon of speed deceleration was alleviated when using the SSOCA model. Moreover, in comparison with FCFS, SSOCA’s advantage always held under different combinations of traffic flow parameters. It should be noted that four ships or more may change their sequences at higher arrival rates. Under such circumstances, the algorithm could also yield better performance, even if the possible additional interactions exist. The average delay time was reduced to 460 seconds for the simple SSOCA model applied to the base case. Only 16.3% of the reductions in delays showed that simple SSOCA was inferior to SSOCA. The average delay time was further reduced to 410 seconds for the base case using the secondary SSOCA model, which shows that the secondary SSOCA is superior to SSOCA. However, the SSOCA and secondary SSOCA results nearly overlapped, which means that only a small improvement was obtained with the SSOCA model. One reason is that a ship could seldom change its sequence beyond two times, considering the arrival time interval and speed variance between the ships set up in the experiments.

To identify the traffic parameters that contribute significantly to the uncertainties of the models’ average delay times, sensitivity analyses [12] of the four models were performed. The results are presented in Tab. 6. Mean value of ship speed was most sensitive to average delay time. Recalling the decrease tendency shown in Fig. 6(b), large ship speeds are beneficial for improving the efficiencies of restricted waterways. The sensitivity of the standard deviation of speed was approximately half the mean speed and was followed by waterway length and rate of arrival. The safety distance was the least sensitive to average delay time. Noting that a constant safety distance was assumed in the simulations, it can be inferred that the assumption did not have an apparent influence on the results. Nevertheless, more investigations examining different safety distance values are necessary. Meanwhile, comparing the four approaches, the sensitivities for SSOCA were basically smaller than those for FCFS and the simple SSOCA model, which reflects the advantage of the SSOCA model vis-à-vis stability.

Tab. 6 Parameter sensitivity of the four models

<table>
<thead>
<tr>
<th>Method</th>
<th>Rate of arrival</th>
<th>Mean speed</th>
<th>Standard deviation of speed</th>
<th>Waterway length</th>
<th>Safety distance</th>
</tr>
</thead>
<tbody>
<tr>
<td>FCFS</td>
<td>1.71</td>
<td>6.03</td>
<td>3.25</td>
<td>1.88</td>
<td>1.08</td>
</tr>
<tr>
<td>SSOCA</td>
<td>1.46</td>
<td>4.81</td>
<td>2.54</td>
<td>1.60</td>
<td>0.83</td>
</tr>
<tr>
<td>simple SSOCA</td>
<td>1.60</td>
<td>5.26</td>
<td>2.80</td>
<td>1.64</td>
<td>0.50</td>
</tr>
<tr>
<td>secondary SSOCA</td>
<td>1.43</td>
<td>4.71</td>
<td>2.45</td>
<td>1.54</td>
<td>0.81</td>
</tr>
</tbody>
</table>

According to Fig. 5, there was a large uncertainty in the results of the simulation model. The 95% confidence intervals are listed in the 3rd column of Pogreška! Izvor reference nije pronaden.. The half width of confidence interval for delay time was approximately 1.5% of the mean value, which shows that the mean values were adequately precise when the
simulations were repeated 200 times. Furthermore, the smaller confidence interval for the SSOCA model was found to be comparable with those for FCFS and the simple SSOCA, indicating that satisfactory convergences can be achieved.

**Tab. 7** Comparisons among FCFS, SSOCA, simple SSOCA, secondary SSOCA and all permutation for the base case

<table>
<thead>
<tr>
<th>Model</th>
<th>Average delay time (s)</th>
<th>Confidence intervals (s)</th>
<th>Computation cost</th>
<th>Optimization</th>
</tr>
</thead>
<tbody>
<tr>
<td>FCFS</td>
<td>550</td>
<td>[542.28 557.72]</td>
<td>0</td>
<td>None</td>
</tr>
<tr>
<td>SSOCA</td>
<td>418</td>
<td>[412.21 423.79]</td>
<td>m</td>
<td>Local</td>
</tr>
<tr>
<td>Simple SSOCA</td>
<td>460</td>
<td>[453.04 466.96]</td>
<td>m</td>
<td>Local</td>
</tr>
<tr>
<td>Secondary SSOCA</td>
<td>410</td>
<td>[404.26 415.74]</td>
<td>2m</td>
<td>Local</td>
</tr>
<tr>
<td>All permutation</td>
<td>Unknown</td>
<td>Unknown</td>
<td>m!</td>
<td>Global</td>
</tr>
</tbody>
</table>

In addition, as an important part of computational complexity theory, the analysis of algorithms [35] provides theoretical estimates for the resources needed by any algorithm that solves a given computational problem. These estimates provide insights into reasonable directions in the search for efficient algorithms. In consideration of low computational complexity, a computation cost model substituted for usual run-time analysis [36] was applied to compare the algorithms. The computation cost model assigns a constant cost to every computation for each ship. The amounts of computations for the different models are also listed in [Pogreška! Izvor reference nije pronaden]. In the FCFS model, ships are only required to report their own details to the VTS, and no computations are needed. A ship needs only make a single set of computations in the SSOCA and simple SSOCA models, but twice the computations are made in the secondary SSOCA model, which is unsatisfactory. Furthermore, as far as the central control mode is concerned, the number of computations even reaches $m$ factorial when using the all permutation algorithm with $m$ ships. Although the all permutation model is more likely to realize global optimizations, computational cost increases greatly with the number of ships. In a word, the SSOCA model is the best option for balancing performance and computation.

6. Conclusions

The SSOCA model, a self-organizing cooperation strategy, has been proposed in this paper. Delay time is presented as an evaluation indicator for traffic efficiency and includes wait time and deceleration time. A mathematic model of delay time with different sequences was deduced based on following theory. A self-organizing cooperation model was offered that enables slower ships to assign higher priorities to faster ships. By obtaining information on nearby target ships, each ship can interactively choose the optimum sequence in accordance with a local benefit delay time model. An Arena-based ship traffic simulation model was constructed to compare the SSOCA model with the FCFS model and two other associated models. The results show that SSOCA model can effectively reduce the average delay times suffered by ships to acceptable levels. Furthermore, the model can also obtain satisfying results with different combinations of factors, including high arrival rates. Moreover, a trade-off between efficiency and computation is achieved by allocating computation burden to each ship.
In future work, an anchorage and berth cooperation model can be considered by taking their capacities as constraints in order to make the self-organizing model more reasonable and practical. In addition, ultra-large-scale ships can only enter waterways during periods of high tides due to their deep draughts. In addition, pilotage is compulsory in particular areas, but there are often too few pilots for ships. These restrictions can also be considered in the models.

Acknowledgement

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Self-organizing cooperation model for ships navigating in restricted one-way waterway

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FREE SURFACE FLOW SIMULATION AROUND AN APPENDED SHIP HULL

UDC 629.5.015.2:629.5.018
Original scientific paper

Summary

This study brings forward the results of previously published work of free surface flow simulation around a fast ship model. Experimental measurements and numerical simulations of a fast bare-hull ship model form are now extended to the same ship form with appendices for a wide range of Froude numbers. The governing equations are discretized by means of an unstructured finite volume mesh. The standard $k$-$\varepsilon$ turbulence model and Volume of Fluid Method to capture the two phase media are used. The total resistance, due to wave and wake fields of the ship model with appendages and the resistance of the appendages alone are calculated numerically, and compared with the experiments. The experiments and computations were performed for 11 different Froude numbers between 0.103 and 0.322. For Froude numbers up to 0.25, numerical simulations found to be quite in agreement with the experiments. It has been found that appendages increase the total drag mainly by increasing the pressure resistance, and the effect of the appendages becomes more important as the flow speed gets higher.

Key words: Computational Fluid Dynamics; Experiment; Turbulent free surface flows; Appendages, $k$-$\varepsilon$ turbulence model.

1. Introduction

In naval architecture, advances in computational facilities have allowed scientists to use numerical simulations for effective design and optimization of hull geometry. Computational Fluid Dynamics (CFD) has advanced quickly in recent years and has become as one of the most important methods which plays crucial role in ship building industries. CFD methods are useful in analyzing flow problems in resistance estimation. While towing tank tests provide better absolute accuracy, CFD techniques can give practical results that are comparable to the towing tank test results with relatively less effort both in cost and time. While viscous flow methods give more accurate results in terms of drag than potential flow methods, potential theory to compute the free surface flow around the ship has been very much in use in recent years (see for instance [1-4]).
With the advent of the faster computers, researchers began studying flow around ship hulls with solving Reynolds-Averaged Navier-Stokes (RANS) equations and used the results in ship design. Subramanian and Vijayakumar [5] used the RANS based simulation for minimizing wake at propeller plane. This work examined the utility of CFD in the analysis of flow in the case of full aft beam vessels having characteristic cut stern shape to facilitate propeller aperture. The study demonstrated new avenues for applications of CFD in the early design stages. In the work of Celik [6], effect of the wake-equalizing duct on the propulsion performance of a chemical tanker was investigated by using RANS based simulation. Kouh et al. [7] studied scale effect on ship form factor for different hull geometries. In their study, form factor was predicted only based on double model calculation, in which influence of the wave making resistance was ignored. Park et al. [8] investigated different skeg geometries using RANS equations. Free surface effect was not taken into account by all these studies.

The following RANS approaches were applied with success for the free surface simulations around ship: Senocak and Iaccarino [9] simulated the turbulent flow around the DTMB 5415 model with free surface and they demonstrated the feasibility of such a simulation. Bucan et al. [10] made a study using both experimental and RANS based numerical solution around a tanker hull at model scale. Results of two numerical simulations performed at design and ballast loading conditions were presented. Zwart et al. [11] described an accurate, efficient algorithm for solving free surface flows around ship hull. The accuracy of the simulation was demonstrated first on Wigley hull. More detailed testing on the DTMB 5415 hull under a variety of conditions also showed good accuracy. Ahmed and Guedes Soares [12] gave a method for simulation of viscous and potential flows around a VLCC hull form. The results compared well with the available experimental data. Various works were performed on the flow around ship hull with a free surface, reviews on the subject can be found in Wackers et al. [13], and Xing et al. [14]. Sridhar et al. [15] studied friction resistance of a ship, Kandasamy et al. [16] investigated RANS solutions for a high speed catamaran, Pranzitelli et al., Tezdogan et al., Ozdemir et al., and Farkas et al. [17-20] have investigated the total drag around ship hulls.

Additionally, CFD is widely used for optimization of the hull form: Szelangiewicz and Abramowski [21] used the CFD code for influence of ship hull form modification on ship resistance and propulsion characteristics. Mahmood and Huang [22] made a study optimizing the bulbous bow. Duy et al. [23] used CFD for the optimal design for a stern shape of a hull. Muscari et al. [24] investigated hull-propeller-rudder interactions phenomena for twin-screw ships. Other studies using CFD codes are Bhushan et al., Gaggero et al., and Kim et al. [25-27].

The appendages can be responsible for an appreciable amount of the total ship resistance. The main appendages of the fast ship considered in this study are the twin shafting and shaft brackets. Normally, separate towing tank tests of a model with and without appendages are used to estimate the appendages resistance. The difference between the two-measured resistances should give the appendages drag. In the present study, CFD and Experimental Fluid Dynamics (EFD) are used to analyze the ship resistance problem of the M 367 fast ship with appendages. Ozdemir et al. [28] reported numerical and experimental results of the bare hull of the fast ship considered here. The fast ship hull form and its appendages were developed by Sener [29]. The model experiments were conducted at Istanbul Technical University Ata Nutku Ship Model Testing Laboratory. The next sections of this paper provides details about the geometry, experimental setup, governing equations, boundary conditions, mesh system, computations, validation, and conclusions.
2. Geometry and experimental setup

This study investigates a fast ship hull with appendages advancing in calm water with a free surface at different speeds. The ship hull is developed by Sener [29]. A detailed discussion of experimental setup is provided in Ozdemir et al [28]. The model is constructed on 1/36 scale with the model number M 367. The ship model was made of wood. Figure 2 shows the constructed model photographs of the M367 with and without appendages. Different CAD render of the hull geometry is depicted in Figure 3. Studs applied at the bow and on the rudder to stimulate turbulent flow (Figure 4). During the model resistance analysis, the air resistance was omitted and the ship the model is tested in free calm water condition free to dynamic trim and sinkage. Table 1 gives the model ship and main ship particulars. Table 2 gives the towing tank test conditions.

Fig. 2 Different views of the M367 ship model with and without appendages
Fig. 3 CAD render of M 367 ship model with and without appendages

<table>
<thead>
<tr>
<th>Table 1 Main particulars of the main ship and the model ship</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Length on the waterline L&lt;sub&gt;WL&lt;/sub&gt; (m)</strong></td>
</tr>
<tr>
<td>-------------------------------------------------------------</td>
</tr>
<tr>
<td><strong>Length between perpendiculars L&lt;sub&gt;BP&lt;/sub&gt; (m)</strong></td>
</tr>
<tr>
<td><strong>Moulded breadth B (m)</strong></td>
</tr>
<tr>
<td><strong>Moulded depth to upper deck D (m)</strong></td>
</tr>
<tr>
<td><strong>Design draft T (m)</strong></td>
</tr>
<tr>
<td><strong>Block coefficient C&lt;sub&gt;B&lt;/sub&gt;</strong></td>
</tr>
<tr>
<td><strong>Midship coefficient C&lt;sub&gt;M&lt;/sub&gt;</strong></td>
</tr>
<tr>
<td><strong>Prismatic coefficient C&lt;sub&gt;P&lt;/sub&gt;</strong></td>
</tr>
<tr>
<td><strong>Waterline coefficient C&lt;sub&gt;WP&lt;/sub&gt;</strong></td>
</tr>
<tr>
<td><strong>Design speed V&lt;sub&gt;S&lt;/sub&gt;</strong></td>
</tr>
<tr>
<td><strong>Displacement volume V (m&lt;sup&gt;3&lt;/sup&gt;)</strong></td>
</tr>
<tr>
<td><strong>Wetted surface area A&lt;sub&gt;WS&lt;/sub&gt; (m&lt;sup&gt;2&lt;/sup&gt;)</strong></td>
</tr>
<tr>
<td><strong>Total rudder area A&lt;sub&gt;R&lt;/sub&gt; (m&lt;sup&gt;2&lt;/sup&gt;)</strong></td>
</tr>
<tr>
<td><strong>Total appendages area A&lt;sub&gt;A&lt;/sub&gt; (m&lt;sup&gt;2&lt;/sup&gt;)</strong></td>
</tr>
<tr>
<td>Test conditions</td>
</tr>
<tr>
<td>-------------------------</td>
</tr>
<tr>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
</tr>
<tr>
<td>Gravitational acceleration (m/s²)</td>
</tr>
<tr>
<td>Kinematic viscosity (m²/s)</td>
</tr>
</tbody>
</table>

Fig. 4 Turbulent studs on ship bow and rudder

3. Mathematical formulation

3.1. Governing equations

The unsteady RANS equations for an incompressible, three-dimensional flow are continuity equation:

\[
\frac{\partial U_i}{\partial x_i} = 0
\]  

(1)

Momentum equation:

\[
\frac{\partial U_i}{\partial t} + \frac{\partial (U_i U_j)}{\partial x_j} = -\rho \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \nu \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \right] - \frac{\partial u_i' u_j'}{\partial x_j}
\]  

(2)

Where \( x_i \) is the spatial coordinate, \( t \) is the time, \( U_i \) is the mean velocity, \( u_i' \) is the fluctuating velocity, \( P \) is the mean pressure, \( \rho \) is the fluid density and \( \nu \) is the kinematic viscosity. The Reynolds stress tensor is modelled by the Boussinesq approximation. The eddy viscosity based standard k-ε turbulence closure model is used. A detailed description of the mathematical model is presented in [28, 30]. The pressure and velocity coupling problem is solved by using the SIMPLE algorithm where the velocity field is first solved using a presumed pressure. Then pressure and velocity fields are corrected with the calculated values of pressure and velocity [31]. Calculations are made in unstructured finite volume mesh for half of the model hull symmetric to its centerline.
3.2. Boundary conditions

The computational domain is limited by different boundary conditions. In the computations the following boundary conditions are used: external (inlet and outflow), symmetry, solid wall and free surface. For near wall flow regions wall function is used, where the viscous layer is not resolved. The dimensions of the computational domain are selected by the recommendations of ITTC procedures in order to prevent wave reflections. Therefore, the computational domain ranges from \(-3.0 \, \text{L}_{BP} < x < 3.0 \, \text{L}_{BP}, \ 0.0 < y < 2.0 \, \text{L}_{BP}\) and \(-1.0 \, \text{L}_{BP} < z < 1.0 \, \text{L}_{BP}\), where half of the body is modelled to decrease the computational domain size and time. The ship axis is located along the x-axis with the bow located at \(x = L_{BP}\) and the stern at \(x = 0\). The still water level lies at \(z=0\). The dimensions of the computational domain satisfy the well-known ITTC procedure. Detailed information about the principles of computational domain dimensions’ selection strategy can be found in [18] and [19]. Also, a detailed description of the boundary conditions is given in [28]. The general view of the computational domain and the boundary conditions are shown in Figure 5.

Fig. 5 The general view of the domain and boundary conditions

3.3. Mesh structure

In this study, an unstructured hexahedral mesh is used. Hexahedral mesh with minimum cell skewness permits flexibility, especially in local mesh refinement for free surface waves. The numerical mesh created for this study is given in Figure 6. In general, grid points are grouped around the hull, its appendages and calm water plane in the vertical range of expected wave heights to provide adequate resolution at the free surface interface. For the
viscous flow simulation, 4 layers of prismatic cells are applied around the hull and its appendages. The stretching factor of prism layers is 1.5. Local mesh refinement is accomplished by means of volumetric controls of predefined geometrical shapes and the total number of the grid points for the mesh structure is shown in Table 3.

**Fig. 6** The mesh structure for the M367 hull with appendages, close-up view
Because of the computational restrictions, the turbulent studs could not be used in the mesh system. The construction of a good mesh is crucial to the success of a CFD analysis. For reliable and dependable drag force results in the simulations of flow around hulls, the researcher should spent hefty time in mesh designing. Mesh refinement through adaption is also very important where needed to resolve the flow field around the hull. The mesh dependency study was carried out by Ozdemir et al. [28] for $Fr=0.201$. As the computations using the same approach with the similar mesh resolution with the bare hull configuration given in [28], a mesh dependency study has not been performed for the appended hull. Medium mesh is quite good to calculate the total resistance.

4. Results and discussions

Model experiment and numerical simulation of appended hull is investigated similar test condition and Froude numbers with the bare hull for calculate the effect of the appendages on the resistance. Simulations presented in this study were performed for 11 selected test conditions. The main motivation is to examine the effect of appendages on the ship resistance. Convergence of the iterative solution is mediated by the normalized residual of free surface elevation is less than $10^{-2}$ and residuals of all the remaining variables are less than $10^{-5}$. The computations are made on an 8 CPU workstation with 3.4GHz, on windows Win7 system. Explanations of the numerical method can be found in [32]. The time step $\Delta t$ is chosen to be 0.01 s based on the ITTC CFD guideline [33].

4.1. Resistance

Table 4 shows the measured and computed total resistance values and the difference between computation and the experiment for given speeds for the M367 ship model with and without appendages. For bare hull, the comparison with the experimental measures is very good for Froude numbers between 0.103 and 0.239. Within the given range, the maximum error is 5.90% at $Fr=0.239$. The RANS computations generally predicted lower total resistance values. The model tests indicate that the computations predicted 9.44% lower total resistance at $Fr=0.264$, 12.40% lower total resistance at $Fr=0.286$ and 17.96% lower total resistance at $Fr=0.322$ compared to the experiments. For the appended hull, the comparison with the experimental measures is very good for Froude numbers between 0.127 and 0.215. Within the given range, the maximum error is 6.92% at $Fr=0.215$. Computed and measured total resistance for M367 model can be seen at Figure 7. Convergence history of the total resistance of appended ship is given in Figure 8. Allowing adequate time for the free surface to develop around the model and the drag force converge, the simulations are calculated for

<table>
<thead>
<tr>
<th>Block #</th>
<th>Block Name</th>
<th>Medium mesh</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Control volume</td>
<td>0.085 L</td>
</tr>
<tr>
<td>2</td>
<td>Free surface</td>
<td>0.01 L</td>
</tr>
<tr>
<td>3</td>
<td>Near ship</td>
<td>0.01 L</td>
</tr>
<tr>
<td>4</td>
<td>Ship’s aft-end</td>
<td>0.002 L</td>
</tr>
<tr>
<td>5</td>
<td>Ship’s fore-head</td>
<td>0.002 L</td>
</tr>
<tr>
<td></td>
<td>Total number of generated cells</td>
<td>1,494,643</td>
</tr>
</tbody>
</table>
50 seconds. Both from Table 4 and Figure 7, mutually experimental and computational results show that the resistance of the hull with appendages is greater than the bare hull as expected. One of the reason is that increasing of the wetted surface area due to the appendages.

<table>
<thead>
<tr>
<th>Fr</th>
<th>V (m/s)</th>
<th>Experiment</th>
<th>CFD</th>
<th>Difference</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td>Bare Hull</td>
<td>Appended Hull</td>
<td>Bare Hull</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$R_{TM}$ (N)</td>
<td>$R_{TM}$ (N)</td>
<td>$R_{TM}$ (N)</td>
</tr>
<tr>
<td>0.103</td>
<td>0.634</td>
<td>1.988</td>
<td>2.088</td>
<td>1.926</td>
</tr>
<tr>
<td>0.127</td>
<td>0.784</td>
<td>2.947</td>
<td>3.167</td>
<td>2.822</td>
</tr>
<tr>
<td>0.149</td>
<td>0.918</td>
<td>3.913</td>
<td>4.287</td>
<td>3.879</td>
</tr>
<tr>
<td>0.168</td>
<td>1.037</td>
<td>4.902</td>
<td>5.436</td>
<td>4.893</td>
</tr>
<tr>
<td>0.186</td>
<td>1.143</td>
<td>5.893</td>
<td>6.519</td>
<td>5.929</td>
</tr>
<tr>
<td>0.201</td>
<td>1.239</td>
<td>6.874</td>
<td>7.593</td>
<td>6.892</td>
</tr>
<tr>
<td>0.215</td>
<td>1.325</td>
<td>7.845</td>
<td>8.662</td>
<td>7.900</td>
</tr>
<tr>
<td>0.239</td>
<td>1.474</td>
<td>9.785</td>
<td>10.745</td>
<td>9.208</td>
</tr>
<tr>
<td>0.264</td>
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<td>12.245</td>
<td>13.270</td>
<td>11.090</td>
</tr>
<tr>
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<td>1.758</td>
<td>14.736</td>
<td>15.886</td>
<td>12.909</td>
</tr>
<tr>
<td>0.322</td>
<td>1.980</td>
<td>19.650</td>
<td>21.066</td>
<td>16.123</td>
</tr>
</tbody>
</table>

Fig. 7 Computed and measured total resistance for the model with and without appendages

Computed and measured appendages resistance is depicted in Figure 9. It can be seen that, appendages resistance alone increase with the growth in the Froude number. Comparison
between experimental and numerical results reveal that for Froude numbers between 0.186 and 0.239, appendages resistance differences are increased. The obtained differences are believed due to the shape of the appendages, which may trigger transition from the laminar flow to the turbulent flow. On the other hand, in the experimental study, turbulence studs were used which may influence the experimental results. A faster flow will produce higher $y^+$ values. The precision of $y^+$ values determines the quality of boundary layer solution, which affects the friction force. During the simulations $y^+$ values are checked in every speed analysis. The numerically evaluated non-dimensional $y^+$ wall distance variations on the model for two different speeds are given in Figure 10. It is seen that the $y^+$ values of the first grid points above the model hull are ranged between 30 and 120, as required.

![Figure 8](image1.png) **Fig. 8** Convergence history of total drag during computation for Fr=0.201 and Fr=0.322

![Figure 9](image2.png) **Fig. 9** Comparison of computed and measured appendages resistance

Figure 11 shows body contours of the pressure coefficient on the M367 ship model with and without appendages for two different speeds (Fr=0.215 and Fr=0.322). The areas with greater pressure coefficient values have a significant contribution to pressure resistance.
Free Surface Flow Around Appended Ship Hull

Yavuz Hakan Ozdemir, Baris Barlas

4.2. Wave field

Concerning the numerical simulation, Figure 12 shows the wave contours of Kelvin type wave pattern around the appended hull for two different speeds (Fr=0.215 and Fr=0.322). Figure 13 depicts, typical bow-wave and stern-wave patterns around the appended hull for two different speeds (Fr=0.215 and Fr=0.322). Figure 14 provides the close-up perspective views of the computed bow-wave and stern-wave patterns and photographs around the appended model during the experiment at the same speed of Fr=0.215. The computation shows the formation of a thin sheet of water close to the bow. At the wake field, transom separations can be seen clearly both in CFD solution and the experiment.
Fig. 12 Global wave pattern for the appended hull for Fr=0.215 and Fr=0.322.

Fig. 13 Local wave patterns in bow and stern regions for the appended hull for Fr=0.215 and Fr=0.322.
4.3 Wake field

Wake survey involves a detailed investigation of the flow characteristics through the propeller disc. The wake field greatly depend on ship type. Each ship hull can be considered to have a unique wake field. Nominal wake is obtained based on wake survey carried out by using either experimental or numerical fluid dynamics methods without the presence of propeller. The nominal wake is crucial in propeller design. In this study nominal wake behind the appended hull is investigated. A Pitot tube was used to measure the velocities in the wake field behind the hull. The Pitot tube and experimental setup is given in Figure 15.
The comparison of the wake plane is depicted in Figure 16. General behavior of the wake distribution seems comparable when matching numerical and experimental results. Although, in some parts of the wake region the values of the computed velocities are greater than the experimental data and the present method does not agree with the experimental results. The authors believe that a more sophisticated turbulence model could solve this issue. When the appendages (propeller shafts and brackets) are taken into account, the computation of the primitive variables near the aft-end part of the hull (especially near the propeller domain) becomes complex. Especially, the helical motion, local separation of the individual appendages, adverse pressure gradient effect and interaction between the devices make the flow structure much more complicated. Figure 17 shows the streamwise velocity contours, similarly, Figure 18 shows the turbulent kinetic energy contours calculated at Fr=0.322. The turbulent kinetic energy (Eq. 4) formed predominantly by the convective terms.
5. Conclusions

This paper presents the numerical and experimental results of the flow field around a fast ship model with appendages. Although some experimental data have been showed for validation of CFD results, the main objective of this study is to assess the performance of CFD for design, analysis and feasibility of such a simulation for naval architects and shipping industry. The following conclusions are reached:

- Both experimental and numerical results show that ship resistance increase with appendages.
- As expected, numerical simulation around ship hull with appendages is much more problemmatic compared to numerical simulation around ship hull without appendages.
- Mesh refinement through adaption is critical, especially near the appendages region to resolve the flow field characteristics.
- The simulated wave pattern around the ship hull is in good agreement with the experimental results.
- The experiments and computations were performed for 11 different Froude numbers between 0.103 and 0.322. It is suggested that $k-\varepsilon$ turbulence model may be considered as a useful tool for predicting viscous flows with free surface around an appended ship model for Froude numbers up to 0.25.
- The effect of the appendages on the ship become more important as the flow velocity increases.
- Appendages increase the total resistance mainly by increasing the pressure resistance, because the total appendages area is insignificant compared to the wetted surface area.
The appendages presence caused an acceleration of the flow. Turbulence kinetic energy is getting higher due to the appendages.

Extension of this study is also encouraged in order to examine the effect of different mesh forms and numerical solver methods on the precision of the obtained results. Towing tank test results may be affected by turbulent studs. Using turbulent studs are a standard procedure in towing tank experiments. The surface of the numerical ship model can be considered as a rough surface by using a roughness parameter where the studs are placed. It is also highly recommended to use a more sophisticated turbulence model without using wall functions to well capture the flow properties around the aft-end of the model with appendages. Inherently a suitable low Reynolds number turbulence model is advised for the future work.

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Free Surface Flow Around Appended Ship Hull

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METHODS OF CRACKS DETECTION IN MARINE STRUCTURES' WELDED JOINTS BASED ON SIGNALS' TIME WAVEFORM ANALYSIS

UDC 629.5.015.4:629.5.083:620.193
Preliminary Communication

Summary

The paper presents two methods of crack detection in ship hulls. The methods are dedicated for structural health monitoring (SHM) of responsible welded joints. The system will be based on vibrodiagnostic - signals will be measured by piezoelectric accelerometers and/or fibre optic sensors. In SHM systems of welded joints of thin-walled structures, a vibrodiagnostic method is the most promising. Its most important advantage is that it is both effective and relatively low costly. There are two general methods of vibrodiagnostic signal analysis: the most frequently used are spectrum analysis and time waveform analysis. The spectrum analysis concerns changes of natural frequency. In marine exploitation conditions, the frequencies changes might be imperceptible. The first method presented in the article is based on the evaluation of the mean value distribution of the amplitude spectrums calculated with the time window method. Second proposed method was based on the determination of damping decrement in function of time. Due to the complexity of the responses run, the proposed method consisted of calculating the damping decrement using the response approximation with different functions. It has been shown that the changes analysis of damping decrement applied to welded plates enables the assessment of the quality associated with the weld. A calculation algorithm as well as and the exemplary results from the proposed methods used for some selected samples with different type of welds are included in the paper. The results of the tests show that the analysis based on proposed methods indicates that they differ significantly depending on the welds, indicating their quality and cracks that are associated with them.

Key words: ship hull diagnostic; welded joints; SHM; damping decrement; time window analysis

1. Introduction

Among various means of transport, maritime transport (except for the air transport) is exposed to relatively highest risk. Seagoing vessels often work under extremely difficult environmental conditions. Moreover, maritime structures like marine vessels, submarines and
offshore structures are exposed to the influence of aggressive marine environment for a very long time. The structures surrounded by a harsh marine environment are exposed to long-term cyclic loadings [1] which come from continuously acting sea waves and short-term extreme loads such as severe storms, seakakes or even collisions. The marine environment causes fast corrosion, erosion and scour processes. Those phenomena increase the size of an existing damage and also initiate its growth [2, 3]. The example of disastrous damage (fracture) of a bulk cargo ship, due to a very strong stormy weather, is presented in Fig.1. This type of structural damage has a catastrophic impact on the safety of the crew, the marine environment, and the ship's cargo.

Fig. 1. A maritime disaster resulted from the collapse of the hull structure [http://portalmarynarski.pl]

Each marine vessel and facility works under the supervision of one of the classification societies. They require a detailed periodic safety inspection of the structure from the stage of its design to construction. The welded joints are one of the key elements which undergo precise diagnostic testing. All responsible welds are tested using measurement techniques generally called NDT (Non-Destructive Testing) [4, 5]. Currently, the hybrid tests are becoming the standardized part of NDT. They use mixed approach that combines two or more methods. Hybrid methods are divided into two groups: using the same physical phenomena and acting on the basis of complementary methods of research. An example of a hybrid approach using the same physical phenomenon is the combination of acoustic emission and ultrasound testing. An example of a hybrid approach that is based on acting on the basis of complementary methods of research is surface testing combined with radiographic and/or ultrasonic testing [4, 6]. NDT techniques are well tested and give the results which are sufficiently reliable. However, they have one basic flaw – they are run periodically. In the intervals between tests, the reliability of the structure is uncertain [7, 8]. Particularly in the case of critical events (extremely strong storm, collision, etc.), it is important to note the possibility of further operation of the facility. This information must include the degree of danger of disasters along with the operational parameters, including the time of its safe use. To this end, the new techniques are being developed. They are known as SHM (Structural Health Monitoring) [9, 10].

Structural Health Monitoring is a natural extension of NDT diagnostics of machines and devices. The fundamental difference between SHM and NDT relies on hardware architecture [11]. In case of SHM systems, the network of sensors is integrated with the object while the sensors network used in NDT is external and independent. SHM systems operate on-line, contrary to NDT. The key element of SHM systems is the automation of their operations (e.g. generating the reports, informing the staff about non-standard parameters) on the base of artificial intelligence technology [12, 13]. Monitoring may be based on the range of, often very different, measurement techniques. The most popular techniques are: methods based on
testing of characteristics of dynamic structures, acoustic emission, study of elastic waves of Lamb Waves type along with spectra finite element method, thermo-vision methods, ultra-fast framing cameras, layered testing of electromagnetic characteristics, vacuum comparative research and the methods based on fibre optic sensors. The aim of SHM systems is to create the measuring and diagnostic system which would enable to assess the technical condition of the structure continuously and in various environmental and operating conditions. The typical SHM system consists of the net of sensors which continuously measure condition of the structure as well as environmental and operating loads [14]. One of the most important information is strain/stress distribution in the structure. One of the most promising sensors for that purpose are those based on fibre optics technology [15]. Two general techniques are analysed. The first one is based on Optical Time-Domain Reflectometer method (OTDR), and the second one is based on Fibre Bragg Grating (FBG) sensors [9, 16]. Amplitude values of the vibration level can be also useable for validation of the structure condition [17]. Typical piezoelectric accelerometers can be used for that type of measurements. The task of system’s central unit is to collect and record measurement signals, analyze them (including selection) and to automatically diagnose occurring risks [9].

Works on the elements of the structure monitoring, meaning: detection, localization and identification of damages, are being intensively developed but mostly they are reduced to laboratory and/or preliminary testing [18, 19]. Moreover, shipbuilding works are relatively underdeveloped (e.g. in comparison to aviation). Complete monitoring complements the detection, localization and flaw identification systems by reliable lifetime prediction of the structure and assessments of its further emergency operation [20]. There are no simplistic but reliable mathematical models for static and dynamic evaluation, parameters (reliably relevant for shipbuilding) in the marine industry. These models should be able to be used in systems based on artificial intelligence. Evaluation of the key measurement elements and their effective selection for such a system is essential [9, 16].

Monitoring based on vibrodiagnostic techniques is one of the most promising types due to its simplicity and relatively low costs. So far, other techniques do not have practical application. For example, tests which use elastic waves require extremely expensive measuring equipment (e.g. 3D laser) difficult to use in operating conditions for such complex construction as the hull of a ship [21, 22, 23]. One of the most important elements in SHM system is recorded signal analysis. There are two general methods of vibrodiagnostic signal analysis: spectrum analysis (most commonly used) and time waveform analysis. The spectrum analysis concerns changes of natural frequency. In marine exploitation conditions the frequencies changes might be imperceptible [1, 24]. The first method built on waveform analysis presented in the article is based on the evaluation of the mean value distribution of the amplitude spectrums calculated with the time window method. The statistical measure used in the form of the mean value is a proposed parameter. The parameter's analysis for a given welded joint may enable an unambiguous assessment of its quality. Second proposed method was based on the determination of damping decrement in function of time. Due to the complexity of the responses run, a proposed method consisted of calculating the damping decrement using the response approximation with different functions. It has been shown that the changes analysis of damping decrement applied to welded plates enables the assessment of the quality associated with the weld.

2. General assumption for the monitoring system

Diagnostic system of the ship hull (thin-walled structure) is a target of the authors' research. The work in abovementioned paper is the first step of the planned research. The main goal of presented work stage is determination of diagnostic parameters sensitivity. Time
window and damping decrement methods have been taken into consideration. After measurement methodology determination, the authors planned next step of the research - research onboard of the real ship (Horyzont II - the ship of Gdynia Maritime University). Excitations will be coming from environmental influencing (e.g. waves). Planned system will be looking for nonlinearities (with using developed diagnostic parameters) in the response signals measured by accelerometers. Proper dynamic characteristics of the ship hull structure will be checked by the system. General assumption of the project of on-line diagnostic (monitoring) system of the thin-walled structure working parameters is analysed in the paper. Three general types of measurements are planned [5, 12, 20]. The first one is a typical vibrodiagnostic measure ment based on piezoelectric accelerometers (PZT). In marine conditions seismic sensors are useful because frequencies of the sea wave loadings are very low. Characteristic of typical PZT starts from 5 Hz in contrast to seismic which starts from values below 1 Hz. The sensors will be placed around the most important welds. Global deformations of the ship hull (and external excitations of the foundation system) will be measured by OTDR techniques. Local strain changes (around the most important welding joints) of the structure will be measured by system based on FBG sensors with interrogators. Continuous recording will be performed by planned monitoring system. Online data analysis allows distinguishing between typical signal (for normal ship operations) and unusual signals coming from extreme loadings. Only those unusual signals will be recorded and thoroughly analysed.

2.1 Optical Time Domain Reflectometer measurement techniques

An optical time-domain reflectometer (OTDR) is an optoelectronic measurement technique used to characterize an optical fibre [9, 16]. The "sensor" for OTDR measurement techniques is a typical commercial, telecommunications fibre. An OTDR injects a series of optical pulses into the fibre under test. Light is reflected back from points along the fibre. The strength of the return pulses is measured and integrated as a function of time, and is plotted as a function of fibre length. Optical Backscatter Reflectometer (OBR) instruments are used for strain and temperature measuring along pure fibre optic. OBR measuring device can measure distributed strain and temperature in standard telecom-grade fibres with high spatial-resolution. The OBR uses swept wavelength interferometry to measure the Rayleigh backscatter as a function of length in optical fibre. The system allows practical measuring of distributed temperature and strain in standard fibre with millimetre-scale spatial resolution over tens to hundreds of meters of fibre with strain and temperature resolution as fine as 1 με and 0.1°K.

Measurement techniques are based on Rayleigh backscatter in optical fibre which is caused by random fluctuations in the index profile along the length of the fibre. The scatter amplitude, as a function of distance, is a random but static property of that fiber can be modelled as a continuous, with a random period. The spectral frequency associated with the Rayleigh backscatter is written in the same form as the reflection frequency of a Bragg grating. Shifts in the fibre index of refraction or in the average perturbation period caused by an external stimulus (like strain or temperature) in turn cause shifts in the local spectral frequency of the Rayleigh backscatter. Accumulated changes along the optical path also manifest themselves as a time shift of the Rayleigh backscatter return loss amplitude pattern. Performing a cross correlation on the backscatter amplitude time domain or frequency domain data accurately measures these spectral and temporal shifts, which are easily scaled to form distributed temperature or strain measurements. An example of the measured signal received from optical backscatter reflectometer offered by Luna firm is shown in Fig. 2.
OBR measuring techniques can be used to measure the distributed spectral shift and temporal shift in the Rayleigh backscatter along an optical fiber. This capability facilitates distributed temperature and/or strain sensing along any standard single-mode fiber and this technique enables robust temperature and strain measurements with high spatial resolution and good accuracy. This measurement capability also provides a practical alternative to fiber Bragg gratings sensors and extrinsic Fabry-Perot interferometric sensor in situations where a large number of closely spaced measurements are desired.

On the base of that kind of the system, measurements of global deformations of ship hull are planned. The OTDR based system is dedicated for analysis of quasi-static deformation of the ship hull. The sampling frequency of that system will be equal to 5 Hz. The length of the "sensor" will be between 30-50 m. It will be placed around the midship section. Highest spatial resolution between two measured points is equal to 10 μm, but for the planned system 10 mm is enough. Expected strain relative accuracy is about 10⁻⁵. Extreme loadings like storms, ship grounding, and freak waves can be measured by the system [25]. Detailed data of continuous distribution of excitation forces acting on propulsion system will be recorded by OTDR based system.

2.2 Fibre Bragg Grating measurement techniques

The systems based on fiber optic technique with Fiber Bragg Grating (FBG) strain sensors are one of the most interesting and promising [13, 17, 26]. In comparison to classical strain measuring method involving electric strain gauges, the new technique using fiber optic technology and FBG sensors is much more stable and the measuring error is much lower. The main benefits of fiber optic (in particular FBG sensors) have been found in their long-term stability and reliability as well as in their insensitivity to the external perturbations like electromagnetic fields [9]. One of the main advantages of FBG sensors is the ability to measure multiple physical parameters [12]. This ability combined with serial multiplexing of FBG sensors allows for multiple parameters to be monitored. This feature is advantageous in applications where minimal intrusion into an environment is required. Another important advantage of FBG sensor is multiplexing ability - many sensors (to over hundred) can be multiplexed to provide measurements across the structure [16]. FBG sensors have several promising assets in comparison to conventional techniques, the main are as follows:

- high sensitivity;
- low sensor’s size and mass;
- sensors can be built into the monitored structure (e.g. composite material);
- immunity to electro–magnetic fields;
- applicability to chemical aggressive surroundings;
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- multiplexing (hundreds of sensors in one channel);
- can support thousands of kilometers unrepeated step out distances;
- self–calibrating and free from signal drift (long term stability).

The history of optical fibres reaches back to the 1960s [12]. But these fibres had a damping of 100 dB/km caused mainly by the chemical impurity of the glass. Today the damping is <0.2 dB/km. In 1978 the effect of photo sensitivity for Germanium doped fibres has been found. Exposure to ultraviolet light induces a permanent change of the refractive index. The next step was to use this effect and write Bragg gratings into fibres which then can reflect very small wavelength peaks. The wavelengths of these peaks change with strain and temperature. First commercial FBG sensor was available in 1995.

Bragg gratings are written into single–mode fibres. These fibres consist of a very small inner core (diameter 4–9 µm) and an outer part (cladding) of pure glass (SiO2) of 125 µm diameter. The core has a higher refraction index caused by high Germanium doping. The core has a higher refraction index caused by high Germanium doping. The difference of refraction index between inner core and cladding causes the light to propagate only inside the small core. The glass fiber is coated with acrylate, polyimide or organic modulated ceramic to protect it especially against water and hydrogen which causes crack growing and can reduce the mechanical stability.

Each single fringe reflects a very small part of all incoming wave. The reflection factor per single fringe is in the range of 0.001% up to 0.1%, depending on how much energy was used to write the Bragg grating and on the percentage of Germanium doping of the fibre core. Each single fringe reflects light with different phase shifts. Therefore, interference is a reason why the most of the light is erased. But the reflections with equal phase shift accumulate in a strong reflection peak. The reflection of whole grating is the sum of all these thousands of very small single reflections. Reflected light travels forth and back in the fibre, therefore reflected light beams of the single grids are in phase if an integer of light wavelength fits into two times the grid distance. The grid spacing can be calculated as follows:

$$A = \frac{\lambda_0}{2n}$$

where:

- $\lambda_0$ – wavelength peak,
- $n$ – refraction index of the fibre.

Because typical values of the FBG is: $\lambda_0=1550$ nm and $n=1.46$, the grid spacing should be equal to 530 nm. If the length of the FBG sensor is equal to 5 mm then the number of fringes in the single sensor should be approximately equal to ten thousand.

A FBG sensor (see Fig. 3) has a periodic structure [16]. When light within a fibre passes through a FBG, multiple reflections take place along the entire length of the grating due to the variations in refractive index. Constructive interference between the forward wave and the contra–propagating light wave occurs when the wavelength of the propagating light in the fibre doubles the grating pitch. This leads to narrowband back–reflection of light. A fibre–optic Bragg grating sensor acts as a filter for light running along the single–mode fibre line. Reflected wavelength $\lambda_B$ is a function of sensor’s strain (affected by external stress and/or temperature field). The reflected wavelength is known as the Bragg wavelength $\lambda_B$ and given by:

$$\lambda_B = 2 \cdot n_e \cdot A$$
where:

\( n_e \) – effective refraction index of the fiber core,

\( \Lambda \) – the period of the index modulation.

Both \( n_e \) and \( \Lambda \) depend on temperature and strain, therefore the Bragg wavelength is sensitive to both strain and temperature. The expansion coefficient of the fibre is negligibly low. The biggest impact results from the temperature dependent change of the refraction index. When a fibre is fixed to a specimen, the FBGs signal changes with the mechanical and temperature caused strain of the specimen. Therefore the thermal coefficient is equal to expansion coefficient of the specimen (not glass). It leads to equation for strain–measuring FBGs:

\[
\varepsilon_m = \frac{1}{k} \cdot \frac{\Delta \lambda_B}{\lambda_B} - \left( \alpha_{sp} + \frac{\alpha_\delta}{k} \right) \cdot \Delta T
\]  

(3)

where:

\( k \) – gauge factor,

\( \alpha_{sp} \) – expansion coefficient of the specimen,

\( \alpha_\delta \) – change of the refraction index.

Because temperature has a very strong impact on the FBG sensors, precise strain measurements can only be achieved with proper temperature compensation. When the FBG sensor is fixed to the specimen on a surface without mechanical strain, it works as temperature compensation sensor. In measurements at very high temperatures (hundreds °C) it must be taken into account that the base wavelengths of the sensor change considerably with temperature.

If FBG sensor is strained, the wavelength of the reflection peaks is shifted. It is necessary to measure these shifts very precisely. Resolution and short–term stability of ±1 pm is required [26, 27]. For laboratory investigations, interferometers are often used. In the commercial equipment usually other principles are applied. An example of modern FBG sensor produced by Micron Optics is presented in Fig. 4. Nowadays the top–class instruments are using tuneable lasers. The interrogator uses broad–band light source and therefore only a very small part of the light energy is related to the small bandwidth of a BFG. Therefore, the reflected peak energy is very low. A tuneable laser concentrates all its energy in an extremely small bandwidth and by sweeping over the whole bandwidth range it scans the spectrum with high power and can provide an excellent signal–to–noise ratio.
The FBG sensors will be mounted in the midship section [23, 25]. Monitoring system based on the FBG sensors will measure local strains (stresses) of the ship hull. The sampling frequency of that system will be equal to 20 kHz. Therefore, that part of monitoring system is dedicated for high-frequency dynamic analysis of extreme loadings. Obviously, quasi-static strains will be also recorded. The typical length of the sensor is between 2-20 mm. For our local strain measurements, 10 mm is the optimum length of the sensor. Three FBG sensors (triangle or star type of arrangement) for each measured point are planned. The sensitivity and accuracy of FBG sensor with a typical interrogator are even higher in comparison to OTDR system; therefore, it is sufficient for our research.

3. Laboratory measurements of the sample of welded joints

The test stand to conduct the testing of welded joints using the vibrodiagnostic method [5] was constructed at the Faculty of Marine Engineering at Gdynia Maritime University. The structure of the stand and its most important parts are presented in Fig. 5. The stand includes the holder (1) which can be used to install the plates (welded test pieces) horizontally - 4-point mounting or vertically – 2-point mounting, vibration analyzer (2) produced by Bruel & Kjaer, two accelerometers (3), modal impact hammer (4) with three interchangeable heads i.e. metal, silicon, and teflon.

During the process of preliminary tests, the plates were installed vertically on the holder as it is presented in Fig. 5. The tests were conducted on four plates. The plate marked by number 0 was homogenous and did not have welded joints. The other three included welded plates that were marked by number: 2202 – the plate that did not have any flaws, 2127 – the plate that had flaws in the form of boundary bonding and 2132 – the plate with simulated cracks along the whole length. All test pieces that had welded joints were tested using the radiographic method before the measurement. It enabled the assessment of the joints quality along with identification and placement of flaws in the plates.
The measurements of vibration generated by the plates were taken in the prepared test stand. The vibration was caused by the impact hammer with different heads: metal, silicon, and teflon. The places of strokes are presented in Fig. 6, described by means of $F1$, $F2$, and $F3$. The results were read by the accelerometers $ACC1$ and $ACC2$.

![Fig. 6. Schematic diagram showing the arrangement of accelerometers (ACC1, ACC2), places of strokes (F1, F2, F3) and plates mounting places in the holders (Δ)](image)

In the article, the calculations were made for the measurement results obtained using a metal head installed on a modal hammer.

4. **Time window method in time-frequency analysis**

The time window method enables simulation of concurrent signal analysis in the field of frequency and time. This opportunity is often used in the analysis of technical systems dynamics, especially electrical or electronic ones. It is used in mechanical systems testing very rarely. In case of time window method, the key element which influences the quality of results is a proper selection of the type and width of the time window. Then the FFT analysis (Fast Fourier Transform) is carried out for each of windows.

The time windows method was used in conducted tests in order to analyze the responses which were recorded by accelerometers placed on plates with welded joints. The tests were done for rectangular window in terms of which various time intervals were checked. The decision to choose the rectangular window comes from the willingness to reduce the distortions of recorded course of response to the least extent. This kind of situation often takes place in case of more complex windows. Then at the cost of accepted changes introduced from the signal, it is possible to eliminate the part of distortions through the window.

Conducted tests aimed at the optimal coverage of spectrums characteristic changes which determine the quality of welds and possible defects. Eventually, the chosen width of the window was 10 ms. The examples of set of amplitude spectrums calculated for time windows are presented in Fig. 7 and 8. The figures contain chosen results obtained for the metal head. The first range covers the time windows from 1 to 12 and the second one from 10 to 21. The three time windows, from 10 to 12, are been repeated in both figures in order to show the transition from one range to another. The analyses were done for the plate without the weld and with different welded joints. The plate response to the signal made with a modal impact hammer with metal and teflon head was registered by accelerometer for all plates.
The comparison analysis of amplitude spectrums, calculated for each window, shows nonlinearities of structures of damaged welded plates. If the quality of welded joint is worse, the nonlinearities are stronger, that is when the defect is shown in the welded joint. In presented example (Fig. 7 and 8) this kind of situation concerns welded plate with the flaw in form of boundary bonding. Moreover, the preliminary analysis of spectrums shows that most diagnostic information comes from first ten time windows which mean about 100ms of analyzed response registered by accelerometer. Presented amplitude spectrums are characterized by similar character of distribution, especially in terms of harmonics, however, important diagnostic information is given by the spectrum of higher harmonics, which analysis requires additional mathematical operations. Proposed parameter to compare the amplitude spectrums calculated for each window of the same sample and to evaluate the nonlinearity resulting from welded joint in a plate was the mean distributions of amplitude spectrums. The mean value is a measure of descriptive statistics which was calculated from the equation (4).

\[
|v| = \frac{\sum_{i=0}^{N} |v(f_i)|}{N}
\]  

where:

- \( |v| \) - the mean of amplitude spectrum velocity calculated for the time window,
- \( f_i \) – the frequency of \( i^{th} \) harmonic,
- \( N \) – the number of harmonics.

The preliminary tests have shown that the analysis of only mean values is not sufficient. However, the relevant diagnostic information about the quality of welded joint is given by mean value distribution for the spectrums obtained from the whole response of time windows. Fig. 9 shows some examples of amplitude spectrums taken from chosen time windows and the values calculated for them.
The distribution shown in Fig. 10, presents the alteration of the mean value calculated for the amplitude spectrum in a given window in function of window number for a chosen welded plate. Presented characteristics show that the mean distribution for windows calculated for the sample without the weld (marked as number 0) differs significantly in comparison to mean distribution of other samples with weld or samples with a flaw in welded joint (marked as: 2202, 2132, 2127). This difference is especially applicable in terms of the level of mean values and cyclic value change between local maximums and minimums. It is also possible to notice that the mean value distributions for the samples with the weld and a flaw are characterized by lower value than for the plates with weld and without flaws.

![Fig. 9. Amplitude spectrums in chosen windows with the designated mean value](image)

![Fig. 10. Standardized distribution of mean value of the spectrums for windows in function of window number](image)

5. **Applying the damping decrement distribution to flaw assessment of welded joints**

Using the vibrodiagnostic method in tests on welded joints allows analyzing the amplitudes of structure vibrations, in this case, made by modal hammer. While testing, it is
possible to record few responses – vibration amplitudes. In order to achieve that, the accelerometers were placed in different locations on the tested plate. Recorded responses take the form of damped oscillatory motion. The weld (including its quality), which connects two plates, influences the disappearance speed of diagnostic signal (the speed of energy dissipation), and this is why it is possible to use a damping decrement in its assessment. The damping decrement is commonly used in assessing the dynamics of many mechanical systems. However, due to the variable nature of response in time, it is impossible to apply regular formula for the logarithmic decrement, expressed as a time-independent constant. Two forms of logarithmic decrement were derived in this work and they are represented by equation (5) and (6). Statement (5) is used to determine averaged structure damping in regard of maximal amplitude; statement (6) is used to assess the damping changes during single vibration periods or its groups.

\[ \Psi_I = \frac{1}{n} \ln \left( \frac{A_0}{A_n} \right) \]  
\[ \Psi_{II} = \frac{1}{n-m} \ln \left( \frac{A_m}{A_n} \right) \]  

where:
- \( \Psi \) – logarithmic decrement,
- \( A_{0,n,m} \) – subsequent amplitude peak values,
- \( n, m \) – subsequent number of amplitude.

In case of welded joints, damped oscillatory motions of responses recorded by accelerometers are characterized by uneven distribution of maxima and minima. Due to variable changes of vibration speed and the need to calculate damping decrement, the possibility of approximations of response runs was checked using the functions which are generally represented by formula (7). In order to choose a proper approximating function, the comparative analysis was conducted, in which the approximations obtained by polynomials from second to fifth degree were collated with an exp function.

\[ v_1(t) = \sum_{i=0}^{N} a_i \cdot t^i \]
\[ v_2(t) = b \cdot \exp(-c \cdot t) \]  

where:
- \( v(t) \) – function approximating the course of speed,
- \( a_i \) – \( i \)-th constant for the polynomial,
- \( b, c \) – constants for the exp function.

The exemplary results of approximations for the response recorded for the sample without flaws and the sample with the flaw in the form boundary bonding are presented in Fig. 11 and 12. The tests are performed using a metal head on the modal hammer.
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By verifying the results of the approximation, it may be concluded that the second-degree polynomial is not a proper one in the test case because in regard to small amplitudes it deviates from true values of responses increasing the approximation error. However, the other functions allow determining the approximate response in a satisfactory way. At the same time, it can be noticed that increasing the degree of polynomials above 4 does not improve the approximation of received response.

Equations (5) and (6) were used to calculate the damping decrement. The following figures present the results obtained for formula (5). The first formula allows calculating the decrement distribution in regard of maximal amplitude, and the second one in regard to chosen range. For the results obtained from each approximating function, the damping decrement was calculated in regard to maximal value \(A_0\). The damping decrement obtained with approximation by exp function is characterized by linear change. Moreover, it was considered to be the reference for other results. Fig. 13 shows changes of damping decrement for the homogenous plate without welded joint along the different approximating functions. It is noticeable that the damping decrements distributions determined for the plate without the welded joint represent the constant type. It reflects the uniform damping, which is observable for plates showing characteristics of linear structure both materially and geometrically.
It is possible to observe higher, time-varying damping for material with a good weld (Fig. 14). Therefore, decrements runs are clearly curved and become concurrent with a decrement calculated from exp function. The intersection with the reference characteristic of the decrement occurs at the 41st peak amplitude of response. This effect is even more noticeable in case of the welded plate with the flaw in the form boundary bonding (Fig. 15) because the intersection occurred already at the 31st peak amplitude of the response.

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**Fig. 13.** Values of damping decrement changes for the plate without weld (0)

**Fig. 14.** Values of damping decrement changes for the plate with a weld without flaws (2202)
6. Conclusions

The presented analysis methods of the response obtained in the vibrodiagnostic testing of welded plates allow concluding that it will be possible to detect the damages of the welded joint in autonomic maritime structural health monitoring. The condition and quality of the weld joint for given plate can be obtained from diagnostic information of the mean value distribution analysis of amplitude spectrums calculated for the time window method. It is possible to use this parameter to detect some defects (cracks) in welds in systems made for autonomous SHM system. Because of similarities between presented characteristics and Poisson distribution, it is possible to introduce proposed method to evaluation of the welded joints on the basis of one parameter which would be expected value calculated for this distribution and marked as $\lambda$. Application of time windows method to evaluation of welded joints enabled the analysis of responses from accelerometers in the field of time and frequency. All differences of frequency characteristics in windows show the nonlinearity of the system and also additional dissipation of vibration energy which is a sign of construction inconsistency. Additional dissipation means flawed welded joint.

In the second presented method, for a plate made of one material and without weld, the damping decrement distribution obtained for different approximating functions is practically linear and parallel to the characteristic of the reference value. In the other cases, i.e. for plates with welded joints, the characteristics of damping decrement clearly curve and tend to intersect with the characteristic of reference decrement. The results show that the characteristics of damping decrement curve faster for the plates which weld has a flaw. Applying various functions to response approximations allowed assessing their usefulness in concluded calculations. From the comparative analysis of the results obtained for the selected functions, it is clear that the second-degree polynomial should not be used in these calculations. On the other hand, increasing the polynomial above the fourth degree also seems unreasonable as the results overlap. As the characteristics show, using a third-degree polynomial gives an incomplete result that is lower than the result obtained for the fourth-degree polynomial.

Proposed evaluation methods of welded joints require further testing of larger number of samples with various flaws in welds.
REFERENCES


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A FUZZY AHP AND ELECTRE METHOD FOR SELECTING STABILIZING DEVICE IN SHIP INDUSTRY

Summary

Roll stabilizer systems are studied for different types of ships by many researchers. It is well known that roll motion is caused by external disturbances (wave, wind etc) and large roll motion can cause capsizing easily. In addition, undesirable roll motion effect badly crew performance and passenger comfort. So, roll reduction has an important role for all types of ships. In our study, we proposed a fuzzy AHP (Analytic Hierarchy Process) and ELECTRE (Elimination and Choice Translating Reality English) method for selecting the most effective roll stabilizing system for a trawler type fishing vessel. Alternatives and criteria in relation to the stabilizers are determined by experts’ experiences and literature review. This paper intends to give a comprehensive procedure for determining the most suitable roll motion stabilizing system of trawler for safety and efficiency fishing in the open literature.

Key words: AHP; ELECTRE; fuzzy sets; roll stabilizing system; trawler

1. Introduction

Although significant advances have been made in fishing vessel, fishing still remains a highly dangerous profession. One of the fundamental means by which the operation of fishing vessel can be improved is through reduction in the roll motion. Excessive roll not only increases fuel consumption but also makes working on deck hazardous, affecting the efficiency and safety of the crew. So, roll motion stabilizing devices are very crucial in severe sea states. Many types of roll stabilizers are suggested for different ships by researchers in literature.


It is known that the purpose of roll stabilizer systems is to minimize the roll amplitudes of a ship. As can be understood from the above-mentioned studies, roll stabilizer systems for ships have been technically evaluated and the rate of reduction of the roll amplitudes has been the most important criterion. However, a technically successful roll stabilizer system may not always be feasible for a ship. Different criteria, such as the economic criterion, can create an obstacle to the selection of a stabilizer system for that ship. Therefore, in this study, considering the different criteria, the most suitable one among the four stabilizer systems for a trawler type fishing boat was determined by means of the fuzzy AHP and ELECTRE method considering the expert opinions on these criteria and alternatives. This paper aims to present and contribute a robust methodological approach utilising AHP and ELECTRE under fuzzy environment which is able to deal with uncertainty of experts’ judgement and expression in decision-making. The proposed approach is capable of for selecting stabilizing device in ship industry.

2. Research methodology

This section initially describes theoretical background of methods used in proposed approach. Then, the section shows how proposed approach is constructed.

2.1 Fuzzy sets

Fuzzy logic, introduced in 1965 by Lotfi A. Zadeh [13], is robust tool to deal with the vagueness, ambiguity and uncertainty of human judgments and assessment in making decisions process. In real world decision making problems, many decisions involve imprecision since goals, constraints, and possible actions are not known precisely [13]. Instead of combining various experiences, opinions, ideas, and motivations of an individual or group decision maker, it is better to convert the linguistic terms into fuzzy numbers. Therefore, the problems of group decision-making have necessary produced fuzzy numbers in practice. A triangular fuzzy number can be defined as a triplet $\tilde{A} = (l, m, u)$ where $l$, $m$ and $u$ denotes lower, medium and upper numbers of the fuzzy which is crisp and real numbers ($x \leq y \leq z$). In this context, Figure 1 shows a triangular fuzzy number. The membership function of a triangular fuzzy number can be defined as follows.

$$
\mu_{\tilde{A}} = \begin{cases} 
0, & x < l \\
(x - l)/(m - l), & l \leq x \leq m \\
(u - x)/(u - m), & m \leq x \leq u \\
0 & x \geq u
\end{cases}
$$

(1)
Fig. 1 Triangular fuzzy number

For any two triangular fuzzy numbers $\tilde{A}_1 = (l_1, m_1, u_1)$ and $\tilde{A}_2 = (l_2, m_2, u_2)$, the mathematical calculation of the two triangular fuzzy numbers can be defined as follows:

The addition operation between the triangular fuzzy numbers;

$$\tilde{A}_1 + \tilde{A}_2 = (l_1 + l_2, m_1 + m_2, u_1 + u_2)$$  \hspace{1cm} (2)

The subtraction operation between the triangular fuzzy numbers;

$$\tilde{A}_1 - \tilde{A}_2 = (l_1 - u_2, m_1 - m_2, u_1 - l_2)$$  \hspace{1cm} (3)

The multiplication operation between the triangular fuzzy numbers;

$$\tilde{A}_1 \times \tilde{A}_2 = (l_1 x l_2, m_1 x m_2, u_1 x u_2)$$  \hspace{1cm} (4)

The arithmetic operation for the triangular fuzzy numbers;

$$kx\tilde{A}_1 = (kx l_1, kx m_1, kx u_1), (k > 0)$$  \hspace{1cm} (5)

$$\frac{\tilde{A}_1}{k} = \left(\frac{l_1}{k}, \frac{m_1}{k}, \frac{u_1}{k}\right), (k > 0)$$  \hspace{1cm} (6)

2.2 Fuzzy AHP

AHP is a general tool for comparing a number of criteria or alternatives according to the an complete goal in a consistent manner [14,15,16]. Decision-makers generally reveal that it is more suitable to answer interval judgments than fixed-value judgements regarding to the vagueness and uncertainty from the subjective perception in the decision-making process [17,18]. This is mostly because usually he/she is unable to specific about his/her perception because of the fuzzy nature of the comparison process [19]. The assessment rate of linguistic data are measured with Triangular Fuzzy Numbers (TFNs) [20]. A TFN can be shown as (\(l|m,u\),
m[u] or (l, m, u). The parameters l, m and u, denote the smallest possible, the most promising, and the largest possible value that describes a fuzzy case, respectively. The membership function of the TFN can be specified as:

$$\mu(x/M) = \begin{cases} 
0, & x < l \\
\frac{(x-1)(m-l)}{m-l}, & 1 \leq x \leq m \\
\frac{u-x}{m-m}, & m \leq x \leq u \\
0, & x > u 
\end{cases}$$  (7)

Several methods have been proposed to address fuzzy comparison matrices. For example, Logarithmic Least Squares Method (LLSM) is proposed by Van Laarhoven and Pedrycz [21] to get triangular fuzzy weights from a triangular fuzzy comparison matrix. A modified fuzzy LLSM is presented by Wang et al. [22]. Buckley [23] employs the geometric mean method to compute fuzzy weights. Chang [24] proposes an extent analysis method, which derives crisp weights for fuzzy comparison matrices. Xu [25] brings forward a fuzzy Least Squares priority Method (LSM). A fuzzy Preference Programming Method (PPM) is also proposed by Mikhailov [26]. Lambda-Max method is proposed by Csutora and Buckley [27] which is the fuzzification of the kmax method.

We use Buckley’s Fuzzy-AHP to find importance weights since it is simple to cover to the fuzzy case and assurances a sole solution to the reciprocal comparison matrix [28]. It is rather easier than the other Fuzzy-AHP approaches. The steps of the applied Buckley’s Fuzzy-AHP algorithm can be presented as follows [23, 29]:

<table>
<thead>
<tr>
<th>Table 1 Linguistic variables for importance weights</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Linguistic variables</strong></td>
</tr>
<tr>
<td>Absolutely Strong (AS)</td>
</tr>
<tr>
<td>Very Strong (VS)</td>
</tr>
<tr>
<td>Fairly Strong (FS)</td>
</tr>
<tr>
<td>Slightly Strong (SS)</td>
</tr>
<tr>
<td>Equally (E)</td>
</tr>
<tr>
<td>Slightly Weak (SW)</td>
</tr>
<tr>
<td>Fairly Weak (FW)</td>
</tr>
<tr>
<td>Very Weak (VW)</td>
</tr>
<tr>
<td>Absolutely Weak (AW)</td>
</tr>
</tbody>
</table>

**Step 1.** Build pairwise comparison matrices among all the criteria. The linguistic variable is assigned according to the Equation (9). It is questioned which is the more important of each two criteria, such as:

$$M = \begin{pmatrix} 
1 & \tilde{a}_{12} & \cdots & \tilde{a}_{1n} \\
\tilde{a}_{21} & 1 & \cdots & \tilde{a}_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
\tilde{a}_{n1} & \tilde{a}_{n2} & \cdots & 1 
\end{pmatrix} = \begin{pmatrix} 
1 & \tilde{a}_{12} & \cdots & \tilde{a}_{1n} \\
1/\tilde{a}_{21} & 1 & \cdots & \tilde{a}_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
1/\tilde{a}_{n1} & \tilde{a}_{n2} & \cdots & 1 
\end{pmatrix}$$  (8)

where,
Step 2. Apply geometric mean to explain the fuzzy geometric mean as follows:

\[ r_i = \left( a_{i1} \otimes a_{i2} \otimes \ldots \otimes a_{in} \right)^{1/n} \]  

(10)

where \( \tilde{a}_{in} \) is fuzzy comparison value of criterion \( i \) to criterion \( n \), thus, is geometric mean of fuzzy comparison value of criterion \( i \) to each criterion.

Step 3. Compute the fuzzy weights of each criterion

\[ \tilde{w}_i = \tilde{r}_i \otimes \left( \tilde{r}_1 \oplus \tilde{r}_2 \oplus \ldots \oplus \tilde{r}_n \right)^{-1} \]  

(11)

where \( \tilde{w}_i \) is the fuzzy weight of the ith criterion, can be indicated by \( \tilde{w}_i = (l\tilde{w}_i, m\tilde{w}_i, u\tilde{w}_i) \). Here \( l\tilde{w}_i \), \( m\tilde{w}_i \), and \( u\tilde{w}_i \) represent the lower, middle and upper values of the fuzzy weight of the ith criterion.

Step 4. Apply Center of Area (COA) method to learn the Best Nonfuzzy Performance (BNP) value of each criterion by the Equation (12).

\[ BNP\tilde{w}_i = \frac{[u\tilde{w}_i - l\tilde{w}_i] + (m\tilde{w}_i - l\tilde{w}_i)]}{3 + l\tilde{w}_i} \]  

(12)

According to the BNP value for each of the alternatives, the ranking of each alternative can then continue.

2.3 Fuzzy ELECTRE

ELECTRE method is first proposed by Benayoun et al. [30]. A detailed comprehensive review of ELECTRE method is presented [31]. They presented for four classification as applied papers; survey, review and overview papers; papers on MCDA method and model selection; preference disaggregation and theoretical and non-application papers application. Fuzzy sets might provide more flexibility to show the vague/imprecise information stemming from the lack of information [32,33,34]. Assume that there is a set \( X \) of alternatives, where \( X = \{x_1, x_2, \ldots, x_n\} \) and assume that there is a set \( C \) of criteria \( C = \{c_1, c_2, \ldots, c_m\} \) and assume that there are \( k \) decision-makers \( D_1, D_2, \ldots, D_k \). Then, the steps of the proposed method are as given below.

Step 1. In a group decision environment, assume that a decision group has \( k \) decision makers, and the rating of alternatives according to each criterion can be calculated as:
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\[ Y_k = \left( \tilde{c}_{ij}^k \right)_{n \times m} = \begin{bmatrix} c_1^k & c_2^k & \cdots & c_m^k \\
\tilde{c}_{11}^k & \tilde{c}_{12}^k & \cdots & \tilde{c}_{1m}^k \\
\vdots & \vdots & \ddots & \vdots \\
\tilde{c}_{n1}^k & \tilde{c}_{n2}^k & \cdots & \tilde{c}_{nm}^k \end{bmatrix} \quad (13) \]

\[ \bar{Y} = \left( \tilde{c}_{ij} \right)_{n \times m} \]

where \( \tilde{c}_{ij} = \left( \frac{\tilde{c}_{ij}^1 + \tilde{c}_{ij}^2 + \cdots + \tilde{c}_{ij}^k}{k} \right) \), \( \tilde{c}_{ij} \) is fuzzy set \( 1 \leq i \leq m, 1 \leq j \leq n, 1 \leq p \leq k \) and \( k \) denotes the number of decision-makers. In this step, average operator is applied as aggregation operation. It is calculated the weight of each criterion by summing the assigned fuzzy sets by experts and then dividing the sum by the number of experts [35,36].

### Table 2 Linguistic variables for alternative ratings

<table>
<thead>
<tr>
<th>Linguistic variables</th>
<th>Fuzzy numbers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very Poor (VP)</td>
<td>(0, 0, 1)</td>
</tr>
<tr>
<td>Poor (P)</td>
<td>(0, 1, 3)</td>
</tr>
<tr>
<td>Medium Poor (MP)</td>
<td>(1, 3, 5)</td>
</tr>
<tr>
<td>Fair (F)</td>
<td>(3, 5, 7)</td>
</tr>
<tr>
<td>Medium Good (MG)</td>
<td>(5, 7, 9)</td>
</tr>
<tr>
<td>Good (G)</td>
<td>(7, 9, 10)</td>
</tr>
<tr>
<td>Very good (VG)</td>
<td>(9, 10, 10)</td>
</tr>
</tbody>
</table>

**Step 2.** Construct the weighting matrix \( W_i \) of the criteria of the \( k \)th decision-maker and build the average weighting matrix \( W \), respectively, shown as follows:

\[ W_k = \left( \tilde{w}_j^k \right)_{1 \times m} = \begin{bmatrix} \tilde{w}_1^k & \tilde{w}_2^k & \cdots & \tilde{w}_m^k \end{bmatrix} \quad (15) \]

\[ \bar{W} = \left( \tilde{w}_j \right)_{1 \times m} \]

where \( \tilde{w}_j = \frac{\tilde{w}_j^1 + \tilde{w}_j^2 + \cdots + \tilde{w}_j^k}{k} \), \( \tilde{w}_j \) is an fuzzy set \( 1 \leq i \leq m, 1 \leq p \leq k \) and \( k \) denotes the number of decision-makers.

**Step 3.** Given \( \tilde{c}_{ij} = \left( c_{ij1}, c_{ij2}, c_{ij3} \right) \); the normalized performance rating for beneficial criterion can be calculated as:

\[ \tilde{n}_j = \begin{bmatrix} \frac{c_{ij1}}{c^*}, \frac{c_{ij2}}{c^*}, \frac{c_{ij3}}{c^*} \end{bmatrix} \quad (17) \]

Where \( c^* = \text{Max } c_{ij} \)
Step 4. Formulate the weighted decision matrix.

\[
\vec{y}_w = \left( \tilde{v}_j \right)_{nm} = \begin{bmatrix} c_1 \tilde{v}_{i1}^k & c_2 \tilde{v}_{i2}^k & \cdots & c_m \tilde{v}_{im}^k \\ x_1 \tilde{v}_{11}^k & \tilde{v}_{12}^k & \cdots & \tilde{v}_{1m}^k \\ x_2 \tilde{v}_{21}^k & \tilde{v}_{22}^k & \cdots & \tilde{v}_{2m}^k \\ \vdots & \vdots & \ddots & \vdots \\ x_n \tilde{v}_{n1}^k & \tilde{v}_{n2}^k & \cdots & \tilde{v}_{nm}^k \end{bmatrix}
\]  

(18)

Where \( v_{ij} = w_i \otimes n_{ij} \), \( 1 \leq i \leq m \), and \( 1 \leq j \leq n \).

Step 5. Specify concordance and discordance fuzzy sets for each fuzzy pairs of \( k \) and \( l \) alternatives \( k, l = 1, 2, \ldots, n; l \neq k \). The set of fuzzy indicators \( J = \{ j | j = 1, 2, \ldots, n \} \) is divided into two different sets as concordance \( \tilde{S}_{kl} \) and discordance fuzzy set \( \tilde{D}_{kl} \)

\[
\tilde{S}_{kl} = \left\{ j \left[ v_{ikj}, v_{2kj}, v_{3kj} \right] \geq \left[ v_{ij}, v_{2ij}, v_{3ij} \right] \right\}
\]  

(19)

Vice versa the complementary subset named discordance set is a set of indicators that for each of them:

\[
\tilde{D}_{kl} = \left\{ j \left[ v_{ikj}, v_{2kj}, v_{3kj} \right] < \left[ v_{ij}, v_{2ij}, v_{3ij} \right] \right\}
\]  

(20)

Step 6. Compute the concordance fuzzy matrix.

Concordance fuzzy index is equal to the sum of fuzzy weights \( \tilde{W} = (\tilde{w}_i)_{nm} \) for those indices which form the set. Thus, concordance fuzzy index \( \tilde{I}_{k,j} \) between \( x_k \) and \( x_j \) is as follows:

\[
\tilde{I}_{k,j} = \sum_{j \in \tilde{S}_{kl}} \tilde{W} \sqrt{b^2 - 4ac}
\]  

(21)

The higher value of \( \tilde{I}_{k,j} \) presents both the superiority and concordance of \( x_k \) to \( x_j \). The asymmetrical concordance fuzzy matrix \( (\tilde{I}_{i,j}) \) as follows:

\[
\tilde{I} = \begin{bmatrix} - & \tilde{I}_{1,2} & \cdots & \tilde{I}_{1,n} \\ \tilde{I}_{2,1} & - & \cdots & \tilde{I}_{2,n} \\ \vdots & \vdots & \ddots & \vdots \\ \tilde{I}_{n,1} & \cdots & \tilde{I}_{n,(n-1)} & - \end{bmatrix}
\]  

(22)

Step 7. Calculate the discordance fuzzy matrix.
Discordance fuzzy index in contrast to the index $\bar{N}_{I,k,l}$ shows that $x_i$ is strongly superior according to $x_j$. The index $\bar{N}_{I,k,l}$ is computed using the members of matrix $\bar{Y}_w$ for each element of discordance fuzzy set as follows:

$$\bar{N}_{I,k,l} = \text{defuzzified} \left( \sum_{j \in D} |\bar{v}_k - \bar{v}_j| \right)$$  \hspace{1cm} (23)

Discordance fuzzy matrix for all pair wise comparisons of alternatives converts into a matrix with exact numbers which is:

$$\bar{N} = \begin{bmatrix} - & \bar{N}_{I,1,2} & \bar{N}_{I,1,3} & \ldots & \bar{N}_{I,1,n} \\ N_{I,2,1} & - & \ldots & \ldots & \ldots \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ \bar{N}_{I,n,1} & \bar{N}_{I,n,2} & \ldots & \bar{N}_{I,n,(n-1)} & - \end{bmatrix}$$  \hspace{1cm} (24)

These have subsidiary relationship such that fuzzy matrix $I$ is descriptive of the weights resulted from concordance indices, and asymmetrical matrix $\bar{N}$ reflects the high relative difference of for each discordance indices.

**Step 8.** Specify the effective concordance fuzzy matrix.

The values of indices $\bar{I}_{k,j}$ of concordance fuzzy matrix should be compared against a threshold value so that the superiority chance of $x_i$ according to $x_j$ is better judged. In the case when $\bar{I}_{k,j}$ exceeds from a minimum threshold $\bar{T}$ this chance increases.

Also, we can compute the average of each arbitrary fuzzy index $\bar{T}$ from concordance fuzzy indices in the following manner:

$$\bar{T} = \frac{1}{n(n-1)} \sum_{k=1}^{n} \sum_{j=1}^{n} \bar{I}_{k,j}$$  \hspace{1cm} (25)

A Boolean matrix $F$ is constructed based upon minimum threshold $\bar{T}$ which has elements 0 and 1 as:

$$f_{ij} = \begin{cases} 1 & \text{if } \bar{I}_{k,j} \geq \bar{T} \\ 0 & \text{if } \bar{I}_{k,j} < \bar{T} \end{cases}$$  \hspace{1cm} (26)

An effective and dominant alternative against the other alternatives is obtained with respect to the each element $I$ in matrix $F$ (effective concordance fuzzy matrix).

**Step 9.** Specify the effective discordance fuzzy matrix.

Elements $\bar{N}_{I,k,l}$ from discordance matrix that is provided in Step 6 is assessed according to a threshold value. This threshold value ($\bar{N}$) is calculated with the following formula.
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\[ \bar{N}I = \frac{1}{n(n-1)} \sum_{i=1}^{n} \sum_{j=1}^{n} \left[ N_{i,j} \right] I \]

(27)

A Boolean matrix \( G \) (effective discordance matrix) is then built as:

\[
\begin{align*}
g_{ij} &= 1 \text{ if } \bar{N}I_{i,j} \leq \bar{N}I \\
g_{ij} &= 0 \text{ if } \bar{N}I_{i,j} > \bar{N}I
\end{align*}
\]

(28)

Dominance relations among alternatives is obtained to unit elements in matrix.

**Step 10.** Specify effective and outranking matrix.

Common elements \( (h_{i,j}) \) is obtained outranking matrix \( (H) \) for making decision from matrices \( F \) and \( G \).

\[ h_{i,j} = f_{i,j} \otimes g_{i,j} \]

(29)

**Step 11.** Eliminate the less attractive alternatives.

The order of relative superiority of alternatives is presented by Outranking matrix \( (H) \). If \( h_{i,j} = 1 \), \( x_i \) is superior to \( x_j \) in terms of both concordance and discordance indices. However, \( x_i \) might be still dominated by other alternatives. Therefore, the condition which makes \( x_i \) an effective alternative is as follows:

\[
\begin{align*}
h_{i,j} &= 1 \text{ for at least one unit element for } l = 1,2,...,n; k \neq l \\
h_{i,j} &= 0 \text{ for all } i \text{ for } l = 1,2,...,n; i \neq k; i \neq l
\end{align*}
\]

(30)

In the cases where these two conditions are not simultaneously fulfilled, the effective alternatives from matrix \( (H) \) can be simply recognized. Then, we can exclude those columns of \( (H) \) which at least have a unit element (1) from matrix \( (H) \) because those columns are dominated by other row or rows. It is that the threshold values of \( \bar{T} \) and \( \bar{N}I \) which are showed in steps 8 and 9 are approximate and used to enable generating a criterion to select the best alternative between all alternatives. As long as Eq. (30) is not true for any of the alternatives, we can increase \( \bar{T} \) and reduce \( \bar{N}I \) until the above condition is satisfied to come up with the best alternative.

3 Application

Roll stabilization systems have been the subject of scientific investigation for many years. It is well known that the rolling motion of a ship is an undesirable feature of its behaviour. The variety of stabilizers have been proposed and installed successfully for the elimination or moderation of ship roll. Recently, interest has centered on the stabilizer types, which is better suited to which type of ship. This paper undertakes a review of the whole field of stabilization devices and proposes selection procedure of the most suitable roll motion stabilization system for a trawler type fishing vessel. Looked at in this way, the four major stabilizers: activated fins, anti-rolling tanks, bilge keel, rudder roll stabilization devices are
taken up one by one, evaluated and discussed in detail. Various criteria for determining the most suitable roll motion stabilizer for a trawler type of fishing vessel are compared. Stabilizers are classified and their relative merits discussed. The general approach developed in this paper is applied for a trawler type fishing vessel as an example.

The hierarchical structure adopted in this study to deal with the problems of selection of the roll motion stabilizing system for a trawler type fishing vessel is shown in Fig 2.

The key dimensions of the criteria for evaluation and selection of the roll motion stabilizing system for a trawler type fishing vessel were derived through comprehensive investigation and consultation with five experts, including two professor in the department of Naval Architecture and Marine Engineering. They were asked to rate the accuracy, adequacy and relevance of the criteria and dimensions and to verify their “content validity” in terms of the stabilizer assessment. Twelve types of criteria of high priority come forth when these criteria are examined. Criteria are coded as $C_i$, where $i$ is the number of relevant criteria as below in Table 3.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Criteria</th>
<th>Symbol</th>
<th>Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>Total Initial Costs</td>
<td>C7</td>
<td>Underwater Noise</td>
</tr>
<tr>
<td>C2</td>
<td>Cargo Carrying Capability</td>
<td>C8</td>
<td>Expensive Pieces Of Equipment</td>
</tr>
<tr>
<td>C3</td>
<td>Crew Performance And/Or Passenger Comfort</td>
<td>C9</td>
<td>Working on Low Speed Range</td>
</tr>
<tr>
<td>C4</td>
<td>Influence On Speed, Power And Resistance</td>
<td>C10</td>
<td>Working on High Speed Range</td>
</tr>
<tr>
<td>C5</td>
<td>Maintenance Requirements</td>
<td>C11</td>
<td>Motion Limitations</td>
</tr>
<tr>
<td>C6</td>
<td>Roll Reduction</td>
<td>C12</td>
<td>Wave Conditions</td>
</tr>
</tbody>
</table>

When aforementioned criteria which differ from one another on the basis of basic characteristics are examined with the intention of categorizing, it appears that each has a relationship with different stabilizer systems. It is also known that criteria in certain experts develop a relationship along with the ones in other experts. As far as factors for criteria are concerned, stabilizer systems in connection with the criteria can be grouped in Table 4.
According to Table 5 the fuzzy linguistic variables of criteria are shown. In this step, the fuzzy importance weights of criteria are determined using fuzzy AHP. The importance of each criterion is evaluated by experts. The linguistic variables are convert to the fuzzy sets using Table 1 and the aggregated fuzzy pairwise comparison is presented in Table 6. Finally, the fuzzy weights of the each criterion are calculated and it is presented in Table 7.

Table 5 The pairwise comparisons of criteria

<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>…</th>
<th>C11</th>
<th>C12</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(E, E, E, E)</td>
<td>(SS, SW, E, SS, E)</td>
<td>...</td>
<td>(SS, E, FS, VS, VS)</td>
<td>(FS, SW, VS, SW, FS)</td>
</tr>
<tr>
<td>C2</td>
<td>(SW, SS, E, SW, E)</td>
<td>(E, E, E, E)</td>
<td>...</td>
<td>(SS, E, FS, FS, VS)</td>
<td>(SS, E, VS, FW, FS)</td>
</tr>
<tr>
<td>C3</td>
<td>(VW, E, SW, SW)</td>
<td>(SW, SW, SW, E, SW)</td>
<td>...</td>
<td>(SW, E, E, FS, FS)</td>
<td>(E, SW, FS, FW, SS)</td>
</tr>
<tr>
<td>C4</td>
<td>(SS, FW, SW, E, SW)</td>
<td>(SS, VW, SW, SS, SW)</td>
<td>...</td>
<td>(SS, FW, E, VS, FS)</td>
<td>(FS, VW, FS, SW, SS)</td>
</tr>
<tr>
<td>C5</td>
<td>(E, SW, FW, FW, WS)</td>
<td>(SS, FW, FW, SW, FW)</td>
<td>...</td>
<td>(SS, SW, SW, SS, SS)</td>
<td>(FS, FW, SS, VW, E)</td>
</tr>
<tr>
<td>C6</td>
<td>(FW, SS, VW, FW, SW)</td>
<td>(SW, E, VW, SW, VW)</td>
<td>...</td>
<td>(SW, SS, FW, SS, E)</td>
<td>(E, E, E, VW, SW)</td>
</tr>
<tr>
<td>C7</td>
<td>(SW, VW, AW, VW, FW)</td>
<td>(E, AW, AW, FW, AW)</td>
<td>...</td>
<td>(E, VW, VW, E, SW)</td>
<td>(SS, AW, SW, AW)</td>
</tr>
<tr>
<td>C8</td>
<td>(SW, SW, SW, E, E)</td>
<td>(E, FW, SW, SS, E)</td>
<td>...</td>
<td>(E, SW, E, VS, VS)</td>
<td>(FS, FW, FS, SW, FS)</td>
</tr>
<tr>
<td>C9</td>
<td>(AW, VW, AW, VW, FW)</td>
<td>(FW, AW, AW, FW, AW)</td>
<td>...</td>
<td>(FW, FW, VW, E)</td>
<td>(SW, AW, SW, AW)</td>
</tr>
<tr>
<td>C10</td>
<td>(AW, FW, SW, FW)</td>
<td>(FW, FW, FW, FW, E)</td>
<td>...</td>
<td>(FW, FW, FS, WS, E)</td>
<td>(SW, FW, SS, FW, FS)</td>
</tr>
<tr>
<td>C11</td>
<td>(SW, E, FW, FW, VW)</td>
<td>(E, FW, FW, FW, VW)</td>
<td>...</td>
<td>(E, E, E, E)</td>
<td>(SS, SW, FS, AW)</td>
</tr>
<tr>
<td>C12</td>
<td>(FW, SS, VW, SS, FW)</td>
<td>(E, VW, FW, FS, FW)</td>
<td>...</td>
<td>(SW, SS, FW, AS, SS)</td>
<td>(E, E, E, E)</td>
</tr>
</tbody>
</table>

Table 6 The fuzzy weights of each criterion

<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>…</th>
<th>C11</th>
<th>C12</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(1, 1, 3)</td>
<td>(0.84, 1.67, 3.4)</td>
<td>...</td>
<td>(3.46, 6.6)</td>
<td>(2.28, 3.53, 5)</td>
</tr>
<tr>
<td>C2</td>
<td>(0.68, 1.13, 2.6)</td>
<td>(1, 1, 3)</td>
<td>...</td>
<td>(1, 1, 3)</td>
<td>(1.26, 3.24, 4.87)</td>
</tr>
<tr>
<td>C3</td>
<td>(0.34, 0.43, 1.24)</td>
<td>(0.36, 0.47, 1.4)</td>
<td>...</td>
<td>(0.36, 0.47, 1.4)</td>
<td>(1.07, 1.91, 2.27)</td>
</tr>
<tr>
<td>C4</td>
<td>(0.51, 0.97, 2.07)</td>
<td>(0.5, 1.36, 2.44)</td>
<td>...</td>
<td>(0.5, 1.36, 2.44)</td>
<td>(1.46, 2.7, 4.04)</td>
</tr>
<tr>
<td>C5</td>
<td>(0.33, 0.39, 1)</td>
<td>(0.33, 0.79, 1.4)</td>
<td>...</td>
<td>(0.33, 0.79, 1.4)</td>
<td>(1.05, 1.87, 3.11)</td>
</tr>
<tr>
<td>C6</td>
<td>(0.3, 0.74, 1.21)</td>
<td>(0.32, 0.39, 1.08)</td>
<td>...</td>
<td>(0.32, 0.39, 1.08)</td>
<td>(0.66, 0.7, 2.04)</td>
</tr>
<tr>
<td>C7</td>
<td>(0.13, 0.17, 0.34)</td>
<td>(0.3, 0.31, 0.75)</td>
<td>...</td>
<td>(0.3, 0.31, 0.75)</td>
<td>(0.31, 0.75, 1.32)</td>
</tr>
<tr>
<td>C8</td>
<td>(0.52, 0.6, 1.8)</td>
<td>(0.67, 1.11, 2.47)</td>
<td>...</td>
<td>(0.67, 1.11, 2.47)</td>
<td>(1.47, 2.71, 4.07)</td>
</tr>
<tr>
<td>C9</td>
<td>(0.11, 0.12, 0.17)</td>
<td>(0.12, 0.14, 0.19)</td>
<td>...</td>
<td>(0.12, 0.14, 0.19)</td>
<td>(0.15, 0.22, 0.52)</td>
</tr>
<tr>
<td>C10</td>
<td>(0.15, 0.22, 0.54)</td>
<td>(0.3, 0.34, 0.81)</td>
<td>...</td>
<td>(0.3, 0.34, 0.81)</td>
<td>(0.34, 0.81, 1.53)</td>
</tr>
<tr>
<td>C11</td>
<td>(0.31, 0.36, 0.95)</td>
<td>(0.32, 0.38, 0.97)</td>
<td>...</td>
<td>(0.32, 0.38, 0.97)</td>
<td>(0.9, 1.76, 2.83)</td>
</tr>
<tr>
<td>C12</td>
<td>(0.48, 1.31, 2.17)</td>
<td>(0.89, 1.34, 2.31)</td>
<td>...</td>
<td>(0.89, 1.34, 2.31)</td>
<td>(1, 1, 3)</td>
</tr>
</tbody>
</table>
Table 7 The fuzzy weights of criteria

<table>
<thead>
<tr>
<th>C1</th>
<th>The fuzzy geometric means</th>
<th>The fuzzy weights</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(21.5, 34.5, 54.1)</td>
<td>(0.06, 0.18, 0.46)</td>
</tr>
<tr>
<td>C2</td>
<td>(1.97, 3.13, 5.02)</td>
<td>(0.06, 0.16, 0.42)</td>
</tr>
<tr>
<td>C3</td>
<td>(1.1, 1.76, 3.39)</td>
<td>(0.03, 0.09, 0.28)</td>
</tr>
<tr>
<td>C4</td>
<td>(1.43, 2.41, 4.09)</td>
<td>(0.04, 0.13, 0.34)</td>
</tr>
<tr>
<td>C5</td>
<td>(0.9, 1.56, 2.85)</td>
<td>(0.03, 0.08, 0.24)</td>
</tr>
<tr>
<td>C6</td>
<td>(0.81, 1.29, 2.42)</td>
<td>(0.02, 0.07, 0.2)</td>
</tr>
<tr>
<td>C7</td>
<td>(0.36, 0.53, 1.14)</td>
<td>(0.01, 0.03, 0.1)</td>
</tr>
<tr>
<td>C8</td>
<td>(1.52, 2.39, 4.22)</td>
<td>(0.05, 0.13, 0.35)</td>
</tr>
<tr>
<td>C9</td>
<td>(0.22, 0.27, 0.58)</td>
<td>(0.01, 0.01, 0.05)</td>
</tr>
<tr>
<td>C10</td>
<td>(0.56, 0.9, 1.9)</td>
<td>(0.02, 0.05, 0.16)</td>
</tr>
<tr>
<td>C11</td>
<td>(0.89, 1.38, 2.69)</td>
<td>(0.03, 0.07, 0.23)</td>
</tr>
<tr>
<td>C12</td>
<td>(1.43, 2.34, 3.72)</td>
<td>(0.04, 0.12, 0.31)</td>
</tr>
</tbody>
</table>

The ratings of stabilizer systems according to criteria are evaluated by five experts, who are working in ship sector as managers and instructors are presented in Table 8. Table 2 is used for conversion of evaluations into fuzzy numbers. Table 9 presents the aggregated judgment of the experts. The normalized decision matrix is obtained and it is shown in Table 10.

Table 8 The comparison of stabilizer systems according to criteria

<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>C3</th>
<th>C4</th>
</tr>
</thead>
<tbody>
<tr>
<td>C5</td>
<td>C6</td>
<td>C7</td>
<td>C8</td>
<td></td>
</tr>
<tr>
<td>A1</td>
<td>(VG, VG, VG, VG, G)</td>
<td>(F, VP, VP, MP)</td>
<td>(VG, VG, VG, VG, VG)</td>
<td>(VG, G, G, VG, VG)</td>
</tr>
<tr>
<td>C9</td>
<td>C10</td>
<td>C11</td>
<td>C12</td>
<td></td>
</tr>
</tbody>
</table>

Table 9. The aggregated fuzzy comparison of stabilizer systems

<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>C3</th>
<th>C4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>(7.8, 9.4, 10)</td>
<td>(5.4, 7.4, 9)</td>
<td>(1.8, 3.8, 5.8)</td>
<td>(5.8, 7.6, 9)</td>
</tr>
<tr>
<td>A2</td>
<td>(3.4, 5.4, 7.4)</td>
<td>(1.4, 2.6, 4.2)</td>
<td>(4.8, 6.6, 8)</td>
<td>(3.2, 5.6, 8)</td>
</tr>
<tr>
<td>A3</td>
<td>(2.2, 4.2, 6.2)</td>
<td>(5.8, 7.6, 9.2)</td>
<td>(4.6, 6.6, 8.4)</td>
<td>(4.2, 6.7, 6)</td>
</tr>
<tr>
<td>A4</td>
<td>(0.4, 1.2, 2.6)</td>
<td>(5.4, 7.2, 8.6)</td>
<td>(5.4, 7.2, 8.6)</td>
<td>(3.4, 5.4, 7.4)</td>
</tr>
<tr>
<td>C5</td>
<td>C6</td>
<td>C7</td>
<td>C8</td>
<td></td>
</tr>
<tr>
<td>A1</td>
<td>(8.6, 9.8, 10)</td>
<td>(2.2, 3.4, 4.8)</td>
<td>(9, 10, 10)</td>
<td>(8.2, 9.6, 10)</td>
</tr>
<tr>
<td>A2</td>
<td>(3.4, 5.4, 7.4)</td>
<td>(5.4, 7.4, 8.8)</td>
<td>(5.4, 7.2, 8.6)</td>
<td>(4.6, 6.2, 7.4)</td>
</tr>
<tr>
<td>A3</td>
<td>(1.8, 3.8, 5.8)</td>
<td>(4.2, 6.7, 6.2)</td>
<td>(3, 5, 7)</td>
<td>(1.6, 3.2, 5)</td>
</tr>
<tr>
<td>A4</td>
<td>(1.4, 3.4, 5.4)</td>
<td>(4.2, 6.2, 8.2)</td>
<td>(2.6, 4.6, 6.6)</td>
<td>(0.8, 1.8, 3.4)</td>
</tr>
<tr>
<td>C9</td>
<td>C10</td>
<td>C11</td>
<td>C12</td>
<td></td>
</tr>
<tr>
<td>A1</td>
<td>(4.8, 6.4, 8)</td>
<td>(4.4, 6.7, 4)</td>
<td>(0, 2, 0.6, 1.8)</td>
<td>(3.8, 5.6, 7.2)</td>
</tr>
<tr>
<td>A2</td>
<td>(5.6, 7.4, 8.6)</td>
<td>(4.8, 6.6, 8)</td>
<td>(1.2, 2.3, 4)</td>
<td>(7.8, 9.4)</td>
</tr>
<tr>
<td>A3</td>
<td>(4.4, 6.2, 8)</td>
<td>(4.8, 6.6, 8.2)</td>
<td>(7.4, 9.2, 10)</td>
<td>(5.8, 7.8, 9.4)</td>
</tr>
<tr>
<td>A4</td>
<td>(3.2, 5.7)</td>
<td>(4, 5.8, 7.6)</td>
<td>(6.6, 8.4, 9.6)</td>
<td>(4.2, 6.7, 6)</td>
</tr>
</tbody>
</table>
Table 10 The normalized fuzzy comparison of stabilizer systems

<table>
<thead>
<tr>
<th></th>
<th>C1</th>
<th>C2</th>
<th>C3</th>
<th>C4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>(0.78, 0.94, 1)</td>
<td>(0.59, 0.8, 0.98)</td>
<td>(0.21, 0.44, 0.67)</td>
<td>(0.64, 0.84, 1)</td>
</tr>
<tr>
<td>A2</td>
<td>(0.34, 0.54, 0.74)</td>
<td>(0.15, 0.28, 0.46)</td>
<td>(0.56, 0.77, 0.93)</td>
<td>(0.36, 0.56, 0.76)</td>
</tr>
<tr>
<td>A3</td>
<td>(0.22, 0.42, 0.62)</td>
<td>(0.63, 0.83, 1)</td>
<td>(0.53, 0.77, 0.98)</td>
<td>(0.47, 0.67, 0.84)</td>
</tr>
<tr>
<td>A4</td>
<td>(0.04, 0.12, 0.26)</td>
<td>(0.59, 0.78, 0.93)</td>
<td>(0.63, 0.84, 1)</td>
<td>(0.38, 0.6, 0.82)</td>
</tr>
</tbody>
</table>

The concordance and discordance fuzzy sets are specified. The fuzzy concordance matrix is calculated and it is presented in Table 11. The fuzzy discordance matrix is calculated. Then, the fuzzy discordance matrix is defuzzified using center of area defuzzification method and the results are presented in Table 12.

Table 11 The fuzzy concordance matrix

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A3</th>
<th>A4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>-</td>
<td>(0.247, 0.706, 1.912)</td>
<td>(0.195, 0.556, 1.489)</td>
<td>(0.211, 0.767, 1.96)</td>
</tr>
<tr>
<td>A2</td>
<td>(0.149, 0.417, 1.235)</td>
<td>-</td>
<td>(0.251, 0.62, 1.397)</td>
<td>(0.236, 0.667, 1.869)</td>
</tr>
<tr>
<td>A3</td>
<td>(0.201, 0.567, 1.609)</td>
<td>(0.127, 0.363, 1.437)</td>
<td>-</td>
<td>(0.339, 0.963, 2.658)</td>
</tr>
<tr>
<td>A4</td>
<td>(0.125, 0.356, 1.186)</td>
<td>(0.16, 0.456, 1.277)</td>
<td>(0.033, 0.16, 0.488)</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 12 The defuzzified discordance matrix

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A3</th>
<th>A4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>-</td>
<td>0.8140</td>
<td>1.0000</td>
<td>0.9402</td>
</tr>
<tr>
<td>A2</td>
<td>1.0000</td>
<td>-</td>
<td>1.0000</td>
<td>1.0000</td>
</tr>
<tr>
<td>A3</td>
<td>0.7583</td>
<td>0.4200</td>
<td>-</td>
<td>0.2750</td>
</tr>
<tr>
<td>A4</td>
<td>0.9829</td>
<td>0.7222</td>
<td>1.0000</td>
<td>-</td>
</tr>
</tbody>
</table>

The effective concordance matrix is specified (Table 13). The effective discordance matrix is obtained (Table 14). Next, we construct the effective and outranking matrix presented in Table 15 by multiplying the effective concordance and discordance level matrices to disregard the effects of the Boolean matrices, separately. The less attractive alternatives are eliminated.
Table 13 The effective concordance level

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A3</th>
<th>A4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>A2</td>
<td>0</td>
<td>0</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>A3</td>
<td>1</td>
<td>0</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>A4</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 14 The effective discordance level

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A3</th>
<th>A4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>A2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>A3</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>A4</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 15 The global matrix

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A3</th>
<th>A4</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>0</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>A2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>A3</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>A4</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Finally, we construct the decision graph that is presented in Fig. 3. This decision graph which is derived from a great deal of imprecise data shows the preferable, incomparable or indifferent action. We obtain the priority sequence of stabilizer systems are as A3>A1 and A4. A1>A2 and there is no compare between A2 and A4 also A1 and A4 according to matrix $H$. Thus, it is clear that the most suitable stabilizer system A3 (Activated Fins). In this study, the most suitable stabilizer system is determined by considering twelve different criteria.
4 Conclusion

Stabilizer systems are used in a wide variety of applications for many years. Recently, the field of stabilizing systems in shipping industry has received the attention of many researchers. It is essentially important the determining of effective stabilizer for ships that serving a specific area. As mentioned above, each area has its own specific requirements. This paper presents the selection procedures of the most effective roll stabilization system for trawler type fishing vessel. Tanks, bilge keels, activated fins and rudder roll stabilization system are examined taking into account their advantages and disadvantages for trawler.

In this study, all criteria determined for roll motion stabilizing systems of trawler are evaluated by experts’ and literature review.

The fuzzy AHP and ELECTRE method is proposed to select the better effective roll motion stabilizing system for trawler ship industry. Since the proposed methodology has the ability of taking care of all kinds of evaluations from experts, it has been successfully applied to a stabilizer selection for trawler ship industry. In application case, four stabilizer system alternatives are compared for determining the most effective roll motion system of trawler ship industry. The ranking of these alternatives has been obtained as A3>A1 and A4. A1>A2 and there is no compare between A2 and A4 also A1 and A4.

Also, the alternative A3(Activated Fins) is always determined as the best alternative as a result of sensitivity analysis. In addition, the fuzzy AHP analysis determined the best criteria as “C1” for the selection process and led to the following ranking of the evaluation criteria: \{C1 (15 %), C2 (13.8 %), C8 (11.2 %), C4 (11 %), C12 (10.2 %), C3 (8.8 %), C5 (7.5 %), C11(7 %), C6(6.3 %), C10 (4.8 %), C7 (2.9 %), C9 (1.5 %)\}. For further research, some other decision making approach such as TOPSIS, VIKOR, Choquet Integral, under fuzzy environment can be on the similar problem and the obtained results can be compared.

REFERENCES


The naval sector of new construction is characterized by high cost programmes and a low degree of definition in the stages of tendering. With these requirements, an initial, highly-competitive budget should be developed with acceptable risk. For this reason, it is usual in this field for a budget Contingency Reserve to be defined, in order to cover probable range increments, as well as probable deviations that could trigger economic losses. The budget Contingency Reserve estimate is usually carried out based on shipyard experience. However, problems arise when the shipyard does not have enough experience in the concrete type of vessel to be built. In this research, the use of an extension of the triangular Monte Carlo distribution model is proposed, with the aim of calculating the likelihood of complying with the calculated budget. From this result, a Contingency Reserve that provides enough security to execute the project within the limits of the economic risk defined by the organization can be calculated. The proposal introduced in this study allows managers to obtain a more optimal estimate of the Contingency Reserve, therefore reducing economic risks.

**Key words:** uncertainty; Budget Contingency Reserve; Monte Carlo simulation, Project Management; shipbuilding projects.

1. Introduction

In shipyards dedicated to new construction, for example, the production of merchant or military vessels and offshore structures, it is common to find difficulties when elaborating budgets, which is mainly due to two factors. The first one is the need for adjustment to current market prices, a necessity that arises in a highly globalised sector and with great competition from Asian countries [1]. The second factor is that generated by the uncertainty that emerges in naval projects. This is due to the fact that shipyards build a limited number of units per series, making it difficult to apply mass production techniques. Therefore, it is difficult to optimize series production lines when only 2 or 3 units of a vessel are produced. In addition, it is common to apply modifications among the different ships in the series, on account of technological changes and the introduction of improvements from one vessel to the next in the
series. This increases the problem of using production techniques from other sectors, such as those of automobile and aerospace. It is therefore advisable to work under the concept of independent unit construction or prototypes [2]. In this respect, the research of Kolić, Lee and Fafandjel [3] identified the necessity of mitigate risks in large engineering construction project (LECP), especially in the case of prototype vessels, using Monte Carlo simulation to execute a contract risk analysis so that shipyard management could choose the contracting model with the least amount of risk.

Shipyards have to take their historical data into account, which are very useful for the generation of production simulation models [4], in order to be more efficient and competitive. Similarly, an alignment between design and manufacturing must be sought [5]. The studies of Song, Woo and Shin [6] on the systematisation and progress of shipbuilding production management for a flexible and agile response, reducing time and cost, show the importance of always maintaining the idea of the shipyard as a company in terms of the fulfilment of the budget calculated for the project.

The agreed on budget in the bidding process will not only have an impact on the production of the vessels, but will also be relevant for the life cycle of the project and the need to extend the lifespan of the ships [7]. At the same time, these budgets will be crucial in order to define the company's competitiveness [8] and the risks they are able to take in reducing the sale price of their ships [9].

It is well known that the design of a ship evolves during its construction and undergoes multiple modifications, as a consequence of the spiral design itself and the improvements applied in the successive ships in the series [10]. But this circumstance is difficult to translate to the project budget, since the budget will evolve and will be revised throughout the life of the project [11]. However, the total amount agreed on in the contract between the manufacturing shipyard and the ship operator will always be maintained, with different management tools put into practice in order to control the cost and the execution time of the shipbuilding projects [12].

Following the model developed by the Project Management Institute (PMI) [13], the project budget is made up of two large components: the cost baseline or control accounts and the management reserve. In turn, the cost baseline is divided into the Contingency Reserve and the work package cost estimates.

Regarding the project budget stipulated in the contract, it becomes key, due to the nature of the bidding process, to develop a calculation as reliable as possible of the total amount at an early stage [14]. These international biddings have turned into very disputed processes, where the different shipyards present their offers with adjusted term and cost offers [15].

Due to the uncertainty of most projects, it is necessary to endow the budget with the relevant entry of Contingency Reserve to bear the impact of cost deviations. These variations are usually produced by range variations, changes in timetable rates, modifications of the suppliers’ prices [16], etc. Contingency Reserves are often viewed as the part of the budget intended to address the “known unknowns” that can affect a project [13].

It is common for companies to calculate management reserves by means of fixed percentages without entering into detailed studies, due to the lack of means to carry out better approximations. That is why different authors have focused their attention on the subject and have tried to develop models capable of covering this need [17].

In the specific case of shipbuilding, being able to estimate the best possible values of the Contingency Reserve is clearly necessary, due to the high cost of the projects to be developed. Therefore, great interest in the definition of calculation models which will be as reliable as possible for the definition of these values, has been expressed. In this sense, there
are several deterministic, probabilistic and modern mathematical methods for obtaining the value of the Contingency Reserve, such as the Fuzzy Techniques and Artificial Neural Network [18]. This research is focused on probabilistic methods, and more specifically, on the methods based on Monte Carlo simulations.

In these types of probabilistic techniques, it is very important to study different probabilistic distributions, because depending on the chosen option, the results may vary. Therefore, it is necessary to know how to make the most suitable choice, depending on the characteristics of the project being analysed. In this regard, several authors have compared different probabilistic distributions with the intention of optimising the results obtained [19, 20]. This research will carry out a study in this line, with the aim of contrasting the results for the prediction of the management reserve, with two different probability distributions for the Monte Carlo method. Another area this type of methodology could be used in is risk analysis. In fact, although in this research the methods developed will only be presented for cost estimation, they could also be developed for the field of risk analysis, as noted in recent publications [21, 22]. Similarly, these methodologies are also employed in areas such as the cost estimate for decommissioning [23] or in the search for optimal maritime harbour design and improvement [24]. The use of the Monte Carlo model can also be implemented in design, prediction and probabilistic analysis tools, such as, for example, those used in collision and grounding analysis [25], in the study of reliability-based inspection planning [26], or in economic modelling, in order to reduce costs at different stages of the project, such as reducing the cost of ballast tank corrosion [27]. All these examples show that the tool and methodology presented in this research are of great applicability in the naval sector.

In order to lessen project control uncertainty, the Monte Carlo method has become a useful method [28]. For this reason, this study proposes a new methodology with the aim of estimating the Contingency Reserve of the budget using the Monte Carlo method. The final purpose is to provide managers with a better global budget, so as to improve the controlling process [29]. As a whole, what is sought is not only the success of the project, but also its efficiency [30], since without a good initial budget estimate for this type of project, costs can exceed the market price and make the project unfeasible [31]. In this respect, it is important to control both the budget and the costs of the project throughout its execution [32]. This is why the use of predictive cost models, such as those developed by Adoko, Mazzuchi and Sarkani are recommended [33].

The proposed methodology is appropriate for both the initial budget calculation and the revision of the budget throughout the project, as current research, such as that of Chou suggests [34]. These calculation tools are of special interest in large-scale projects, where the requirements to meet deadlines and costs are even more demanding. Therefore, recent works such as [35], focus on the importance of the schedule and budget of long-term projects, like those involving shipbuilding, where project definition at the earliest stages will help to achieve the above commented objectives.

2. Objectives

This research has as its main goal the estimation of the Contingency Reserve of new shipbuilding projects. In order to achieve this objective, the Monte Carlo method will be employed by using different probabilistic distributions and its verification will be realised through a specific case study: the construction program of a series of 3 ferries.

Results will be focused on improving decision making for shipyard managers. Moreover, the principal advantages and disadvantages of each calculation model employed
will be specified. In addition, several guidelines will be developed in order to be able to implement each variant of the Monte Carlo model. Therefore, it will be applicable to other large scale projects, either in the naval or other sectors like automobile, aerospace, metallurgic, aeronautic, etc.

3. Monte Carlo method application procedure

With the aim of accomplishing the stipulated goals, it is necessary to become familiar with the calculation options of the Contingency Reserve through the Monte Carlo method, as well as the basic criteria for its proper execution.

As a consequence, in the following paragraphs, the methodology employed for the Monte Carlo simulations will be described, as well as the recommended calculation methodology for the different budgets with which it can be accurately used.

3.1 Description of the Monte Carlo method for the budget Contingency Reserve estimate.

The budget Contingency Reserve of a large scale project should be calculated with particular care. If its value is too high, the project will not be competitive in the bidding processes, [15] while if the value is too low, it could lead to losses, even causing the company to go bankrupt [36].

Inside this framework, a method as reliable as possible for the attainment of the Management Margin should be developed. In this sense, the Monte Carlo method has been established [13], and its refinement and precision has been sought, employing different probability distributions.

Before using the Monte Carlo technique, a preliminary budget is calculated, also estimating the probability of its compliance. The most current bibliography in Project Management [13, 36], recommends that the final budget for the project have a reliability rating of between 80% and 90%.

The difference between the base budget, calculated using the different budget estimation techniques [37, 38], and the budget calculated with a reliability of between 80% and 90% will be the amount that comprises the Contingency Reserve concept.

This number is significantly high, with values that vary between 5% and 15% of the total quoted [13], which implies that it should be calculated as accurately as possible.

With this aim, the present research develops the Monte Carlo method by using two alternatives: triangular distribution and an extension of it.

3.2 Triangular distribution for the Monte Carlo method

Triangular distribution, whose density function is given as

\[
f(x | a, m, b) = \frac{2}{b-a} \begin{cases} 
\frac{x-a}{m-a} & \text{if } a \leq x < m \\
\frac{b-x}{b-m} & \text{if } m \leq x \leq b 
\end{cases}
\]  

(1)

requires three parameters or points \((a, m \text{ and } b)\) for its calculation. In the budgetary context [39] these points are defined as:
The first one, the base budget or most likely budget, is obtained by the different internal calculation processes and corresponds to the probabilistic distribution mode. This budget can be calculated using different procedures. This will depend on the shipyard and can range from the use of historical data to the use of a formula adapted for the type of vessel to be built. In the case of this research, the formulation developed by Alvariño-Castro, Azpiroz-Azpiroz and Meizoso-Fernández [37] has been used. This formulation is employed to estimate the budgets of merchant vessels in the initial stages of the projects, starting from the basic data of the vessels, such as the length, breadth, deadweight, installed power, etc.

The most optimistic budget is the lowest, corresponding to the lower end point of the probabilistic distribution.

At the other extreme, the most pessimistic budget is the highest, and corresponds to the upper end of the probabilistic distribution. In Figure 1, the three points that define the triangular distribution are shown as employed for the calculation of the budget.

![Triangular distribution](image)

**Fig. 1** Triangular distribution applied to budget generation: (a) Density probability function; (b) Cumulative distribution function

By definition, the integral of the density curve will have the value of 1, as can be observed in Figure 1b. In this regard, depending on the cost value calculated for the most optimistic, the most pessimistic and the most likely budgets, the density value assigned to the point of the most likely budget will vary, always maintaining the principle that the integral of the resulting curve will be 1.

In Project Management, it is common to use the Monte Carlo method with the triangular distribution for budget generation, since the literature has validated its use [40].

### 3.3 Extension of the generalized triangular distribution for the Monte Carlo method

The extension of the generalized triangular distribution [41], known as TSP using its English acronym (Two-Sided Power distribution), whose density function is given as

\[
f(x | a, m, b, n) = \begin{cases} 
\frac{n}{b-a} \left( \frac{x-a}{m-a} \right)^{n-1} & \text{if } a < x < m \\
\frac{n}{b-a} \left( \frac{b-x}{b-m} \right)^{n-1} & \text{if } m < x < b \end{cases} \quad \text{and } n > 0 \tag{2}
\]
extends the classic triangular distribution in Figure 1, through a fourth parameter or calculation point, n, for which it will be necessary to provide more information about the model [42]. This new alignment produces a distribution of density with very diverse shapes, as can be observed in Figure 2.

This fourth point modifies the shape of the distribution curve, due to the fact that it provides two types of data: its value on the x-axis, i.e. the cost value given by the specialist, and the probability of compliance with this budget, which is translated into the value of the integral of the probability curve up to this point, which will give a concave or convex shape to the curve.

Fig. 2 Density probability functions of TSP distribution applied to budget generation

In order to operate with this distribution in the budget generation context, 4 points are necessary [43]:

- Base budget or most likely budget
- Most optimistic budget
- Most pessimistic budget
- The value of the n parameter, obtained from the data provided by the specialist

The three first points are similar to the ones described in the previous section. The fourth point is the one that is new and necessary for this extension. The calculation of this fourth point can be carried out using the following information:

- A budget value provided by a specialist in the area to be analysed
- The assignation of the compliance probability value given by the specialist

The criteria and proceedings for the attainment of each one of the four necessary points, and especially the value of n, will be developed in the following sections.

3.4 Calculation procedure for the base budget of a new shipbuilding project

The first aspect that is necessary to calculate is that corresponding to the base budget of the new shipbuilding project, for one ship or for a series of ships. This value is needed for the distributions employed with Monte Carlo, as a base for all the budget processes of the shipyard [16].

As has been indicated, revisions will be carried out throughout the life of the project, and will evolve according to design and construction changes. The increases or decreases will be managed using the Contingency Reserve. Therefore, the budget value cannot exceed, in
any case, the sum of the Contingency Reserve and the initial base budget \([36]\). If this situation occurs, the budget will be greater than that stipulated in the initial contract, which would mean that the project would have to be written off.

For base budget calculations, there are different proceedings. The most exact one is to start from a calculation executed previously by the shipyard. However, if the shipyard has never executed constructions of this type of vessel before, it is recommendable to use other methods, such as that of Alvariño-Castro, Azpíroz-Azpíroz and Meizoso-Fernández \([37]\) or to follow the cost groups that are developed in the technical manuals of the US Navy \([38]\), represented in Figure 3:

![Fig. 3 Structure of the base budget](image)

Regarding the 900 "Support Services" concept, its scope includes all the indirect concepts needed to carry out the project, such as scaffolding, cleaning, insurance for building and construction, management costs, etc.

### 3.5 More optimistic and more pessimistic budgets of the calculation procedure for ships

Starting from the base budget, disaggregated into every concept shown in Figure 3, the most optimistic and the most pessimistic budgets are developed.

For this aim, a certainty grade is assigned to each concept \([13]\), and corresponding to each certainty grade, a perceptual grade of pessimistic and optimistic value, referenced to the base budget of each concept.

The certainty or definition grade is defined according to the maturity of the design and the knowledge of each concept at the time of budget calculation \([16]\). The most recent project management theories propose 3 degrees of certainty to be assigned to each concept \([13]\), which are reflected in Table 1. In order to assign a certainty grade to the correct budget item, it is necessary to evaluate the knowledge that the yard has of the item to be assessed, as well as or whether the scope of these activities is known in great or too little detail. An example is the case of the cost of painting. If the surface area in square meters to be painted has been defined for each type of surface finish, a high degree of certainty can be assigned, but if the exact number of square meters and the requirements required for each type of painting are not available, a low degree of budgetary certainty will be provided for this item.
Table 1 Evaluation of the degree of definition of budget concepts

<table>
<thead>
<tr>
<th>Definition grade</th>
<th>Optimistic budget</th>
<th>Pessimistic budget</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>-25%</td>
<td>+75%</td>
</tr>
<tr>
<td>Medium</td>
<td>-10%</td>
<td>+25%</td>
</tr>
<tr>
<td>High</td>
<td>-5%</td>
<td>+10%</td>
</tr>
</tbody>
</table>

Once the definition grade of each evaluated allotment is assigned, the results will be added together, thus obtaining the most optimistic and pessimistic overall budgets.

By obtaining each of these two values, together with the value of the base budget, the three necessary points for the execution of the simulation of the Monte Carlo triangular distribution will be obtained.

3.6 Criteria for the allocation of the degree of compliance with the budget provided by specialists

Referring to the budget provided by the specialist, as has been mentioned in section 3.3 two types of data are required: the budget and the grade of likely compliance.

Regarding the amount of the budget, it is important to choose the specialist carefully, selecting the person who has the best knowledge possible to contribute to this value. This specialist could be part of the shipyard personnel or an external supplier. Research, such as [44], investigates variations in project management practice. For this reason, the evaluation should be different when the specialist is internal rather than external.

When the concepts correspond to work carried out by the shipyard's own personnel, this amount will be requested from the workshop or department that is to carry out the work. In the case of an external supplier, an estimate will be requested from the supplier.

To assign the likelihood of compliance with the budget, the specialist should be evaluated. In order to do this, the following criteria are proposed:

- In the case of an in-house specialist, two values will be taken into account: the first will correspond to the level of automation and optimisation of the workshop where the work will be carried out. The second will be set according to the stability of the business in which the work will be carried out. For example, in the case of a welding workshop, market fluctuations in the prices of the materials used for welding will be assessed.

- Similarly, when referring to an external supplier, two aspects will be valued: the first one corresponding to the supplier’s qualification, following the parameters Tier One, Tier Two or Tier Three [45, 46]; Tier parameters are assigned by the shipyard to its suppliers, evaluating the quality of the services provided, that is, if the expectations of the products have been met within the established deadlines and costs. This assessment is reviewed periodically to keep the information updated, and thus allowing decisions regarding the choice of suppliers to be made. The second valuation will be based on the stability of the business.

Each of these assessments is assigned a certain degree of compliance probability, as shown in Tables 2 and 3, and the final compliance probability with the specialist budget will be obtained as the average between the results of the two assessments.

Consequently, the budget provided by the specialist may be higher or lower than the shipyard's estimate. If it is lower, the probability of compliance should be less than 25%. Conversely, if it is higher than the shipyard’s budget, the probability of compliance should be
higher than 50%. These two considerations are necessary for the distribution of the TSP to be well defined.

Applying the above consideration, which is necessary in order to use the TSP distribution, two application ranges are obtained, which are 0% to 25% if the specialist's budget is lower than estimated, and 50% to 100% if it is higher. In order to assign a specific value within each interval, these will be broken down into 3 levels of certainty, following the criteria that were employed in assigning the 3 levels of confidence to the base budget, as defined in Table 1, thus proposing the allocation shown in Tables 2 and 3.

Table 2 shows the values to be allocated in the event that the budget estimated by the specialist is lower than the basic budget calculated by the sales department.

<table>
<thead>
<tr>
<th>Automation level and optimization of own workshop or confidence level in the supplier</th>
<th>% Confidence</th>
<th>Stability level of the business</th>
<th>% Confidence</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>25%</td>
<td>1</td>
<td>25%</td>
</tr>
<tr>
<td>2</td>
<td>20%</td>
<td>2</td>
<td>20%</td>
</tr>
<tr>
<td>3</td>
<td>15%</td>
<td>3</td>
<td>15%</td>
</tr>
</tbody>
</table>

In the event that the budget transmitted by the specialist is higher than calculated, the confidence values shown in Table 3 will be assigned.

<table>
<thead>
<tr>
<th>Automation level and optimization of own workshop or confidence level in the supplier</th>
<th>% Confidence</th>
<th>Stability level of the business</th>
<th>% Confidence</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>90%</td>
<td>1</td>
<td>90%</td>
</tr>
<tr>
<td>2</td>
<td>75%</td>
<td>2</td>
<td>75%</td>
</tr>
<tr>
<td>3</td>
<td>50%</td>
<td>3</td>
<td>50%</td>
</tr>
</tbody>
</table>

Once the base budget values, the optimistic budget, the pessimistic budget and the specialized budget are available, along with the degree of confidence for each concept, either the Monte Carlo simulation or the TSP distribution can be carried out.

4. Case study

In order to verify the use of the Monte Carlo method with the two probability distributions described in the previous section, the triangular distribution and the Two-Sided Power distribution, a specific case study will be presented, consisting of a project to construct a series of three ferry-type vessels.

To apply the methodology described above, the following series of steps will be followed:

- Description of ship.
– Calculation of the base budget for the series of vessels.
– Definition of the most optimistic and most pessimistic budgets. This step will be oriented to the application of the Monte Carlo method with triangular distribution.
– Specialist budget and assigned degree of achievement, completing the previous step with the definition of the values needed to apply the TSP distribution.
– Contingency Reserve estimation, comparing the results obtained from the application of the two proposed distributions.

4.1 Characteristics of the ship under study

As the first point of the investigation, a ferry which is representative of the sector must be selected. For this purpose, the Ferry “Volcán de Tindaya” has been chosen. It is a ferry belonging to the ARMAS shipping company.

The choice of this ship is due to the fact that it contains the most relevant properties of the ships of its type, such as dimension, velocity, passenger capacity and route of operation. The “Volcán de Tindaya” services the route Fuerteventura (Spain) – Lanzarote (Spain), which it carries out in approximately 40 minutes, at a service speed of 16 knots. The main characteristics of this ship are reflected in Table 4.

Table 4 Main characteristics of the base ship

<table>
<thead>
<tr>
<th>Concept</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall length</td>
<td>78,10 m</td>
</tr>
<tr>
<td>Length between perpendiculars</td>
<td>65,50 m</td>
</tr>
<tr>
<td>Extreme breadth</td>
<td>15,50 m</td>
</tr>
<tr>
<td>Depth to lower deck</td>
<td>4,80 m</td>
</tr>
<tr>
<td>Depth to upper deck</td>
<td>9,80 m</td>
</tr>
<tr>
<td>Mean draft</td>
<td>3,30 m</td>
</tr>
<tr>
<td>Deadweight</td>
<td>450 t</td>
</tr>
<tr>
<td>Net tonnage</td>
<td>1114 t</td>
</tr>
<tr>
<td>Gross tonnage</td>
<td>3715 t</td>
</tr>
<tr>
<td>Designed sea speed</td>
<td>16 kn</td>
</tr>
<tr>
<td>Range</td>
<td>2300 miles</td>
</tr>
<tr>
<td>Superstructure</td>
<td>3 levels</td>
</tr>
</tbody>
</table>

It is capable of transporting up to 700 passengers, distributed among different lounges and other spaces, such as terraces, cafeterias, etc. The crew consists of 18 people who sleep on board. In addition, it has 110m of 3m wide paved surface for the transport of trucks, and 480m of 2m wide paved surface for the transport of cars. All vehicles are transported in a continuous round cargo space from bow to stern. For the loading and unloading of vehicles, the ferry has two stern ramps of 6.5m in length and 5.5m in beam for vehicles of up to 48 tons. It also has a bow rudder to give access to the bow ramp.

4.2 Base budget of the ship

Following the structure of the technical manuals of the US Navy [38], the base budget of each of the concepts that are shown in Figure 3 has been developed.
The calculations have been based on the reference formulation of Alvariño-Castro, Azpiroz-Azpiroz and Meizoso-Fernández [37], updating the rates to current market values. Table 5 serves as a summary of the first-level concepts.

**Table 5 Base Budget of the construction of a series of 3 Ferries**

<table>
<thead>
<tr>
<th>Concept</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 - Hull structure</td>
<td>4.215,381 €</td>
</tr>
<tr>
<td>200 - Propulsion plant</td>
<td>1.871,150 €</td>
</tr>
<tr>
<td>300 - Electric plant</td>
<td>3.467,586 €</td>
</tr>
<tr>
<td>400 - Electronics and control</td>
<td>582,000 €</td>
</tr>
<tr>
<td>500 - Auxiliary services</td>
<td>3.288,006 €</td>
</tr>
<tr>
<td>600 - Equipment and qualification</td>
<td>2.789,402 €</td>
</tr>
<tr>
<td>800 - Technical services</td>
<td>1.130,470 €</td>
</tr>
<tr>
<td>900 - Support services to the manufacturing</td>
<td>4.521,882 €</td>
</tr>
<tr>
<td>Engineering and manpower</td>
<td>40.310,000 €</td>
</tr>
<tr>
<td><strong>3 vessels total cost</strong></td>
<td><strong>62,175,877 €</strong></td>
</tr>
</tbody>
</table>

4.3 The most optimistic and the most pessimistic budgets

Following the criteria established for the definition of the most pessimistic and optimistic budgets, work has been done in each of the concepts of level 2 in the budget, which contained a total of 35 concepts, in some cases falling to level 3 in order to estimate the value in greater detail.

To each one of these last level concepts, a value of definition coefficient (high, medium, low) has been assigned, and the coefficients defined in Table 1 have been applied to it.

Table 6 shows the summary of the values obtained from the optimistic and pessimistic budgets. These values refer to each of the first-level budget concepts. The definition coefficient shown in the table is the average of the coefficients applied to level 2 of the cost concepts, which is the level at which the calculations have been made, and therefore an indicative value that cannot be directly applied to the values in Table 6.

As indicated above, the calculations have been carried out at a more detailed level of cost concepts, level 2, which is too extensive for presentation. Each of the level 2 concepts has been assigned a definition coefficient (low, medium, high), as this is a case study focused on the verification of estimation methods, and not a budgeting case study. The coefficients have been arbitrarily assigned to cover the widest possible range of combinations in order to better evaluate the Monte Carlo method.
Table 6 Development of the optimistic and pessimistic budgets

<table>
<thead>
<tr>
<th>Concept</th>
<th>Definition coefficient</th>
<th>Optimistic budget</th>
<th>Base budget</th>
<th>Pessimistic budget</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 - Hull structure</td>
<td>High</td>
<td>3,998.772 €</td>
<td>4,215.381 €</td>
<td>4,630.157 €</td>
</tr>
<tr>
<td>200 - Propulsion plant</td>
<td>Medium</td>
<td>1,684.036 €</td>
<td>1,871.150 €</td>
<td>2,338.939 €</td>
</tr>
<tr>
<td>300 - Electric plant</td>
<td>Medium</td>
<td>3,120.827 €</td>
<td>3,467.586 €</td>
<td>4,334.483 €</td>
</tr>
<tr>
<td>400 - Electronics and control</td>
<td>Low</td>
<td>436.500 €</td>
<td>582,000 €</td>
<td>1,018,500 €</td>
</tr>
<tr>
<td>500 - Auxiliary services</td>
<td>Low</td>
<td>2,479.464 €</td>
<td>3,288.006 €</td>
<td>5,785.416 €</td>
</tr>
<tr>
<td>600 - Equipment and qualification</td>
<td>High</td>
<td>2,649.933 €</td>
<td>2,789.402 €</td>
<td>3,068.343 €</td>
</tr>
<tr>
<td>800 - Technical services</td>
<td>Medium</td>
<td>1,017.636 €</td>
<td>1,130.470 €</td>
<td>1,413.383 €</td>
</tr>
<tr>
<td>900 - Support services to the manufacturing</td>
<td>Medium</td>
<td>4,070.543 €</td>
<td>4,521.882 €</td>
<td>5,653.533 €</td>
</tr>
<tr>
<td>Engineering and manpower</td>
<td>High</td>
<td>38,294.500 €</td>
<td>40,310,000 €</td>
<td>44,341,000 €</td>
</tr>
<tr>
<td><strong>3 vessels total cost</strong></td>
<td></td>
<td><strong>57,752,212 €</strong></td>
<td><strong>62,175,877 €</strong></td>
<td><strong>72,583,754 €</strong></td>
</tr>
</tbody>
</table>

4.4 Budget provided by the specialists and their degree of compliance

As for the specialist budget, each final level concept has been studied, defining, in the first place, whether the work will be done by its own staff or by a supplier.

The second step is to obtain the degree of compliance, which will be applied to the budget provided by each specialist. For this purpose, the criteria in Table 2 have been followed, assigning each concept a value between 1 and 3 for the level of optimisation of the workshops themselves or the level of the supplier, and another value of 1 to 3 for the stability of the business.

Taking the values of Tables 2 and 3 corresponding to these allocations and according to whether the specialist budget is higher or lower than the base budget, the degrees of compliance are obtained. Table 7 shows the example of the group "100 - Hull structure".

Table 7 Specialist budget and confidence level assigned for the concepts of the group “100 – Hull structure”

<table>
<thead>
<tr>
<th>Concept</th>
<th>Base budget</th>
<th>Specialist budget</th>
<th>% Specialist compliance</th>
<th>Own staff/ Supplier</th>
<th>Workshop level or Tier</th>
<th>Business</th>
</tr>
</thead>
<tbody>
<tr>
<td>110 - Rolled steel</td>
<td>2,762,439 €</td>
<td>2,800,000 €</td>
<td>75%</td>
<td>Own</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>120 – Cast and Forged Parts</td>
<td>44,985 €</td>
<td>43,000 €</td>
<td>20%</td>
<td>Own</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>130 - Rudders</td>
<td>205,350 €</td>
<td>200,000 €</td>
<td>20%</td>
<td>Own</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>140 - Auxiliary materials</td>
<td>227,076 €</td>
<td>243,440 €</td>
<td>75%</td>
<td>Own</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>150 - Surface Preparation</td>
<td>624,627 €</td>
<td>650,500 €</td>
<td>70%</td>
<td>Own</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>160 - Painting</td>
<td>344,757 €</td>
<td>368,600 €</td>
<td>83%</td>
<td>Supply</td>
<td>1</td>
<td>2</td>
</tr>
</tbody>
</table>
4.5 Contingency Reserve estimation

Once all the budget values for each defined concept have been calculated, the Monte Carlo simulation can be carried out.

When it comes to getting the 4 budget points (base, optimistic, pessimistic and specialist) as well as the triangular distribution, the TSP distribution can be used. Figure 4 shows the results obtained by carrying out the Monte Carlo method with the two distributions, making a total of 1 million simulations for each distribution.

![Fig. 4 Monte Carlo results, a) triangular distribution, b) TSP distribution](image)

With these results, the Contingency Reserve can be calculated as the difference between the obtained budget of the simulation for the desired grade of confidence and the ship construction base budget. The level of risk assumed will vary according to the type of project to be carried out and the company's experience in this field [47].

Table 8 shows the results for both distributions, which are obtained by taking the compliance values of 80%, 85% and 90% of the million simulations that have been executed for the two probability distributions studied for the Monte Carlo method. The calculations have been made by programming the models using Matlab.

<table>
<thead>
<tr>
<th>Confidence grade</th>
<th>Triangular distribution budget</th>
<th>TSP distribution budget</th>
<th>Triangular distribution Contingency Reserve</th>
<th>TSP distribution Contingency Reserve</th>
</tr>
</thead>
<tbody>
<tr>
<td>80%</td>
<td>64.987.462 €</td>
<td>64.856.604 €</td>
<td>2.798.603 €</td>
<td>2.667.745 €</td>
</tr>
<tr>
<td>85%</td>
<td>65.186.089 €</td>
<td>65.014.343 €</td>
<td>2.997.230 €</td>
<td>2.825.484 €</td>
</tr>
<tr>
<td>90%</td>
<td>65.434.446 €</td>
<td>65.212.860 €</td>
<td>3.245.587 €</td>
<td>3.024.001 €</td>
</tr>
</tbody>
</table>

5. Results

The results obtained throughout the research have been classified into two main groups: those obtained from the triangular distribution model and those obtained with the TSP distribution.
5.1 Results obtained with the triangular distribution

In relation to the triangular distribution, it’s use for project management is well known and has been validated by literature [40]. In the case of this research, a Contingency Reserve of between 2.8 and 3.2 million euros is achieved, which amounts to between 4.5% and 5.2% of the ship’s base budget.

The choice of the margin of confidence is based on the level of risk that the organization can assume in the new contract [36], as well as the needs that the company has to enter in a new sector [2].

5.2 Results obtained with the extension of the generalized triangular distribution

The TSP distribution shows more favourable results than the ones obtained with the triangular distribution, resulting in a Contingency Reserve of between 2.6 and 3 million euros. This is between 4.3% and 4.9% of the base budget.

Note that this distribution is calculated at a higher information level, which suggests that this estimate is more correct than that made with the triangular distribution.

5.3 Comparison of the results obtained with the different distributions employed with the Monte Carlo method

By comparing the results of the simulations using the triangular distribution and the TSP distribution, Figure 5 is obtained.

![Fig. 5 Comparison of the resulting distributions after the Monte Carlo simulations](image)

As can be seen, the distribution of the TSP covers a smaller range of values, which can be translated into a better definition of the model and therefore, the results with this distribution can be assumed to be more reliable.
6. Conclusions

The importance of estimating a correct budget in the bidding stages is essential in order to be able to compete under the best circumstances. If the company presents too high a budget, this could result in a loss of competitiveness for the tender. Conversely, if the company opts for contracts with too low a budget, the result could be a breach of contract, leading the yard to bankruptcy.

The present research contributes with an estimation method of the Project Contingency Reserve in the early budget calculation stages, examining two evaluation options regarding the quantity of information that could be obtained from the different components that make up the budget structure.

Both the triangular distribution (three-point) and its generalization (four-point) are valid for resolving the value of the Contingency Reserve, the latter being preferable if precise information is available for use by specialists. This extension of the Monte Carlo method can also be used in risk analysis, where it will improve uncertainty, both in the areas of the term and cost of the risks analysed with triangular distribution or beta distribution. This is a field of study in which various research projects are currently being carried out.

Finally, it can be concluded that the correct estimation of the Contingency Reserve will allow us to have the most accurate budgets possible, in order to bid in international tenders and therefore, be able to compete with the risk margins that the organisation considers appropriate to assume.

REFERENCES


Obtaining the budget Contingency Reserve through the Monte Carlo method: study of a ferry construction project


Summary

Familiarisation is an important factor of safety on technologically advanced ships. International Safety Management Code (ISM Code) states that the Company should establish procedures to ensure the familiarisation process, but the exact way in which familiarisation should be carried out and the duration of the process are not determined. Familiarisation is often regarded as formality although it should not be the case at all. Non-compliance with the required familiarisation procedures and flaws in the safety system often result in human error. The latter is a major cause of numerous sea accidents. The research published has revealed that shipping companies and seafarers often fail to follow the prescribed procedures and perform familiarisation inadequately.

This research is based on a survey of Croatian and Montenegrin deck and engine officers. The survey results indicate weaknesses in the familiarisation and handover processes on board ships. Therefore, suggestions are made for enhancing the existing procedures, aiming at a more efficient familiarisation and handover, particularly aboard technologically advanced ships.

Key words: Familiarisation; Safety at sea; Handover; Seafarers; Technologically advanced ships

1. Introduction

Familiarisation is a process of introduction to the ship, duties, and crew, which every seaman must go through upon joining the ship. It includes familiarisation with the ship systems and equipment, emergency procedures, and procedures described in the Ship Safety Management System – SMS manual. It is a demanding process, especially on technologically advanced vessels. One of the essential components of efficient familiarisation is the handover procedure between officers (deck or engine) when leaving / joining the ship. Handover is a procedure of exchange of responsibilities and work duties of two officers (deck or engine) in the same rank. It is usually obtained in ports while the ship is alongside or at anchor. The main distinction between familiarisation and handover is that familiarisation is a checklist
prescribed by the Conventions and the Company, and handover is a process including off-signing and on-signing officer describing completed and pending jobs, maintenance which needs to be done, etc. The common feature of both familiarisation and handover is that there is no written regulation on procedure duration. Investigations of sea accidents have revealed that insufficient familiarisation is one of the major causes of accidents [1].

In 1998, the International Maritime Organization (IMO) issued the International Safety Management Code (ISM Code) whose regulations became part of the International Convention for the Safety of Life at Sea - SOLAS Convention. The ISM Code served as a framework for creating the Safety Management System and Safety Management Manual, a handbook defining all safety procedures and checklists, including the procedures regarding familiarisation and handover on board ships [2].

Familiarisation is one of the first procedures that a crew member experiences when signing on. Upon joining the ship, the new crew member receives a familiarisation checklist from the officer in charge. The familiarisation checklist varies from one seafarer to another, depending on their rank, department, and ship type. According to the International Convention on Standards of Training, Certification and Watch-keeping for Seafarers (STCW), familiarisation is divided into Basic Safety familiarisation, Ship-specific and Security familiarisation. The checklist commonly consists of a part which must be completed on the date of joining and before taking the watch, and a part which must be completed as soon as possible, but not later than one week after joining [3].

The first part includes the procedures and duties such as:

- Be able to communicate with other persons on board on elementary safety matters, understand safety information symbols, signs and alarm signals, identify muster and embarkation stations and emergency escape, locate and don lifejackets, understand and execute security duties assigned to him etc.
- Know what to do if a person falls overboard, fire or smoke is detected, the emergency signal or boat signal is sounded.
- Watch-keeping procedures and arrangements (for all officers and watch-keeping ratings).

The other part includes the procedures and duties such as:

- Learn to operate the CO2 / FOAM /DRY POWDER / HALON Fixed Fire-Fighting System, operate the Emergency Generator, and deploy the Emergency Towing Arrangement, etc.
- Get acquainted with procedure for handling garbage and use of associated equipment, Sewage Treatment Plant, etc.
- Understand the Company’s Management System, Quality, Safety, Health and Environment Protection Policy and Drug & Alcohol Policy, etc.

The prescribed seven-day-period is a relatively short time frame for an efficient familiarisation, especially when a crew member is aboard ship for the first time or is appointed/promoted to a higher rank. The quality of familiarisation is questionable if a crew member joins a vessel that in terms of technology is more demanding than the vessel on which he/she served before. Technologically advanced vessels require specific/specialised knowledge and skills, thereby making familiarisation procedures more comprehensive [4]. The length and quality of familiarisation largely depends on the type of ship and its propulsion, cargo-handling gear and the facilities and arrangement of the bridge and engine room [5].

The bridge arrangement and the advanced features of navigation equipment, as prescribed by the Safety of Life at Sea Convention (SOLAS), are essential items on the
familiarisation agenda [6]. The SOLAS convention defines the minimum requirements regarding the quantity and features of the ship equipment and devices. The shipping company management selects the ship equipment and its manufacturer / provider, depending on the available budget and abilities [7]. The equipment may vary in quality, design and additional abilities/functions. The features that SOLAS does not deem mandatory may create difficulties in the process of familiarisation of an officer and may lead to human error.

An officer should be prepared and trained to use new technologies and equipment fitted to the ship prior to signing on [8]. Still, it is common practice to start familiarising an officer with the new systems at the moment he/she joins the ship. Responsible companies invest in their seafarers by providing them with adequate training before they start dealing with new on-board technologies and systems. In addition, upon joining the ship they are given longer familiarisation time before taking charge of operating the new systems. However, few companies are able to invest additional funds in acquiring expensive simulators for shore-based training of their seafarers, or to allow more days for handover procedure.

The reports of the United States Coast Guard (USCG) and port state control inspections often underline the issue of familiarising crew members with specific ship systems [9,10]. Ships and companies may be liable to relatively heavy fines due to non-compliance with the familiarisation procedure, especially when the crew have not been familiarised with the use of systems that can cause pollution. Poor familiarisation and lack of training was one of the major causes of sea accidents of the vessels “Orsula” [11], “Louis Jolliet” [12], “MS UND Adriyatik” [13], and “CSL Thames” [14].

After a research conducted in 2015 [6], the officers surveyed stated that the quality of familiarisation and handover does not depend on the education degree. In addition, the research revealed that the officers were content with the one-week familiarisation period. On the other hand, they considered the six-hour period of handover as insufficient. These results seemed contradictory, so that subsequent interviews with the respondents were performed in order to provide clarification. Actually, the officers provided such information because they sailed on the sister ships or the vessels featuring similar technologies [6].

2. Hypothesis

Familiarisation is a factor that considerably affects the safety of navigation, especially on technologically advanced ships [7,15]. The familiarisation process is often performed in an inadequate way, and the seafarers and their companies are not sufficiently aware of the risks that may arise due to poor familiarisation. SOLAS regulations referring to the process of familiarisation have been laid out in a very general way [16]. The mode of performing familiarisation is vaguely defined and leaves the implementation at the discretion of the seafarers and maritime shipping companies. The aim of the survey was to determine how well familiarisation is performed and to gain insight into how many maritime officers realise the importance of the problem of familiarity, and how many officers understand the importance of familiarisation.

3. Methodology

The survey of seafarers was conducted at the Faculties of Maritime Studies in Split and Kotor. The target group consisted of Croatian and Montenegrin deck and engineer officers who were about to take their exams for acquiring the ranks of Chief Officer/Chief Engineer on ships of more than 3,000 GT, i.e. powered by the main propulsion machinery of 3,000 kW or more. Some questionnaires were completed by the attendants of the Special Education Programme for the certification of seafarers at the Faculties of Maritime Studies in Split and
Kotor. The respondents were officers of different ages, with at least 3 years of sea service in the capacity of deck officers or engineer officers.

Croatian and Montenegrin officers go through similar high-school and higher education systems. Both countries provide Special Education Programme for the certification of seafarers who wish to acquire the rank of Chief Officer on ships of more than 3,000 GT, or the rank of Chief Engineer on vessels with propulsion of 3,000 kW or more. One of the requirements for attending such a programme includes the maritime high-school degree (featuring programmes that comply with STCW A-II/2 Convention), and at least 36 months of sea service as watch-keeping officer.

The survey covered 400 officers (n), including:
- 238 deck officers (59.5%) and
- 162 engineer officers (40.5%).

The questionnaire was written in Croatian and Montenegrin languages, comprising 16 questions with a range of alternative responses. The questions referred to the familiarisation and handover procedures on board ships, and the information closely related to these procedures. The target information included previous sea service, ship type, education, ship department and rank, duration of familiarisation and handover, modes of reduced familiarisation periods, and familiarisation as navigation safety issue.

The survey was conducted throughout 2015 and 2016 by survey assistants whose role was to explain the purpose of the survey and deal with potential ambiguities. Some of the questionnaires were e-mailed to ships, along with a letter asking the master to ensure adequate environment for trustworthy responses. Deviations from the responses expected were checked in subsequent interviews. Part of the material obtained was examined, and additional explanations were required from the respondents.

4. Results and discussion

By analysing the questionnaire, data were obtained of the type of ships that officers were sailing. Some vessels are technically more complex than others, therefore the time of familiarisation is longer. In this regard, it is considered that familiarisation of seafarers, especially deck officers, is most complex on passenger ships, offshore ships, and liquid cargo ships. Marine engineer officers have a prolonged time of familiarisation on ships that have a specific propulsion (e.g. off-shore vessels with additional propulsion systems), and passenger ships due to the complexity and size of the ship's engine room.

Out of the total number of officers who took part in the survey (n=400):
- 16 officers (4%) served on passenger ships,
- 56 officers (14%) served on liquefied gas carriers,
- 78 officers (19.5%) served on tankers,
- 48 officers (12%) served on bulk carriers.
- 144 officers (36%) served on container ships and vehicle carriers,
- 44 officers (11%) served on off-shore vessels, and
- 14 officers (3.5%) served on other types of vessels.

With regard to the technological complexity of vessels, it can be noted that only 12% of the officers sail on bulk carriers, whereas the others (88%) sail on technologically advanced ships featuring special cargo-handling, navigation or propulsion systems.
In the survey conducted, it is stated that experience had positive influence on the familiarisation process duration time. Therefore, many companies embark officers on sister ships or similar vessels in order to reduce familiarisation duration.

1. The question “Your sea experience with the type of ship/engine? “ (n=400) was answered as follows:
   - 8 officers had up to 1 year of experience with the same type of ship/engine (2%),
   - 52 officers had 1 – 3 years of experience with the same type of ship/engine (13%),
   - 62 officers had 3 – 5 years of experience with the same type of ship/engine (15.5%),
   - 278 officers had 5 – 10 years of experience with the same type of ship/engine (69.5%).

2. The question “Your rank during the last contract? “ (n=400) provided the following results:
   - As the 1st Officer – 102 officers (25.5%),
   - As the 2nd officer – 170 officers (42.5%),
   - As the 3rd officer – 56 officers (14%), and
   - As the Chief Engineer / Master – 72 officers (18%).

   The time of familiarisation depends on the rank aboard vessel. A higher rank (management level) means more complex duties aboard and, proportionally, more familiarisation time and more complex procedures.

   Most of the officers (66.5%) were junior officers, i.e. 2nd or 3rd deck or engineer officers.

3. The question “How does your company reduce familiarisation time? “ (n=400) provided the following responses:
   - By serving on sister ships – 224 officers (56%)
   - By simulator-aided training – 12 officers (3%)
   - Joint service of the handover officers until the next port of call or further – 70 officers (17.5%)
   - The company does not consider familiarisation as an issue – 94 officers (23.5%).

   Most of the respondents stated that their company reduced the period of familiarisation by employing officers on sister ships or ships featuring the same technologies (56%). A major concern is the response of 23.5% officers whose companies do not regard familiarisation as an issue. This percentage is rather high, as it indicates that there are still a number of maritime shipping companies which do not consider familiarisation process as a relevant factor of the safety of navigation.

   Likewise, the survey revealed a low percentage (3%) of familiarisation by means of simulator training. This low percentage can be an indication that the companies are not ready or able to invest in the expensive shore-based training for seafarers with the aid of simulators.

4. The responses to the question “Do you consider familiarisation time sufficient when joining the ship?” were analysed with regard to the respondents’ sailing experience (Question: “Years in seafaring service?“), and the following results were obtained (Figure 1):
Out of 28 officers with less than 4 years of sea service (7%), 22 officers responded “yes” (78.57%), whereas 6 responded “no” (21.43%)

Out of 116 officers with 4 – 10 years of sea service (29%), 94 officers responded “yes” (81%), and 22 “no” (19%)

Out of 140 officers with 10 – 15 years of sea service (35%), 108 responded “yes” (77.15%), whereas 32 responded “no” (22.85%)

Out of 116 officers with sea service longer than 15 years (29%), 96 officers responded “yes” (82.76%), and 20 responded “no” (17.24%).

Naturally, the experience gained on board sister ships considerably affects the quality and duration of familiarisation. Previous sea service on similar ships may reduce the time required for proper familiarisation. Exceptions refer to promotions to higher officer ranks, when the period of familiarisation may be longer.

The results show that the officers provided similar responses concerning familiarisation time (from 77.15% to 82.76%) regardless of the sea-service time. A recent research [6] revealed that familiarisation time does not depend on the education of officers, as similar responses are provided by high-school diploma-holders and higher-education degree-holders.

Figure 1 – Relation between seafarers sea service time and their opinion about sufficiency of the familiarisation duration time (n=400)
5. As for the sea-service time (Question: “How many years of sea service do you have?”), the responses were statistically analysed with regard to the question: “Do you consider the handover time sufficient when signing on/off? “ (Figure 2):

- Out of 28 officers with less than 4 years of sea service (7%), 14 officers responded “yes” (50%), and 14 officers responded “no” (50%)
- Out of 116 officers with 4 – 10 years of sea service (29%), 88 responded “yes” (75.86%), whereas 28 officers responded “no” (24.14%)
- Out of 140 officers with 10 – 15 years of sea service (35%), 94 responded “yes” (67.15%), and 46 responded “no” (32.85%)
- Out of 116 officers with sea service longer than 15 years (29%), 92 officers responded “yes” (73.3%), whereas 24 responded “no” (20.7%).

Most of the experienced officers (>4 years of sea service) believed that the handover time was appropriate. Handing over is a more demanding process if officers are promoted to a higher rank. Unfortunately, companies usually change crews in smaller groups and do not make any difference between the cases in which an officer is promoted to a higher rank, or if he/she signs on in the same rank. This way of changing crew is used to reduce costs such as delay of departure of ships, travel costs, etc.

![Diagram](image)

**Figure 2** – Relation between seafarers sea service time and their opinion about sufficiency of the hand-over duration time (n=400)
6. The responses to the question “How many years of sea service do you have?” were analyzed with regard to the question: “On the basis of your experience, do you believe that familiarisation is an important issue for safe navigation?” (Figure 3):

- Out of 28 officers (7%) with less than 4 years of sea service, 12 responded “yes” (42.85%) whereas 16 officers responded “no” (57.15%)
- Out of 116 officers with 4 – 10 years of sea service (29%), 58 responded “yes” (50%) and 58 officers responded “no” (50%)
- Out of 140 officers with 10 – 15 years of sea service (35%), 72 responded “yes” (51.43%) whereas 68 officers responded “no” (48.57%)
- Out of 116 officers with sea service longer than 15 years (29%), 64 responded “yes” (55.17%) and 52 officers responded “no” (44.83%).

When asked about the familiarisation as an issue for the safety of navigation, the officers had divided opinions. Regardless of their experience, approximately half of the seafaring officers (42-55%) did not consider familiarisation as an issue. In later interviews (during checking of answers), the officers stated that they had never heard of anything dangerous regarding the problem of familiarisation, or the vessels causalities due to insufficient familiarisation [10, 11, 12, 13].

![Figure 3](image-url)

**Figure 3** – Relation between seafarers sea service time and their consideration/opinion about familiarisation procedure as an safety issue (n=400)
7. With regard to their rank (Question: “What rank did you have on your latest contract?“), the officers were asked: “How long was the handover procedure when signing on/off your latest ship?”. The statistics provided the following results (Figure 4):

- **1st Officers**: 102 seafarers signed on in this rank – 25.5%:
  - 25 had the handover lasting up to 2 hours (24.5%),
  - 14 had the handover lasting for 2 – 4 hours (13.7%),
  - 11 had the handover lasting for 4 – 6 hours (10.8%),
  - 52 had the handover lasting for more than 6 hours (51%).

- **2nd Officers**: 170 seafarers signed on in this rank – 42.5%:
  - 56 had the handover lasting up to 2 hours (33%),
  - 53 had the handover lasting for 2 – 4 hours (31.2%),
  - 14 had the handover lasting for 4 – 6 hours (8.2%),
  - 47 had the handover lasting for more than 6 hours (27.6%).

- **3rd Officers**: 56 seafarers signed on in this rank – 14%:
  - 20 had the handover lasting up to 2 hours (35.7%),
  - 6 had the handover lasting for 2 – 4 hours (10.7%),
  - 8 had the handover lasting for 4 – 6 hours (14.3%),
  - 22 had the handover lasting for more than 6 hours (39.3%).

- **Chief engineers / masters**: 72 seafarers signed on in this rank – 18%:
  - 17 had the handover lasting up to 2 hours (23.6%),
  - 8 had the handover lasting for 2 – 4 hours (11.2%),
  - 14 had the handover lasting for 4 – 6 hours (19.4%),
  - 33 had the handover lasting for more than 6 hours (45.8%).

The analysis of the results indicates that senior officers (Chief Officers, Masters, 1st Engineers and Chief Engineers) had longer periods of handover, i.e. more than 6 hours. Junior officers (second and third officers) had shorter periods of handover, i.e. up to 2 hours. The extended handovers present deviations from the results expected, due to promotion to a higher rank or taking up new duties.
With regard to the duties / ranks on their latest contracts (Question: “What rank did you have on your latest contract?”), responses to the question “How long was familiarisation when signing on/off your latest ship?” were analyzed (Figure 5):

- **1st Officers** (102 seafarers signed on in this rank – 25.5%):
  - 85 had the familiarisation lasting up to 7 days (83.33%),
  - 14 had the familiarisation lasting for 7 - 15 days (13.73%),
  - 3 had the familiarisation lasting for 15 – 30 days (2.94%),
  - No one had the familiarisation longer than 30.

- **2nd officers** (170 seafarers signed on in this rank – 42.5%):
  - 142 had the familiarisation lasting up to 7 days (83.5%),
  - 22 had the familiarisation lasting for 7 - 15 days (12.9%),
  - 3 had the familiarisation lasting for 15 – 30 days (1.8%),
  - 3 had the familiarisation lasting for more than 30 days (1.8%).

- **3rd officers** (56 seafarers signed on in this rank – 14%):
  - 39 had the familiarisation lasting up to 7 days (69.6%),
  - 17 had the familiarisation lasting for 7 - 15 days (30.4%),
  - No one had the familiarisation longer than 15 days.

- **Chief engineers / masters** (72 seafarers signed on in this rank – 18%):
  - 66 had the familiarisation lasting up to 7 days (91.6%),
  - 3 had the familiarisation lasting for 7 - 15 days (4.2%),
  - 3 had the familiarisation lasting for 15 – 30 days (4.2%),
  - No one had the familiarisation longer than 30.
Regardless of the rank they had when joining the latest ship, most of the officers experienced familiarisation lasting for up to 7 days (from 69.6% to 91.6%). The fact that some officers had familiarisation lasting from 7 to 15 days (13.73% and 12.9% for the 1st and 2nd Officers respectively) is the result worth noting. Deviations arise from the promotion to the new rank/duties. Few officers who joined their ship as 2nd officers had the period of familiarisation from 30 to 45 days. Deviations from the expected results arise from the promotion to a new rank/duties or from changing the company (new familiarisation procedures). Deviations can also be noted among 3rd officers, as 30.4% of them had familiarisation lasting for 7-15 days. The reason for this is their joining the ship as officers for the very first time (promotion from cadets to officers), so that longer periods of familiarisation were required.

![Figure 5 - Relation between seafarers ranks during the latest contracts and duration of familiarisation](image)

According to the survey, it results that the officers examined do not consider familiarisation as an important issue on board. They also think that duration of familiarisation is satisfactory. Familiarisation depends on their rank in the shipboard hierarchy. Higher ranks accept familiarisation more seriously and need more time for it. Junior Officers find that duration of familiarisation is satisfactory.

Most examined officers accept the procedures of companies in the matter of familiarisation (e.g. signing on the same or sister ships, etc.) as a good way of its improvement.
5. Recommendations

Familiarisation process can and should be improved by:
- training ashore – STCW basic training, Ship specific training (ECDIS, Cargo handling, etc.),
- training on board (CBTs, Feedback across company etc.),
- joining the vessels with similar technologies, e.g. when a seafarer is familiarised with container ships, the company tends to keep him/her on this type of ship,
- back-to-back contracts – type of arrangement where the seafarer always returns to the same ship, at least for 4 highest ranks (Master, Chief Engineer, C/O and 1 A/E),
- defining the exact time of familiarisation and handover,
- extended handover,
- introduction of junior officer training – the period of time given to the officer prior to his/her promotion to a higher rank,
- Officer Matrix requirements – an officer must have sufficient experience, i.e. sea-service time in the rank on a particular type of vessel, and be with the same company for some time,
- strict definition of the procedures of familiarisation and handover by Safety Management System (SMS),
- regular performance of emergency drills in compliance with the international conventions, etc.

Familiarisation problem could be solved by embarking the same crew on the same or sister vessels.

Vessel ergonomics should be improved, especially in the parts which are used for command, control, and supervision. This solution has already been known in the airline industry, where the pilot is certified for flying a specific type of airplane with which he/she has enough experience collected during training on simulators and as co-pilot on the airplane. The same solution can be applied on board vessels.

One of the most discussed ways of dealing with familiarisation refers to the navigation equipment and development of the S–Mode, i.e. Standardized Mode of operation of navigational equipment. The Nautical Institute is developing the concept of the standard “S–Mode switch”. When activating the switch, navigation device would turn to the pre-set mode with the same settings across the equipment, regardless of the manufacturers [17]. However, it is important that the S–Mode does not limit the producer innovation and enhancement of the navigation equipment. There is an IMO document (MSC/95/19/12) with a three-year plan whereby the drafting of an S-Mode guideline for the design of shipboard navigational equipment along with notes for training implications would be completed by the end of 2019 [18,19].

6. Conclusion

The familiarisation as a process of on-board adaptation still remains insufficiently defined, in particular from the viewpoint of the duration and quality of the process. The latter is usually described within Safety Management System (SMS) rulebooks of individual companies and ships so that it presents the matter related with the quality management. In real
life, it is common practice to cut the time of familiarisation and handover in order to reduce the costs of manning.

Although the SMS can be adjusted to the type of vessel, it is necessary to make the procedures clearer.

Embarking on sister ships or similar ships is a good way of reducing familiarisation time.

Administration should consider the possible solution of ergonomic vessels, especially in the parts where the probability of human error is highest. This practice has already been in use in air transport.

Training should be obtained on simulators which are not generic but specific, and more similar to shipboard design.

Future research should thoroughly examine familiarisation and handover processes when officers are promoted to the higher rank for the first time, and when officers change the employer or the type of vessel.

It is recommended that the time of familiarisation is reduced through shore-based, simulator-aided training, uniform design of controls, and innovations such as S-Mode switch in order to enhance the safety of navigation and cut the costs.

7. References


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Practice of and attitudes toward familiarisation on board: Survey of Croatian and Montenegrin maritime officers


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TOWARD SHIPBUILDING 4.0 - AN INDUSTRY 4.0 CHANGING THE FACE OF THE SHIPBUILDING INDUSTRY

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Review paper

Summary

The Shipbuilding 4.0 at the principles of the Industry 4.0 will transform the design, manufacturing, operation, shipping, services, production systems, maintenance and value chains in all aspects of the shipbuilding industry. Over the last few years, the fourth industrial revolution has spread in almost all industries. The whole world is talking about Industry 4.0 which has increased implication in the manufacturing process and the future of the work. The impact of the Shipbuilding 4.0 will be significant. In the past, shipbuilding industry where continuously improved with new machines, software and new implemented organizational restructuring. In today shipbuilding industry, there are three main problems that are considered; production efficiency, the ship safety, cost efficiency and energy conservation and environmental protection. In order to create new value, the ship must become a Smart Ship capable of “thinking”, and to be produced in Smart Shipbuilding Process. The aim of this article is a review of the present academic and industrial progress of this new industrial revolution wave in the shipbuilding sector called Shipbuilding 4.0 (Shipping 4.0, Maritime 4.0, Shipyard 4.0). Reviewed publications were analyzed different topics and level of improvements in the industrial aspects of the society. The implementation of the Shipbuilding 4.0 in the shipbuilding industry, presents the future, creating new value in the process, creating new demands with reduction in production and operational cost while increasing production efficiency.

Key words: Industry 4.0; Shipbuilding 4.0; Smart Ships; Smart Shipbuilding Process; Review of the online available publications; Shipyard 4.0
1. Introduction

Industry 4.0 has been defined as "a collective term for technologies and concepts of value chain organization" which draws together Cyber-Physical Systems, the Internet of Things and the Internet of Services (Jasperneite J. “Was hinter Begriffen wie Industrie 4.0 steckt”, Web magazine computer-automation.de, 19th December 2012.), but the Industry 4.0 concept has existed since 1991, since its introduction by Mark Weiser, [1]. He described the vision of the future with the term of “Ubiquitous Computing”. From that period a lot of the things become reality as are smart mobile phones, cars as wheeled computer system, smart homes.

The first industrial revolution was introduction of water and steam-powered mechanical manufacturing, the second industrial revolution was the implication of electrically-powered mass production; the thirds industrial revolution was introduced to use of electronic and information technologies (IT) to achieve further automation of manufacturing, [2,3]. The Industry 4.0 is based on the Cyber-Physical Systems, it represents the mass customization of the products turned to the wishes of the customer with the implementation of the intelligent, smart and optimal solutions embedded in the products visible on the Figure 1, [4, 5].

![Fig. 1 The four stages of the Industrial Revolution, [2]](image)

The Industry 4.0 is strategic initiative and it represents the synonym for the transformation of today factories into Smart Factories which will be capable to overcome the challenges of the product lifecycle, highly customized products and to stay in the race with ubiquitous competitors. The smart products from the Smart Factories are customized, identifiable and know their current status and target state, [2]. The whole concept is based on Cyber-Physical Production System (CPPS), Internet of Things, Big Data and Internet of Services and interaction of the real and virtual world, [6]. It presents development that changes the overall traditional industries and includes design, technology and innovation cycles which is seen as an important strategy to remain competitive in the future, [7]. The smart products from the smart factories will allow the "last minute" changes to the customer requirements. This dynamic business and engineering process enables the production, delivery and flexibility to disruption and failure during production. For the smooth functioning of the concept, it will be important horizontal and vertical integration through across the value chain, [2].
This paper will analyze, review of the new concept Shipbuilding 4.0 as the follow-up of the Industry 4.0 concept applicable in the shipbuilding industry.

The shipbuilding industry where continuously improved with new machines, software and new implemented organizational restructuring; but still is facing difficulties with large number of changes during construction and large number of ships series led to the loss of control over costs and quality control. The burden of the crisis submitted the shipyards that could not fully meet quality requirements, safety, cost efficiency and the shipbuilding market fluctuations. All above have generated losses.

The paper is organized as follows. Chapter 2 introduces the Industry 4.0 in Maritime Sector and changes in the shipbuilding industry while the Chapter 3 is presented Shipbuilding 4.0 the new industrial path. In the Chapter 4 is presented Shipyard 4.0 design principles and strategy. Chapter 5 presents an implementation concept, methodology proposal. Finally, there are conclusion and guidelines for the future research.

2. Industry 4.0 transformation in Maritime Sector and Shipbuilding industry

Industry 4.0 has strong impact on the worldwide industries and the all aspects of the human society. In this chapter will be presented its influence in Maritime Sector, and the transformation that will come in the shipbuilding industry. Figure 2 illustrates overall connectivity in the Cyber-Physical System of the Maritime 4.0.

![Fig. 2 Exchanging data via Cyber-Physical System, [2].](image)

According to the Capgemini’s recent report, smart factories as the base of the Industry 4.0 will be adding up to 500 billion dollars in value to the global economy. Also, almost 76% manufacturers in the world already have some level of a smart factory initiative.

The main items of the Industry 4.0 applicable in the Maritime Sector and supported industries are:

- Cyber-Physical systems,
- Big Data,
- Digitization of the Industry,
- Internet of Things,
- Internet of the Services,
• Strong and liable cooperation in the whole supply network,
• Human-machines, interaction,
• Production logistics with Cyber-Physical embedded system,
• Agile, digital, flexible production,
• Standardization,
• Changes of the demographic and cultural society,
• Significant changes in the education system.

Almost all mayor players in shipbuilding industry are preparing themselves for the changes that will come in next 10 to 20 years, and strongly working on their own steps toward fourth industrial revolution. In the history, the industrial revolution usually brings the strong changes in the all aspects of the human society supported by the governments:

• In the Germany:
  The official "Industrie 4.0" document was originally released in 2013 by the Germany Federal Ministry for Economic Affairs and Energy. Industry 4.0 (Industrie 4.0) keywords means a development that changes the traditional industries fundamentally, [7]. Almost 12 billion Euros will be invested into the innovations and turnover the shipbuilding industry in the Shipbuilding 4.0. With the high export rate of 74 per cent confirms the Germany the leader in the international marine equipment sector, [8]. According to the "Maritime Agenda 2025" and the "Study on Industrial and Technological Competences in the Naval Sector", the maritime industry has the annual turnover of EUR 18 billion, and abt. 70 to 80 percent of the value is added in the shipbuilding. The defense budget (military) is growing and the German naval shipbuilding is becoming the push industry for shipbuilding innovations in Shipbuilding 4.0 era.

• In the USA:
  American corporations founded the Industrial Internet Consortium (IIC), and five of them started with the increasing the market size of the Internet of the Things in the shipbuilding industry. In the US military shipbuilding was increased current shipbuilding goal from 308 to 355 ships per year. All the new vessels are more technologically advanced and complicated than any previous generation, faster and better than ever before. Also, the ships of the new generation will last longer and be more adaptable to changing the needs throughout their lives. Today US shipbuilding, especially the Navy, facing a tidal wave of increased demands; the NNS (Newport News Shipyard) shipyard is on his way to the Shipbuilding 4.0 digitization process. It is expected, that this approach will generate 15 percent more cost savings over the traditional shipbuilding methods, [9].

• In the China:
  The Chinese government in 2015. issued "Made in China 2025" the initiative strategy draws direct inspiration from Germany Industry 4.0. The plan is creating the innovation centers from 15 to 2020 year to 40 by 2025. Shipbuilding 4.0 in the Chinese shipbuilding industry is called 5S, a ships operation intelligent service system that features Sea, Ship, System, Smart and Services. The smart demo ship is highlighted priority development in "Made in China 2025", [10, 11]. This means ship status safety assessment, ship energy efficiency monitoring, analysis, assessment and optimization, status, assessment and maintenance optimization, sea route, ship navigation and operational control, all connected via Big Data. The delivery of the vessel is expected to be at the end of the 2017, [12].
In the Japan:
The Industry 4.0 idea had announced in 2014, but it is become official in 2016, with the joint declaration of future German-Japanese cooperation signed by two parties in Tokyo. According to the McKinsey & Co, Japan is behind the US and German in the researches. The Industry 4.0 in the Japan is named Industrial Value Chain Initiative (IVI), which is working on the collaboration of the companies, [13].

In the Korea:
Korean version of the Industry 4.0 is called "Manufacturing Innovation 3.0 Strategy" and is published in 2015. Also in 2015 as part of the new change in the industrial innovation, the Korea has launched an innovation center in the shipbuilding capital of Ulsan, Busan. Three major shipbuilders Samsung Heavy Industries, Daewoo Shipbuilding & Marine Engineering and Hyundai Heavy Industries will be in charge of the new Ulsan innovation center. It is expected to be abt. 25 hundred patents dedicated to building the smart ships in the smart shipbuilding environment based on the idea of the Shipbuilding 4.0, [14].

In the Spain:
In 2014 the Galicia started the Agenda for Industrial Competitiveness, as a tool for planning, industrial policy implemented by the Xunta de Galicia in the period 2015-2020. The ACLUNAGA (Cluster Association of Naval Galicia) is a collaboration of the Galician Innovation Agency - GAIN, IGAPN and AIMEN Technology Center has first started with the implementation of the Shipbuilding 4.0 in 2016. The Ferrol shipyard Navantia, Navy Company is a leader of the changes in the shipbuilding sector, [16].

In the Australia:
The digital transformation of the Shipbuilding 4.0 era started in the Adelaide Australian Navy shipyard. The transformation of the shipyard will start in 2018, by investment of 1.5 billion Australian dollars in the design and engineering to become the most advantages Navy shipyard in the world. Almost 100 million Australian dollars will be invested in information and technologies, [17].

The strong changes toward new industrial revolution in the shipbuilding industry are also in the level of the preparation phase in the Norway, Sweden, UK, France, Finland, and Brazil.

3. Shipbuilding 4.0 - new industrial path

The review principles in the article, analyses the overall progress of the forth industrial revolution in the shipbuilding sector named: "Shipbuilding 4.0", "Shipyard 4.0", "Shipping 4.0", "Naval 4.0", "Maritime 4.0", "Smart Yards", "Smart Shipbuilding". The keywords that are explicitly mentioned in the articles were carried out to provide a preliminary view of the Fourth Industrial Revolution in shipbuilding, shipping, vessel operation, progress of new shipbuilding standards, technologies and shipbuilding enabling features.

The world’s shipbuilding industry at the beginning of the Fourth Industrial Revolution is at a historic turnaround. There were a lot of changes in the international shipbuilding industry that will be different compared to the previous three industrial revolutions. Some shipyards failed to follow the changes that were coming with the revolutions, but some shipyards were followed revolutionary changes and succeed to survive and make profit. The modern shipbuilding industry is expected to remain healthy in the future, especially in the market segments of higher added value and with larger sales value. All of these resulting on
the higher requirements for design, ship construction and operation which are crucial for success. The goal of Shipbuilding 4.0 is intelligent shipyard which is characterized by adaptability, resource efficiency and ergonomic but also close integration between shipowner and shipbuilder with the crucial cooperation shipyards - suppliers, the partners in the business and value processes, [18].

In the 2013 the International Maritime Organization (IMO) introduced regulation, energy efficiency design index (EEDI), the regulation that defines the energy efficiency standard for the new ships (IMO 2016). Meeting the cap required from the shipbuilders and operator significant care to fulfill the requirement; all ships built after 2025 will be at least 30% more fuel efficient, [17]. Eco-friendly shipping and the energy efficiency are now the key criteria for the construction of the new ships, [19]. This involves almost all systems on the ships; it requests the environmentally friendly shipbuilding and progressive supplier solutions.

A major transformation is underway in the shipbuilding sector; from the shipowners, and fleet operators, it is requested to order and develop more energy-efficient, reliable and environmentally friendly ships with better overall performance and lower operating costs. Most of them requests from the shipbuilders to meet these requests and their challenges before signing the contract. Shipbuilders have to design and built ships faster and better than ever before. This requires a totally different approach to ship design and construction. The shipbuilders need to meet the shipowner expectations to have rapidly modernized the fleet with the energy efficiency systems solutions on the ship.

Shipbuilding sector needs to improve total enterprise collaboration, synchronization and productivity, as well as the lifecycle ship maintenance and support by optimization of the shipbuilding process. During the design of the new vessel, there are several tasks that need to be accomplished: achieving greater performance, lower ownership cost, higher fleet availability and reliability, and greater quality and compliance with the latest marine safety and regulatory requirements. The ships need to be easier to build and repair; it needs to be lowered ship construction, service and total ownership costs.

Today, especially European shipyards mostly building special type of the vessels, usually unique and highly complex, with the special requirements in term of the quality, safety and design and production. In the production of the new unique vessel, need to be designed state-of-the-art tools and designed processes. New type shipbuilding supports design and manufacturing trends that are moving toward higher creativity, lower cost, and better respond to the shipowner need, while at the same time preparing the on-demand production of optimal and intelligent solutions, [18, 20].

To give the best results on the Internet of Things revolution, the shipbuilders will have verified and trusted suppliers, linked with shipyard and service personnel with production schedules and 3D models of all aspects of the design. Shipbuilders need to benefit from the new innovative solutions to provide immediate access to just the right relevant data; and service teams will benefit from this data to the appropriate supply chain to reduce service and overhaul cycle times. There are quite a few challenges arising out of this, namely the need for new approaches in collaboration and transaction support in terms of bridge-connected to suppliers and subcontractors.

In Shipbuilding 4.0 era, the shipyards need to have skilled engineers, specialists in technical science, but also in IT science and all relevant technical data from successful ship classes. In the design phase, it needs to be minimized the design period, but also the engineering costs of future classes of ships. This holistic solution spans the entire shipbuilding enterprise and lifecycle to enable shipbuilders to integrate their organizational knowledge,
 automate processes throughout the product lifecycle and improve efficiency, accuracy and execution to reduce time-to-delivery.

Several features need to be implemented in the Shipbuilding 4.0, [16]:

- Identification and classification of the shipbuilding standards,
- Internet networks,
- Wireless sensors,
- Software and hardware,
- Horizontal integration,
- Vertical integration,
- End-to-end integration,
- Smart services,
- Human-machine, interaction,
- Machine tool components,
- Safe and environmentally friendly production,
- Interoperability,
- Digital modeling technology,
- Virtualization,
- Visualization technology,
- Automation,
- Industrial internet,
- Cloud Computing,
- Big data,
- Flexibility (Plug-and-Work),
- Safety and Security (data privacy),
- Analysis of the practical solutions and industrial applications, still there is a lack of any,
- Training and Continuing Professional Development.

One thing is certain: the shipping industry is undergoing massive change. The “Digital Twin” concept is an example: The ability to reproduce ships digitally, and the use of drones fitted with high-tech cameras will reduce the effort involved in ship maintenance and mitigate safety risks", this was the words of Mr. Knut Ørbeck-Nilssen CEO of DNV GL Maritime on the Maritime Future Summit 2016 in SMM Hamburg. This was the first Summit, which gathers major players and decision makers in shipbuilding and shipping industry of the future. The mainly CEO’s were presented the latest themes and visions of the shipping and shipbuilding of the future of their companies and all of them are based on the idea of the Shipbuilding 4.0, Maritime 4.0, Shipping 4.0. All the speakers presented importance of the collecting and having data from the beginning of the design phase till ship delivery and operation in whole life cycle.

4. Shipyard 4.0 - design principles and strategy

The shipbuilding industry has a special attention of the governments, especially in the naval shipbuilding as the leader of the reorganizations and implementation improvements driven by the Shipbuilding 4.0 ideas. The sector of the Navy industry is the driver of development and innovations. The Shipbuilding 4.0 raises the question of how the shipyards need to be changed, to survive in the market. What added value can be offered to the shipowners, but also how should the
shipyards need to organize the work, in order to survive in the transitional period of transformation and adaptation to the Shipbuilding 4.0, but at the same time survive among shipyards competitors from the Far East and the rest of the Europe. The opportunity lies in the digitization of the shipyard. The key to success is the availability, exchange and processing of relevant data and information overall in the shipbuilding process. This involves one of the most important thing; it is collected Big Data, but also standards, norms, traceability of the process with clear guidelines and control. Full implementation is still a long away, but the Shipbuilding 4.0 is a change of the whole working world, it changes the human society with new job definition and working processes. For the shipyards the digitization is chance for surviving. The most important thing for the shipyard is efficient production processes which are the main competitive factor. Today shipbuilding is characterized by single units or small series production.

Goal to survive is further simplification of the production processes and continuous improvement of the production quality, integrated digitized design, efficient purchasing and lean logistic, high quality of welding processes with the modular production method which pushing upward shipyards. In the situation when the shipyards are preparing the new ship contract, it is very important to design innovative solutions together and in the closer cooperation with the selected bridge-connected suppliers.

For the overall project success is also crucial good cooperation between shipbuilders and shipowners, what brings several benefits: lean, flat point of contact, minimization of the coordination, conflicts and the disagreements are simplified, the risk of loss of the profit is reduced. [18]. With the new close cooperation between the shipbuilders and suppliers, in the very early design phase need to be defined main ship and production systems: energy and fuel saving solutions, main engines and propulsion systems, ship design and hull optimization, machinery, public spaces, technical areas, cabins, deck machinery, but also shipbuilding equipment, surface preparation, welding and preliminary building technology.

During the early definition of the project, the shipyards need to follow green shipping solutions, climate and environmentally friendly solutions and maritime safety technologies which require extensive maritime expertise and knowledge. In the Germany, as the leader of the new industrial revolution innovations, for the shipbuilding sector, the state budget has been increased to 25 million euro for the 2016 year, almost 7.5 million euro more than 2015 year, (Dry dock Magazine, 2016.).

The digitization of the design and ship construction makes the pillars of the overall digitization, which promotes Shipbuilding 4.0, but requires further development of:

- Process optimization,
- Standardization,
- Digital interconnectivity,
- Optimization of information flow,
- Interfacing the material management and information management within the entire supply chain.

The idea of the overall digitization in shipbuilding industry is visible on the Figure 3.
Availability of the collected digital data from basic design to ships operation in the digital Shipbuilding 4.0 will be essential. Discontinuities in the data flow will affect in loss of profit. Shipbuilders need to prepare the procedures provided by the suppliers to create a standardized, unified, coordinate infrastructure for design, production and maintain the ship. In case of lack of real-time collaboration opportunities for discovery of the design options that could improve a ship capability, accelerate its construction and reduce costs, [21].

5. Shipyards 4.0 - case study/implementation concept proposed methodology

The shipyards in the era of Shipyard 4.0 will use the power of algorithms to rely on their ability to predict the future in the digital world of big data, machine learning and predictive analytic. The shipyard needs to use the algorithms and simulations and their ability to derive from the mass of data produce information that humans can follow and understand. Generally speaking the algorithms allow having knowledge of the future results from the collected data, [21, 23].

The shipyards of the future constantly looking for the new ways to use virtual methods for engineering and commissioning and virtual support. The Smart Shipyard in the era of the Shipyard 4.0 via Cyber-Physical Systems will be organized as integrated, flexible, efficient, green, production process. The real time quality control in self-optimized and autonomous system, [23]. This is especially important for the shipyard there were turned into building special purpose vessels. In the contractual phase, there is a lot of the new requirements for the specific structural needs also adjusted and customized shipbuilding processes and operational conditions, [24]. Usually this period is always too short for the designers and is creating high cost pressure to high demands on the quality. Balancing this act will reinforce in the Shipyard 4.0 by promoting innovative and modern methods that will meet the challenges, [22].

One of the first published results of the Shipyard 4.0 implementation presented as a result of the cooperation between University of Coruña (UDC) and Navantia Spain Navy shipyard. With this cooperation has been designed the Smart Pipe System for monitoring, identification, traceability of the pipes used in the ship construction in the shipyard. The research result shows the implementation of the UHF Radio Frequency Identification (RFID) technology as the foundation of the CPS system in the Shipyard 4.0 era. The presented concept improved the
smarter energy consumption, efficient pipe logistic and shipyard production, usability of the shipyard facilities and real-time yield optimization, [16]. Some of the largest European shipyards from Italy, Spain, Germany, Finland started with the implementation of the Shipyard 4.0 concept. Implementation concept is different in every shipyard according to the present technological level and financial the possibility of the investment in the research and development in this concept.

In this article also have been analyzed the research done on the Faculty of Electrical Engineering, Mechanical Engineering and Naval Architecture in Split, 2016 year, named the project "INSEN - Croatian innovative company", [25]. The scientists from the Department of the Industrial Engineering, were researched, which degree of development have the Croatian industry, and what is the level of the industrial development in the respect of the Industry 4.0. This research is very important for the interviewed companies as potential future bridge-building suppliers of the Croatian shipyards. Beside the interviewed companies in the research were also included university experts. The results showed Croatian industrial companies on the level of the Industry 2.15; it is visible the long way for the Croatian industrial companies toward the Industry 4.0. The scientists from the Department of the Industrial Engineering started with the design of the "INSEN - Model of the innovative company", it represents the Smart, innovative factory adapted to the Croatian industrial environment, and the Croatian industrial way toward to Industry 4.0. With the solid foundation in the Croatian industrial environment dedicated to Industry 4.0, is the possible progress of the Croatian Shipbuilding Industry toward Shipbuilding 4.0.

If the Croatian shipyards want to become the digital shipyards of the future, very important role will be dedicated to the suppliers who will be included in the simulation modeling of the innovative ship system solutions. According to the concept of Shipyard 4.0, a modern shipyard will have an outsourcing ration of more that 80% of the contract value and there is a strong demand for sophisticated interfacing and integration of external sources into the shipyards own project management, [22].

In that respect the interconnection, digitization and integration of the whole industry will have a direct impact on this industrial revolution in Croatian shipbuilding industry.

On the "GALP - Green and Lead production conference" held in November 2017 in Zagreb has officially presented a government strategy, for the digitization of the Croatia industry. It was presented "Digitalizing Impulse 2020 - National platform of the Republic Croatia" with the strategy for digitization of the overall Croatian society, [26]. Today's reality is present as the CRO Improve Card 2017 visible on the Figure 4.

![CRO Improve Card 2017](image-url)

Fig. 4 CRO Improved Card 2017, [26].
Plan for digital transformation of the Croatian society and industry in period from 2017 to 2020 is visible on the Figure 5.

What brings about digital transformation and how the change business processes and model are presented on the "Digital Revolution Conference" organized by the Croatian Employers Association and sponsored by the Ministry of the Economy, held in February 2018 in Zagreb.

On the conference is presented the plan for the digitization of the Croatian society and the several key goals are presented:

1) Improving the regulatory framework for encouraging innovation and investment in the context of the EU’s single digital market,
2) Creating new business models for small and medium-sized businesses,
3) Encourage the development of research and innovation in digital technologies,
4) Developing gigabit networks as the core Internet infrastructure of Things,
5) Cybernetics and data security,
6) Enhance the education of digital skills for all ages,
7) The implementation of Industry 4.0 concepts in selected industry sectors.

This plan for the digitization of the Croatian industry have to include also the shipbuilding industry to stay in race forward Shipbuilding 4.0.

Most of the Croatian shipyards have turned into the building of the special purpose vessels. Croatia has long shipbuilding tradition, good geostrategic position, scientific and education system, and a lot of suppliers which can adopt their production system to the good and strong connection to the shipyards. According to the Croatian Shipbuilding Corporation, the shipbuilding sector is still one of the most important industrial sectors of the Republic of Croatia, by its share in the employment (2%-5% with the subcontractors up to 10%), by its GDP share (0.8%-1.8%) and by the exports (10%-15%), [27]. The Croatian Shipbuilding Corporation (CSC) held in Zagreb was established by the Croatian Government in 1994, as the cooperative organization covering the coordination of the Croatian Shipbuilding Industry on the international shipbuilding market. The main CSC’s tasks are to provide expert
monitoring of the restructuring process and modernization program of four largest Croatian shipyards:

"Uljanik" shipyard, "3. maj" shipyard, "Brodotrogir" shipyard and the "Brodosplit" shipyard.

The Croatian shipbuilding industry operates both on domestic and international market and export orientation is the dominant feature.

"Uljanik" shipyard is located in the Pula, founded 1856. From that time from the shipyard were delivered over 200 newbuildings. This shipyard designs, constructs and builds all types of the ships; from the ships for transportation of oil products, bulk cargo, containers, cars, passengers. Last technological rebuilding has been done in 1998. Since then there have been no comprehensive improvements in the shipyard technological production process. The industrial level is abt. 2.15. The number of the employees 2659.

"3. maj" shipyard is located in Rijeka, founded 1892 year. This shipyard designs, constructs and builds all types of the ships; from the ships for transportation of oil products, bulk cargoes, cars. Last technological rebuilding has been done in 1998. Since then there have been no comprehensive improvements in the shipyard technological production process. The industrial level is abt. 2.15. From 2013 year shipyard has become integrated in "Uljanik" shipyard group.

"Brodotrogir" shipyard is located on the island Ćiovo, by the Trogir, founded 1944 year. The product range includes tankers, floats, passenger ships, tugs and rescue vessels manufactured for the domestic market or exported. Last technological rebuilding has been done in 1998. Since then there have been no comprehensive improvements in the shipyard technological production process. The industrial level is abt. 2.15. It was privatized in 2013 by the Kermas Energija Company. The number of the employees 341.

"Brodosplit" shipyard, the largest Croatian shipyard is located in the Split, founded 1922 year. They have delivered about 350 vessels, with a total deadweight of over 9 million tons, including many tankers, both panamax and non-panamax, as well as container ships, bulk carriers, dredgers, and passenger ships. 80% of the ships built are exported to foreign contractors. Last technological rebuilding has been done in 1998. Since then there have been no comprehensive improvements in the shipyard technological production process. The industrial level is abt. 2.15. It was privatized in 2013 by the DIV Company. The number of the employees 2300.[27]

The companies analyzed in the article are the Croatian companies dedicated to the design and production of the ship equipment as being: deck machinery, diesel engines, transformers, electric motors, navigation system, loading computer, cranes. The industrial level of the companies are 2.15 still far from the level of the Fourth Industrial Revolution.

The branch of Industry represents a significant source of employment for the shipyards and the small and medium size companies, suppliers as the subcontractors with their operations and the services to the shipyard.

To fulfill all design and production requirement and regulations the new innovative solutions, the sector has to transform into "integrated digital shipyard". The pressure to balance high demands on the quality for lower cost, while on the other hand shipyard need to build the ship faster, better, safer with the lowest cost for the shipowners. So it has to be used with the latest advances (robotics, 3D printing, augmented reality of the Industrial Internet of the Things (IIoT)). This new concept has to turn to the shipowners needs, and it is very important that the shipyard is capable to enable the process with the dynamic engineering and flexible, adapting to errors cause by the design and production shipyard process or the suppliers.
The Croatian shipyard will adopt the Shipyard 4.0 model customized for the Croatian specific conditions.

The Shipyard 4.0 model merges the shipbuilding process with the enabling technologies as is:

- Robotic process automation,
- Virtual and augmented reality,
- Virtual modeling and simulation,
- 3D modeling, digital twin,
- Additive manufacturing,
- Big data and analytics,
- Ubiquitous connectivity and The Internet of Things,
- Secure cloud,
- Cyber security,
- Health, safety and environment,
- New materials,
- Artificial Intelligence,
- Autonomous vehicles.

Based on the available results of research and analysis of Croatian shipyards compared to the several European shipyards from Italy, Spain and Germany, Industrial level 3.2 (shipyard experts estimation), which are working on the implementation model of the Shipyard 4.0 concept; in order to achieve the proposed objectives, the following four preliminary phases of the Shipyard 4.0 implementation methodology to be applied, [28, 29]:

5.1. Phase 1: Definition of the problems and the implementation goals

This phase involves the introduction of the shipyard facilities, processes, material and information flow that will be prepared/evaluated, collected for the detail definition of the goals and deadlines.

1. Definition of the problems in the process and causes, it needs to be clearly defined what are the problems of the existing shipyard production process, what are the causes and what need to be improved,
2. Definition of the goals and level of the implementation that need to be achieved, clearly definition of the improvements, what is expected from the implementation of the new Shipyard 4.0 shipbuilding processes,
3. Definition of the responsibilities, deadlines; flow chart with clearly definition of the responsibilities and the implementation time frame deadlines.

As part of this phase are usable different types of the flowcharts, diagrams and modeling and simulation software tools. In the era of the digitized integrated shipyard of the future is visible usage of the huge number of the IT solution dedicated to the specific part of the production process integrated in the Big Data.

The four phases of the proposed implementation methodology are visible on Figure 6.
Fig. 6 Proposed methodology for the Shipyard 4.0 implementation model
5.2. Phase 2: Definition of the input data and conceptual implementation modeling

The main objectives of this phase are the preliminary design of an implementation model according to the collected relevant input data, evaluation of the conceptual implementation model.

The tasks of this phase are:

1. Definition and preparation of the input data and preliminary implementation model;
2. The preliminary design of the implementation model for the future shipbuilding process, it is started with the design of the digital implementation model for the new shipbuilding process with the selected implementation modeling tools,
3. Evaluation of the alternatives as a result of the developed implementation modeling.
4. Selection of the implementation model/solution.

Method and tools used in this phase is CAD, 3D modeling tools, simulation modeling tools, PLM, flowchart tools.

5.3. Phase 3: Establishment and verification the implementation model

The main objectives of this phase are the establishment and verification of the implementation model of the new shipbuilding process.

The tasks of this phase are:

1. Identification and systematization of the collected data, evaluation of the sorted selected data,
2. Establishment of the selected data and preparation of the implementation tools,
3. Verification of the collected, selected data in the implementation model,
4. Definition of the constraints; elimination of the errors from the implementation model.

Method and tools used in this phase is CAD, 3D tools, simulation modeling tools, PLM, flowchart tools, methods of the decision making.

5.4. Phase 4: Evaluation / analysis and improvements of the selected optimal implementation model.

The main objectives of this phase are the evaluation, improvement of the selected implementation model of the new shipbuilding process.

The tasks of this phase are:

1. Evaluation and analysis of the selected optimal implementation model,
2. Improvements of the selected optimal implementation model,
3. Recalculation and recheck of the functionality of the selected model,
4. Application of the selected implementation model.

The results of the proposed methodology will be visible after shipyard case study implementation as the further research.

Preliminary assessment of the time frame for the proposed methodology application in the one of the Croatian shipyard presented in the Figure 7.
Fig. 7 Preliminary assessment, implementation time frame

Preliminary assessment concept items to be applicable in the case study shipyard:

1. Preparation of the shipyard and the shipbuilding standards necessary for the safe flow of data and information,
2. Preparation the Value Stream Mapping of the production process, the Lean management. The shipyard needs to use resources, in the less time with the higher productivity, to have satisfied customer and more satisfied employees. During this production process the analyzed shipyard has already made some steps toward savings the electric energy. Some workshops use the sensors for the automatic turning down the power during the third shift and during the weekends, [28],
3. Strongly raising the level of the knowledge and skills of the employees to work in the digitized industry. As it is currently happening in the car industry, there is a need for the highest level of formal university education. The future shipyard will request to have technical experts with the very good IT skills,
4. Overall, networking and digital connectivity through the whole ship building process from design, purchasing, planning, financing, production by using wireless technologies. Shipyard need to develop new adopted model that will be capable to operate in the digital shipbuilding industry,
5. Safety and security of the exchanged data and information. Digital integration of the overall shipbuilding sector; integration of the shipyards, suppliers, operators, classification societies and port operators. In the interconnected integrated shipyard is crucially had the safe and in time exchanging information.

6. Conclusion

Today, the whole world is working on the fourth industrial revolution, Industry 4.0. All industries searching the possibilities for changes and improvements, so as the Croatian industry with the strategic program for implementation of the digitization of the society and industry. In 2015 year, almost all huge players in the shipbuilding industry started to research the methods for the implementation of this new revolution as the Shipbuilding 4.0.
In the article was presented review of the Industry 4.0 and the way how it is changing the shipbuilding industry as the Shipbuilding 4.0. It was analyzed overall changes in the world shipbuilding industry and finally the changes and preliminary methodology for the implementation of the Shipyard 4.0 implementation model in the Croatia case study shipyard.
For the further research it will be analyzed application of the proposed methodology in the selected Croatian shipyard.

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Nomenclature

The following abbreviations are used in manuscript:
BD - Big Data
CC - Cloud Computing
CPS - Cyber-Physical Systems
CPPS - Cyber- Physical Production System
DT - Digital Twin
IIT - Industrial Internet Consortium
IoS - Internet of Services
IoT - Internet of Things
IVI - Industrial Value Chain Initiative
VDMA - Verband Deutscher Maschinen- und Anlagenbau
5C - Connection, Conversation, Cyber, Cognition and Configuration

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THE PRELIMINARY DESIGN OF A SCREW PROPELLER BY MEANS OF COMPUTATIONAL FLUID DYNAMICS

UDC 629.5.016: 629.5.035
Original scientific paper

Summary

The assessment of hydrodynamic characteristics of a screw propeller in open water test is of crucial significance in the preliminary design stage of a ship. The open water characteristics can be accurately predicted by towing tank measurements. Taking into account the significant advances in computational fluid dynamics which has enabled the numerical assessment of the open water characteristics, the time and the cost of propeller design are significantly reduced. Open water characteristics can be assessed numerically using potential and viscous flow theory. The former one allows faster and simpler preliminary determination of open water characteristics. Within this paper, numerical simulations are performed for Gawn series propellers utilizing open source code OpenProp based on the moderately loaded lifting line theory and commercial software package STAR-CCM+ based on viscous flow theory. The latter one is more time consuming regarding the time required for the preparation of simulation as well as computational time. The obtained numerical results are compared with regression polynomials based on the experimental data. The validation of the results has pointed out that OpenProp can be used as practical and efficient tool in preliminary design of screw propellers.

Key words: screw propeller; open water test; potential flow theory; viscous flow theory

1. Introduction

The most important phase in the design of a screw propeller is the preliminary design phase during which the geometry of the screw propeller is chosen, in order to ensure the required thrust at a certain ship speed. Nowadays, the hydrodynamic characteristics of a screw propeller are most commonly determined utilizing experimental methods, i.e. open water test (OWT) is performed [1]. In the past different approaches based on the potential flow theory have been developed for modelling of the flow around a screw propeller. Numerical methods for the determination of the hydrodynamic characteristics of a screw propeller are according to the order of complexity [2]: momentum theory, lifting line method, lifting-surface method, Boundary Element Method (BEM) or panel method and Reynolds Averaged Navier-Stokes
(RANS) equations. Numerical simulations based on the potential flow theory provide a rapid estimation of the hydrodynamic characteristics of a screw propeller in OWT and therefore are of great benefit to ship designers. The lifting line method is the fastest and easiest to apply, as it does not require a complex preparation of the three-dimensional representation of a screw propeller. Because of its computational efficiency, it has been a key factor for a number of years in the preliminary design of a screw propeller [3]. On the other hand, methods for the determination of the hydrodynamic characteristics of a screw propeller based on the viscous flow theory are used as well. With the development of computers in last years these methods have taken an increasingly important role in screw hydrodynamics.

The possibilities of improving the geometry of a screw propeller at the preliminary design stage using methods based on the potential flow theory are presented in [4]. Validation of the results was performed on DTMB 4119 and DTMB 4381 screw propellers for which the experimental data were available. The proposed method showed that the geometry of the DTMB 4119 screw propeller was optimal. Furthermore, geometric improvements had been achieved by the rearrangement of the pitch along the blade of a screw propeller by applying the same method for the DTMB 4381 screw propeller. In this way, a more favorable distribution of circulation along the blades of a screw propeller was achieved. Gaggero et al. [5] have performed the geometry optimization of the Contracted and Tip Loaded (CLT) propeller using BEM. The reliability of the design method and achieved improvements in the screw propeller geometry had been validated by extensive RANS calculations. The authors also performed unsteady calculations using unsteady BEM to obtain amplitudes of induced pressure impulses. The obtained results were compared with the available measurements and with results for the geometry of original CLT propeller and the authors confirmed the improvements for the new geometry. Bertetta et al. [6] have investigated the influence of the cavitation on the noise for a Controlable Pitch Propeller (CPP) outside the operating point using BEM. The multiobjective optimization algorithm coupled with BEM led to the optimum geometry of CPP for different pitches with the aim of reducing cavitation and noise. Outside the operating point of the screw propeller, the new geometry produces less noise and less pressure impulses than the original geometry. Lee et al. [7] have proposed an optimization code coupled with the lifting-surface method for the design of a screw propeller. The method uses a vortex theory to calculate the induced speeds at certain radii of a screw propeller. The validation of the results was carried out on DTNSRDC 4119 propeller with a homogeneous velocity field as well as on a series of DTNSRDC propellers that had more complex geometry. A study of the hydrodynamic characteristics of a screw propeller, operating near the free surface, for different advance coefficients and various submergence of propeller models was performed in [8]. The impact of the scale effects was evaluated by testing two models in different scale. The results of the numerical simulation included the free surface pattern, the velocity field around the screw propeller, and the hydrodynamic characteristics of the screw propeller. Islam et al. [9] have studied the influence of domain size and discretization parameters on the time required to run RANS simulation of OWT in the commercial software package STAR-CCM+ as well as their impact on the accuracy of the results. Based on the results for 32 different domains for the same screw propeller, the authors found the optimum domain dimension and domain discretization parameters utilizing design of experiments. The obtained results were validated by comparison with the experimental data. Califano and Steen [10] have performed RANS simulations using the commercial software package Fluent to investigate the effects of ventilation on the hydrodynamic
characteristics of a screw propeller while operating in the heavy sea. The authors concluded that predicted dynamic loads utilizing numerical simulations showed satisfactory agreement with the experimentally obtained ones, but only at the upright position where the blade is piercing the free surface. For the other angular positions the thrust is overestimated. This was attributed to the inability of RANS solver to resolve the tip vortex. Furthermore, it was shown that the tip vortex had a very important role in the ventilation of conventional propellers, which was not a case for surface-piercing propellers. Subhas et al. [11] have carried out a numerical simulation of OWT using RANS solver in Fluent for the prediction of the pressure field and the velocity around the screw propeller, as well as the cavitation occurrence. The comparison of the obtained results with experimental ones showed that Computational Fluid Dynamics (CFD) can be used for prediction of the cavitation occurrence. Prakash and Nath [12] have performed a numerical simulation of OWT for a four-blade propeller of Wageningen B series in Fluent utilizing unstructured grid. The validation of the obtained results was performed by comparison with the results obtained using regression polynomials which were derived based on experimental results. Krasilnikov et al. [13] have investigated the scale effects utilizing RANS simulations with the main focus on the impact of blade skew, propeller loading and blade area ratio. Müller et al. [14] have analyzed the impact of the scale effects on screw propeller for large container ships. The authors performed the numerical simulations of OWT within commercial software package CFX for 23 screw propellers in model scale and full-scale. Based on the obtained results, they proposed a scaling method for evaluation of OWT characteristics in full-scale. Lee and Paik [15] have performed numerical simulations of partially submerged screw propeller under the bollard condition. The authors showed that thrust and torque of partially submerged propeller decrease significantly with an increase in rate of revolution. In [16], the author has investigated wake characteristics of Contra Rotating Propeller (CRP) in OWT and self-propulsion conditions. In addition, Paik studied the effect of rudder on wake characteristics in self-propulsion test and concluded that the presence of the rudder has no significant influence on wake whereas a significant influence on thrust and torque was noticed. Kinaci and Gokce [17] have studied effect of propeller on free surface elevations for benchmark Post Panamax ship, Duisburg test case. For this purpose, the authors performed numerical simulations of resistance and self-propulsion tests. The authors concluded that free surface causes the increase of pressure on the propeller and hull stern as well as the efficiency of the propeller.

In this paper OWT characteristics are studied numerically utilizing lifting line theory and RANS. Numerical simulations are performed for five Gawn series propellers with expanded area ratio equal to 1.1. Pitch to diameter ratio of propellers with three blades is varied in the range from 0.7 to 0.9. The obtained numerical results are compared with regression polynomials based on the experimental data. Furthermore, the flow around the screw propeller in OWT is analyzed. This paper is organized as follows: Section 2 provides governing equations for lifting line and viscous flow theory, while in Section 3 numerical setup is given. Section 4 provides the obtained results, while in Section 5 conclusions drawn from this research are given.
2. Governing equations

2.1 Lifting line theory

This subsection presents the theory of moderately loaded lifting line enhanced with vortex lattice for computing the induced velocities. Propeller blade is represented by a lifting line, with trailing vorticity aligned to the local flow velocity. The induced velocities are computed using a vortex lattice theory, with helical trailing vortex filaments shed at discrete stations along the blade. The blade itself is modelled as discrete sections, having 2D section properties at each radius. Loads are computed by integrating the 2D section loads over the span of the blade [18].

Using the lifting and viscous forces for infinitely many radial sections along the propeller blades, i.e. knowing the dependence of these two forces on the radial position along the propeller blades ($r$) and under the assumption of identical propeller blades, the total thrust and torque can be calculated as follows [18]:

$$T = Z \int_{r_h}^{R} \left[ F_i \cos(\beta) - F_v \sin(\beta) \right] dr (e_a)$$

(1)

$$Q = Z \int_{r_h}^{R} \left[ F_i \cos(\beta) + F_v \sin(\beta) \right] r dr (-e_a)$$

(2)

where $Z$ is the number of blades, $F_i$ is the magnitude of lifting force, $F_v$ is the magnitude of viscous force, $\beta$ is the angle of the resultant inflow velocity ($V^*$), $e_a$ is the axial direction, $r_h$ is the propeller hub radius and $R$ is the radius of the propeller. Figure 1 shows the forces and velocities acting on the blade section.

![Fig. 1 Forces and velocities acting on the blade section, [18]](image)

The magnitudes of lifting and viscous forces can be determined as follows:

$$F_i = \rho V^* \Gamma$$

(3)
\[ F_r = \frac{1}{2} \rho \left( V^* \right)^2 C_D c \]  

(4)

where \( \rho \) is the fluid density, \( V^* \) is the total resultant inflow velocity, \( \Gamma \) is the magnitude of circulation, \( C_D \) is the section drag coefficient and \( c \) is the section chord. The section drag coefficient \( C_D \) is determined by means of charts for known hydrodynamic profile and Reynolds number calculated on the basis of the chord length.

The circulation is computed from 2D lift coefficient, which is given as follows:

\[ C_L = \frac{2 \Gamma}{V^* c} \]  

(5)

A standard propeller vortex lattice model is used to compute the axial and tangential induced velocities \((u_a^*, u_t^*)\). In the vortex lattice formulation, a propeller with \( Z \) blades is modelled as a single representative radial lifting line, partitioned into \( M \) panels. The induced velocities are computed at control points on the lifting line at radial locations, by summing the velocity induced by each horseshoe vortex as follows:

\[ u_a^*(m) = \sum_{i=1}^{M} \bar{u}_a^*(m,i) \Gamma(i) \]  

(6)

\[ u_t^*(m) = \sum_{i=1}^{M} \bar{u}_t^*(m,i) \Gamma(i) \]  

(7)

where \( \bar{u}_a^* \) and \( \bar{u}_t^* \) are the axial and tangential velocity induced at the certain radial location by a unit-strength horseshoe vortex surrounding panel \( i \).

Lifting line theory does not take into account the induced velocities. Therefore, the vortice lattice theory is used for the calculation of the induced velocities, \( \bar{u}_a^* \) and \( \bar{u}_t^* \) as follows:

\[ \bar{u}_a^*(m,i) = \bar{u}_a(m,i) - \bar{u}_a(m,i + 1) \]  

(8)

\[ \bar{u}_t^*(m,i) = \bar{u}_t(m,i) - \bar{u}_t(m,i + 1) \]  

(9)

where \( \bar{u}_a(m,i) \) and \( \bar{u}_t(m,i) \) are axial and tangential velocities induced at the certain radial locations by a unit-strength helical vortex filament at the panel end point with the vector direction of the circulation approaching the lifting line by right-hand rule. These components are calculated using formulae by Wrench (1957), [18].

More details regarding the lifting line theory can be found within [18, 19].

2.2 Viscous flow theory

In this subsection, basic physical laws for the description of the incompressible viscous flow along with the equations for Moving Reference Frame (MRF) method are given.

The law of conservation of mass in differential form is defined as [20]:

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0 \]
The preliminary design of a screw propeller by means of computational fluid dynamics

\[
\frac{\partial \bar{u}_i}{\partial x_i} = 0
\]  \hspace{1cm} (10)

Reynolds-Averaged Navier Stokes equations for incompressible flow are given as [20]:

\[
\frac{\partial \bar{u}_i}{\partial t} + \frac{\partial}{\partial x_j} \left( \bar{u}_i \bar{u}_j + u'_{ij} \right) = -\frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_i} + \frac{1}{\rho} \frac{\partial \tau_{ij}}{\partial x_j}
\]  \hspace{1cm} (11)

where \( \bar{u}_i \) is the time averaged velocity vector, \( u'_{ij} \) is the Reynolds stress tensor divided with \( \rho, \bar{p} \) is the time averaged pressure and \( \tau_{ij} \) is defined as follows:

\[
\tau_{ij} = \mu \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right)
\]  \hspace{1cm} (12)

In order to close an unclosed set of equations (10) and (11), Shear Stress Transport \( k - \omega \) (SSTKO) turbulence model is introduced. This model includes modification for the influence of low Reynolds numbers. SSTKO is an empirical model, where one equation includes turbulent kinetic energy (\( k \)) representing the velocity scale, and the other equation takes into account the dissipation rate (\( \omega \)) representing the length scale.

Moving Reference Frame (MRF) method is utilized within numerical simulations of OWT, which are performed as steady simulations. OWT is carried out for each advance coefficient \( J \), whereby the rate of revolution of a screw propeller is kept constant and advance speed is varied. In the case of MRF, the governing equations are solved together with additional acceleration terms. The computational domain is divided into stationary and rotational part. From the stationary point of view, the absolute velocity and the relative velocity are related with the equation:

\[
u_{ui} = u_{pi} + \varepsilon_{ijk} \omega_{pj} r_k
\]  \hspace{1cm} (13)

where \( \omega_{pj} \) is the angular velocity and \( r_k \) is the position vector from the origin of the MRF to the center of a control volume.

The law of conservation of mass and RANS equations in MRF written with relative velocity are as follows [21, 22]:

\[
\frac{\partial \bar{u}_{pi}}{\partial x_i} = 0
\]  \hspace{1cm} (14)

\[
\frac{\partial \bar{u}_{pi}}{\partial t} + u_j \frac{\partial \bar{u}_{pi}}{\partial x_j} + \varepsilon_{ijk} \frac{d \omega_{pj}}{dt} r_k + 2 \varepsilon_{ijk} \omega_{pj} u_{pk} + \varepsilon_{ijk} \varepsilon_{jlm} \omega_{pl} \omega_{pm} r_k = -\frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_i} + \nu \frac{\partial^2 \bar{u}_{pi}}{\partial x_j \partial x_j}
\]  \hspace{1cm} (15)

3. Numerical setup

In this section numerical setup used within numerical simulations of potential and viscous flow around a screw propeller is presented.
3.1 Lifting line theory

Numerical simulations of potential flow are performed within open source code, OpenProp. Input data for the simulation are: profiles, chord length and pitch to diameter ratio of a screw propeller at different radii and maximum thickness distribution along the blade. In Figure 2, three-dimensional representation of the Gawn series propeller obtained from OpenProp is shown.

![3D representation of the Gawn series propeller](image)

Geometry of Gawn series propeller is taken from [23]. Gawn series propellers have a uniform face pitch, segmental blade sections, constant blade thickness ratio and zero skewness. Within OpenProp, a text file is created which contains the position of the segmental blade sections in 3D in nondimensional form normalized with $D$. The shape of the hub is assumed to be cylindrical with diameter equal to 0.2 $D$. Text files are created for five propellers with the same expanded area ratio equal to 1.1 and with pitch to diameter ratio in the range from 0.7 to 0.9 with step of 0.05. Numerical simulations are performed for a range of advance coefficients from 0.1 up to advance coefficient for which the thrust is negative. It should be noted that at higher $P/D$ ratios, i.e. for 0.85 and 0.9, at lower advance coefficients screw propeller is more loaded. Therefore, for these two propellers, numerical simulations are performed from $J=0.15$ and $J=0.2$ respectively, since within OpenProp moderately loaded lifting line approach is used. Equations (1) and (2) are solved in a discrete form within OpenProp.

3.2 Viscous flow theory

Numerical simulations of viscous flow are performed within commercial software package, STAR-CCM+. Governing equations are discretized using finite volume method, which are solved in a segregated manner. All numerical simulations are performed as steady simulations for several advance coefficients in range from 0.1 to 0.75 with a step of 0.05. The change of advance coefficient is achieved by variation of the inlet velocity, while rate of revolution is kept constant. The simulations for each advance coefficient are stopped after 4000 iterations when residuals drop at least four order of magnitude, i.e. iteration uncertainty is negligible, Figure 3. The computational domain, shown in Figure 4, is discretized using
unstructured hexahedral mesh with following meshing tools: Prism layer mesher, Trimmer and Surface remesher according to the recommendations from [9].

Numerical simulations are performed for five Gawn series propellers of same expanded area ratio equal to 1.1, which differ in pitch to diameter ratio ranging from 0.7 to 0.9. The mesh for all five propellers has around 2.5 million cells. The mesh is refined through entire domain within the diameter equal to 1.4 $D$, and vertically near the propeller in the region equal to 1.4 $D$, as can be seen in Figure 5. The refinement in rotation region can be seen in Figure 6.
The special care is given to near wall treatment, where near wall cells are generated taking into account that $y^+$ of the first cell near wall has value above 30, since wall functions are applied, Figure 7. It is important to note that underrelaxation factor for velocity is set to 0.5 and for pressure to 0.1.

The boundary conditions are applied as follows: velocity inlet for inlet boundary, pressure outlet for outlet boundary, slip wall for lateral surface of the cylinder and no slip wall for the propeller.
4. Results

Within this section, the results of OWT obtained utilizing lifting line method and RANS equations are presented. Furthermore, the viscous flow around the screw propeller in OWT is analysed. The obtained numerical hydrodynamic coefficients in OWT are compared with regression polynomials presented in [23], which are obtained from the experimental measurements as follows:

\[ K_T = \sum_{n=1}^{39} C_n \cdot J^s \cdot \left( \frac{P}{D} \right)^s \cdot \left( \frac{A_e}{A_0} \right)^u \cdot Z^v \]  \hspace{1cm} \text{(16)}

\[ K_Q = \sum_{n=1}^{47} C_n \cdot J^s \cdot \left( \frac{P}{D} \right)^s \cdot \left( \frac{A_e}{A_0} \right)^u \cdot Z^v \]  \hspace{1cm} \text{(17)}

where \( C_n, s, t, u, v \) are coefficients given in [23], \( K_T \) is the thrust coefficient, \( P/D \) is the pitch to diameter ratio, \( A_e/A_0 \) is the expanded area ratio and \( K_Q \) is the torque coefficient.

The open water efficiency can be calculated as follows:

\[ \eta_O = \frac{J}{2\pi K_Q} \]  \hspace{1cm} \text{(18)}

OWT diagrams obtained utilizing lifting line theory and RANS simulations are compared with regression polynomials, Figures 8-12. It can be noticed that satisfactory agreement between numerical results obtained using viscous flow theory and regression polynomials is achieved. Differences in \( K_T \) and \( 10K_Q \) obtained using RANS and with regression polynomials are relatively low. However, as results for \( K_T \) obtained with RANS underestimate the solution obtained by regression polynomials and results for \( 10K_Q \) obtained with RANS overestimate the solution obtained by regression polynomials, the obtained
differences in $\eta_0$ are even more pronounced. Results of numerical simulations based on the lifting line theory have larger deviations from regression polynomials for thrust and torque coefficients at lower values of $J$. The reason for this is that propeller is more loaded at lower $J$ values. Mathematical model in OpenProp uses an assumption of moderately loaded propeller blade and thus the radial component of induced velocity is neglected. However, heavily loaded propellers have significant value of induced radial velocity and for that reason OpenProp can be used for simulations in the range of $J$ where the propeller is moderately loaded. Trends of hydrodynamic coefficients obtained with RANS simulations are the same as the ones obtained using regression polynomials, while the ones obtained with lifting line theory show significant deviations at lower $J$ values. However, in the range of $J$ where propeller is moderately loaded, trends of hydrodynamic coefficients obtained with numerical simulations based on lifting line theory and regression polynomials are the same. As the most of screw propellers operate in the range of $J$ where propeller is moderately loaded, it can be concluded that OpenProp can be used for preliminary design of a screw propeller.

![Fig. 8 OWT diagram for P/D=0.7](image1)

![Fig. 9 OWT diagram for P/D=0.75](image2)
Fig. 10 OWT diagram for $P/D=0.8$

Fig. 11 OWT diagram for $P/D=0.85$

Fig. 12 OWT diagram for $P/D=0.9$
Tables 1-5 show results of OWT obtained by OpenProp (potential flow), STAR CCM+ (RANS) and regression polynomials for different pitch to diameter ratio.

### Table 1: OWT results for P/D=0.7

<table>
<thead>
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<th>$K_T$-CFD</th>
<th>$K_T$-OpenProp</th>
<th>$10K_{\phi}$-EXP</th>
<th>$10K_{\phi}$-CFD</th>
<th>$10K_{\phi}$-OpenProp</th>
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### Table 2: OWT results for P/D=0.75

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### Table 4: OWT results for P/D=0.85

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The preliminary design of a screw propeller by means of computational fluid dynamics

Deni Vlašić, Nastia Degiuli, Andrea Farkas, Ivana Martić

Table 5: OWT results for P/D=0.9

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<td>0.0899</td>
<td>0.165</td>
<td>0.046</td>
<td>0.147</td>
</tr>
</tbody>
</table>

The pressure field on the pressure side of the propeller for two values of $J$ is shown in Figure 13 for propeller with $P/D=0.8$. It can be noticed that on the pressure side of the propeller for higher $J$, the overpressure is lower since the load on the propeller is lower. The pressure field on the suction side of the propeller with $P/D=0.8$ for two same values of $J$ is shown in Figure 14. For higher $J$, the underpressure on the suction side of the propeller is reduced. The reason for this is an increase in inflow velocity and decrease of the load on the propeller. Since approximately two-thirds of the thrust are caused by the underpressure on the suction side of propeller blades [1], the thrust is lower at higher values of $J$.

![Fig. 13 Pressure distribution on the pressure side of the screw propeller at $J=0.2$ (left) and $J=0.55$ (right)](image-url)
In Figure 15, the pressure field of propeller with $P/D=0.8$ in the vertical plane is shown. The larger difference in the pressure between the pressure and suction side of the propeller can be noticed at lower $J$ value, which is caused by larger load on the propeller blades.
The preliminary design of a screw propeller by means of computational fluid dynamics

Deni Vlašić, Nastia Degiuli, Andrea Farkas, Ivana Martić

The velocity distribution in the vertical plane of propeller with $P/D = 0.8$ for two values of $J$ is shown in Figure 16. The larger difference in velocity distribution is present at lower $J$ value due to larger thrust at lower $J$ value.

The streamlines behind operating propeller with $P/D = 0.8$ for two values of $J$ are shown in Figure 17. A larger trailing vortex can be noticed for lower $J$ value and therefore the open water efficiency is lower due to the loss of kinetic energy.

![Figure 17 Streamlines behind operating propeller at $J = 0.2$ (left) and $J = 0.55$ (right)]

5. Conclusion

In this paper the numerical simulations based on potential and viscous flow theory were performed for five Gawn series propellers. The obtained numerical results of OWT were compared with the available regression polynomials. An input for numerical simulations based on the lifting line theory were the discrete values of the chord length of the blade sections at different radii. Within this research the applicability of lifting line theory in preliminary design of a screw propeller was investigated. Trends of hydrodynamic coefficients obtained using lifting line theory and regression polynomials are the same for the range of $J$ values where the propeller is moderately loaded, which is of great importance since the most of screw propellers operate in this range.

3D models for five Gawn series propellers were generated and used as an input data for RANS simulations. It was shown that trends of hydrodynamic coefficients obtained using RANS simulations and regression polynomials based on the experimental data are the same. Also, the detail analysis of the flow around screw propeller in OWT was performed. RANS simulations provide a complete insight into the flow around screw propeller, i.e. pressure and velocity distributions as well as streamlines. This is a valuable benefit of RANS simulations compared to simulations based on the lifting line theory.

The satisfactory agreement between numerically obtained results and the ones obtained using regression polynomials was achieved for both lifting line and viscous flow theory in the range of $J$ values where the propeller is moderately loaded. Therefore, OpenProp can be used as a practical and efficient tool in preliminary design of screw propellers as simulations based on the lifting line theory are significantly faster and simpler than RANS simulations. However, once the optimal geometry is obtained using lifting line theory, RANS simulations
should be performed in order to obtain detail insight into the flow phenomena around the propeller.

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WAVE INDUCED COUPLED MOTIONS AND STRUCTURAL LOADS BETWEEN TWO OFFSHORE FLOATING STRUCTURES IN WAVES

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Summary

As oil or gas field moves deeper offshore area, offshore offloading operations such as Tandem or Side-by-Side arrangement between two floating structures take place in many locations throughout the world and also have many hydrodynamic problems. Therefore, the researches on the motion response and hydrodynamic force including first and second order between two floating structures are needed to have the more safe offloading operability in waves. In this paper, prediction of wave induced motion responses and structural loads at mid-ship section with hydrodynamic interaction effect between two offshore floating structures in various heading waves are studied by using a linearized three-dimensional potential theory. Numerical calculations using three-dimensional pulsating source distribution techniques have been carried out for hydrodynamic pressure distribution, wave exciting force, twelve coupled linear motion responses, relative motions and wave loads of the barge and the ship in oblique waves. The computational results give a good correlation with the experimental results and also with other numerical results. As a result, the present computational tool can be used effectively to predict the wave induced motions and structural loads of multiple offshore floating structures in waves.

Key words: coupled motion; hydrodynamic interaction; relative motion; structural loads;

1. Introduction

Until now, offshore offloading operations take place in many locations throughout the world. In general, many offshore operations involve the use of two or more floating structures, which are positioned closely to transfer oil or gas during offloading such as oil FPSO and shuttle tanker, and LNG-FPSO and LNG carrier. So they affect each other's motion responses and wave loads through hydrodynamic interaction between two bodies in waves. Consequently, the large motions and wave loads between two floating bodies, which would cause the down time of offloading system failure and damage of ship hull by collision, etc. Because of these serious
problems during offloading operations, it is very important to study the motion behaviors and wave loads between two offshore floating structures due to the hydrodynamic effect in waves.

However, an experiment of motions and wave loads between two floating structures are very expensive and time-consuming work to design the offshore floating structures in the basic design stage. Therefore, the development of theoretical and numerical prediction program for hydrodynamic pressures, wave exciting forces, motion responses and wave loads between two floating structures are necessary to prudent design for multiple offshore floating structures.


In this paper, we describe the wave induced motion and first and second order structural loads prediction program which have been developed by Kim and Ha [13], Ha and Kim [14, 15] and Kim [16] using three-dimensional linearized potential theory and three-dimensional source distribution technique. The wave loads at a particular ship cross section can be obtained by the integration of external hydrodynamic forces and mass inertia forces on one side of the cut. In order to validate the developed theoretical and numerical calculation program between two floating bodies, the comparisons are performed for Kodan [2]'s experimental and 2-D results, Fang and Kim [5]'s 2-D results and Fang and Chen [6, 7]'s 3-D results for barge and ship, and Pinkster [20]’s experimental results for rectangular barge, respectively.

2. Theoretical Background and Mathematical Formulation

To describe the motions responses, the first and second order wave loads between two floating structures in waves, we consider 3 sets of right-handed orthogonal coordinate systems as shown in Fig. 1. O-XYZ is the space fixed coordinate system. \( O_A-X_AY_AZ_A \) and \( O_B-X_BY_BZ_B \) are the oscillatory coordinate systems fixed with respect to mean position of ship A and ship B, respectively. The O-XY plane coincides with the undisturbed free surface, the X-axis in the direction of the body’s forward and the Z-axis vertically upward.

The position vector of a point on the hull of the body relative to the body fixed system axes can be written as follows:

\[
\vec{\alpha} = \vec{\xi} + R_1\vec{r} + R_2\vec{r}' + O(\varepsilon^3) \tag{1}
\]

where \( \vec{\alpha} \) represent a small oscillatory displacement and \( R_1 \) and \( R_2 \) are the transformation matrices due to the translational and rotational movement of the body, respectively.
The Oscillatory coordinate systems $O_A\cdot X_A Y_A Z_A$ and $O_B \cdot X_B Y_B Z_B$ are used to describe the body motion in six degrees of freedom with complex amplitudes $\xi_j (j=1,2,\ldots,12)$. Here $j=1,2,3,4,5,6$ represent surge, sway, heave, roll, pitch and yaw for ship A, respectively and $j=7,8,9,10,11,12$ represent surge, sway, heave, roll, pitch and yaw for ship B, respectively as shown in Fig. 1.

The total unsteady potential for a sinusoidal wave excitation with encounter frequency, $\omega_o$, can be expressed as

$$
\phi^{(1)}(x, y, z, t) = \left[ \phi_i^{(1)} + \phi_p^{(1)} + \sum_{j=1}^{12} \xi_j^{(1)} \phi_j^{(1)} \right] e^{-i\omega_o t}
$$

(2)

where $\phi_i^{(1)}$ is the incident wave potential represent the incident waves; $\phi_p^{(1)}$ is the diffraction potential represent the disturbance of the incident waves diffracted from the body; $\phi_j^{(1)}(j=1,2,\ldots,12)$ represent the radiation potentials due to oscillations of the two floating bodies in calm water with unit amplitude.

The incident wave potential was given as follow

$$
\phi_i^{(1)} = -\frac{ig}{\omega_o} \zeta e^{kz} e^{i(k\cos\beta + k\sin\beta)} e^{-i\omega_o t}
$$

(3)

where $\omega_o$ is the wave frequency, $\zeta$ the wave amplitude, $k$ the incident wave number, $k=\omega_o^2/g$, and $\beta$ an arbitrary heading angles ($180^\circ$ for head sea).

The individual potentials have to satisfy in the fluid domain, on the free surface, the submerged body surface, the sea bed and a suitable far-field radiation condition at infinity.

**Laplace equation**

$$
\nabla^2 \phi^{(1)} = 0\quad \text{in the fluid domain}
$$

(4)

**Linear free surface condition**
\[- \omega^2 e \phi^{(1)} + g \frac{\partial}{\partial z} \phi^{(1)} = 0 \quad \text{on } z = 0 \quad (5)\]

**Body boundary condition for diffraction potentials**

\[\frac{\partial}{\partial n} (\phi_i^{(1)} + \phi_D^{(1)}) = 0 \quad \text{on ship A and ship B} \quad (6)\]

**Body boundary condition for radiation potentials**

\[
\frac{\partial}{\partial n} \phi_j^{(1)} = -i \omega_e n_j \quad (j = 1, 2, ..., 6) \quad \text{on ship A} \quad (7a)
\]

\[
\frac{\partial}{\partial n} \phi_j^{(1)} = 0 \quad (j = 7, 8, ..., 12) \quad \text{on ship B} \quad (7b)
\]

\[
\frac{\partial}{\partial n} \phi_j^{(1)} = 0 \quad (j = 1, 2, ..., 6) \quad \text{on ship A} \quad (8a)
\]

\[
\frac{\partial}{\partial n} \phi_j^{(1)} = -i \omega_e n_j \quad (j = 7, 8, ..., 12) \quad \text{on ship B} \quad (8b)
\]

where,

\[
(n_1, n_2, n_3) = \vec{n} \quad \text{on ship A}
\]

\[
(n_4, n_5, n_6) = \vec{r} \times \vec{n} \quad \text{on ship A}
\]

\[
(n_7, n_8, n_9) = \vec{n} \quad \text{on ship B}
\]

\[
(n_{10}, n_{11}, n_{12}) = \vec{r} \times \vec{n} \quad \text{on ship B}
\]

with \(\vec{n}\) is the outward unit normal vector on ship A and ship B and \(\vec{r}\) is the position vector with respect to the origin of the reference frame on ship A and ship B.

**Sea bed condition**

\[\frac{\partial}{\partial n} \phi^{(1)} = 0 \quad \text{for } z \rightarrow -\infty \quad (9)\]

**Radiation condition at infinity**

\[
\lim_{kr \to \infty} \sqrt{kr} \left( \frac{\partial \phi^{(1)}}{\partial n} - ik \phi^{(1)} \right) = 0 \quad \text{at } kr \rightarrow \infty \quad (10)
\]

The wave force including first and second order can be obtained by the direct integration of the pressure acting on the two-body hull surface in waves. The pressure in the fluid field is obtained by the use of the Bernoulli’s equation and expressed in the form as follow,

\[
P = -\rho \left[ gz + \frac{1}{2} (\nabla \phi)^2 + \left( \frac{\partial}{\partial t} - U \frac{\partial}{\partial x} \right) \phi \right] \quad (11)
\]

The total fluid forces and moments acting on the each submerged bodies can be calculated through integration of pressure over a wetted region.
\[ \vec{F} = \int_{S_A + S_B} P \hat{n} \, ds, \quad \vec{M} = \int_{S_A + S_B} P (\vec{r} \times \hat{n}) \, ds \quad (12) \]

The wave exciting force \( F_i \) can be divided into the incident wave part, \( F_i^I \), and the diffraction part, \( F_i^D \) in the form excluding speed terms.

\[ F_i = F_i^I + F_i^D = -i \rho \omega_e \int_{S_A + S_B} (\phi_i + \phi_D)n_i \, ds \quad (13a) \]

To save the computing time and to avoid the numerical error near narrow free surface gap (Newman [17], Malenica et al [18], Zalar et al [19], Chen [21], and Pauw et al [22]) between two floating bodies, Haskind [23] relation is applied in Kim and Ha [13] with various separation distance cases.

\[ F_i = -i \rho \omega_e \int_{S_A + S_B} \phi_i n_i \, ds + \rho \int_{S_A + S_B} \frac{\partial \phi_i}{\partial n} \, ds \quad (13b) \]

Expanding the pressure, force and moment as a perturbation expansion series, the zero order and higher order forces can be calculated by collecting terms of same order.

**Zero-order Force**

\[ \vec{F}^{(0)} = -\rho g \int_{S_A + S_B} z \, \hat{n}^{(0)} \, ds = (0,0,\rho g V) \quad (14) \]

where \( V \) is the displaced volume of ships. Zero order force is the hydrostatic force due to the buoyancy of an each floating structures.

**First-order Force**

\[ \vec{F}^{(1)} = -\rho \int_{S_A + S_B} \left( \frac{\partial \phi^{(1)}}{\partial t} \right) \hat{n}^{(0)} \, ds - \rho g \left( 0,0,\xi_{3,9}A_w - \xi_{5,11}A_wX_{c.f} \right) \quad (15a) \]

\[ \vec{M}^{(1)} = -\rho \int_{S_A + S_B} \left( \frac{\partial \phi^{(1)}}{\partial t} \right) (\vec{r} \times \hat{n}^{(0)}) \, ds - \rho g V \left( 0, \xi_{4,10}\overline{GM_T}, \xi_{5,11}\overline{GM_L} \right) \quad (15b) \]

where \( A_w \) is the water plane area, \( X_{c.f} \) is the longitudinal center of flotation, \( V \) is the displacement, \( \overline{GM_T} \) is the transverse meta centric height, \( \overline{GM_L} \) is the longitudinal meta centric height, respectively.

**Second-order Force**

The mean second order force and moment can be written by the time average of equation (12), (13):
\[
\vec{F}^{(2)} = \frac{1}{4} \rho g \int_{L_o} |\zeta_r^2| \vec{n}_2^{(0)} dl \\
- \frac{1}{4} \int_{\tilde{S}_o} |\nabla \tilde{\phi}^{(1)}|^2 \vec{n}^{(0)} ds \\
+ \frac{1}{2} \text{Re} [\tilde{R}_1^{*} \vec{F}^{(1)}] \\
+ \frac{1}{2} \rho \omega_e \int_{\tilde{S}_o} \text{Im} [\tilde{\alpha}^{*} \cdot \nabla \tilde{\phi}^{(1)}] \vec{n}^{(0)} ds \\
+ \frac{1}{2} \rho U \int_{\tilde{S}_o} \text{Re} [\tilde{\alpha}^{(1)*} \cdot \nabla \tilde{\phi}_{\chi}^{(1)}] \vec{n}^{(0)} ds \\
+ \frac{1}{2} \rho A_w X_c f \text{Re} [\tilde{\xi}_{4,10}^{*} \tilde{\xi}_{6,12}] \vec{k}
\]

where the superscript ~ symbol denotes the complex amplitude of the variable, * the complex conjugate and

\[
\vec{n}_2^{(0)} = \left( \vec{n}_1^{(0)}, \vec{n}_2^{(0)}, \vec{n}_3^{(0)} \right) / \sqrt{\vec{n}_1^{(0)2} + \vec{n}_2^{(0)2}}
\]

Similarly, the second order mean moment can be written:

\[
\vec{M}^{(2)} = \frac{1}{4} \rho g \int_{L_o} |\zeta_r^{2}| \left( \vec{r} \times \vec{n}_2^{(0)} \right) dl \\
- \frac{1}{4} \int_{\tilde{S}_o} |\nabla \tilde{\phi}^{(1)}|^2 \left( \vec{r} \times \vec{n}^{(0)} \right) ds \\
+ \frac{1}{2} \text{Re} [\tilde{R}_1^{*} \vec{M}^{(1)}] \\
+ \frac{1}{2} \rho \omega_e \int_{\tilde{S}_o} \text{Im} [\tilde{\alpha}^{*} \cdot \nabla \tilde{\phi}^{(1)}] \left( \vec{r} \times \vec{n}^{(0)} \right) ds \\
+ \frac{1}{2} \rho U \int_{\tilde{S}_o} \text{Re} [\tilde{\alpha}^{(1)*} \cdot \nabla \tilde{\phi}_{\chi}^{(1)}] \left( \vec{r} \times \vec{n}^{(0)} \right) ds
\]

where \(\vec{M}^{(1)}\) is the first order moment.

The motion-induced force is obtained by integration of radiation potential for the submerged body.

The motion induced force is
Wave Induced Coupled Motions and Structural Loads between Two Offshore Floating Structures in Waves

Mun Sung Kim, Kwang Hyo Jung, Sung Boo Park

\[ E_i = -i \rho \omega_e \int_{S_A+S_B} \sum_{j=1}^{12} \xi_j \phi_j n_i \, ds = \sum_{j=1}^{12} T_{ij} \xi_j \quad \text{for} \quad i = 1,2,...,12 \quad (18) \]

where,

\[ T_{ij} = \omega_e^2 A_{ij} - i \omega_e B_{ij} \quad (19) \]

The terms \( A_{ij} \) and \( B_{ij} \) are added mass and damping coefficients, respectively.

\[ A_{ij} = \frac{\rho}{\omega_e} \text{Im} \left[ \int_{S_A+S_B} \phi_j n_i \, ds \right] \quad (20) \]

\[ B_{ij} = \rho \text{Re} \left[ \int_{S_A+S_B} \phi_j n_i \, ds \right] \quad (21) \]

3. **Numerical Procedure**

The diffraction and radiation potential can be represented by the distribution with the density \( \sigma(Q) \) on the surface \( SA \) and \( SB \) in the form.

\[ \int_{S_A+S_B} \sigma(Q) G(P,Q) \, ds(Q) = 4\pi \phi(P) \quad \text{for} \quad P \text{ inside fluid} \quad (22) \]

where the Green’s function \( G(P,Q) \) is a source function at the field point \( P \) due to an unknown source density at the source point \( Q \). The unknown source density \( \sigma(Q) \) can be found by imposing the body boundary conditions equation (6), (7) and (8) and it gives

\[ -\frac{1}{2} \sigma(P) + \frac{1}{4\pi} \int_{S_A+S_B} \sigma(Q) \frac{G(P,Q)}{\partial n} \, ds(Q) = V_n \quad \text{for} \quad P \text{ body surface} \quad (23) \]

where, \( V_n \) is the normal component of the velocity on the body surface and is given in equation (6), (7) and (8). Due to the arbitrariness of the body surface and the complexity of the Green’s function, equation (23) is solved numerically. The body surface \( SA \) and \( SB \) are replaced by a number of small \( N \) surface panels of area.

\[ -\frac{1}{2} \sigma_i + \frac{1}{4\pi} \sum_{j=1, j \neq 1}^{N} \sigma_j A_{ij} \frac{G(P_i,Q_j)}{\partial n} = V_{ni} \quad (i = 1,2,...,N) \quad (24) \]

In order to calculate the Green’s function more efficiently, Telste and Noblesse [24]’s techniques have been used in this program.

4. **Equations of Motion for two floating bodies**

Under the assumption that the responses are linear and harmonic, the twelve linear coupled differential equations of motion for two floating bodies can be written in the following form
\[
\sum_{j=1}^{12} \left[ -\omega_e^2 (M_{ij} + A_{ij}) - i\omega_e B_{ij} + C_{ij} \right] \xi_j = F_i \quad \text{for } i = 1, 2, \ldots, 12
\]  

(25)

where \( M_{ij} \) is the generalized mass matrix for the ship A and ship B, \( C_{ij} \) the restoring force matrix for ship A and ship B, respectively, \( \xi_j \) the complex amplitude of the response motion in each of the six degree of freedom for each body, and \( F_i \) the complex amplitude of the wave exciting force for ship A and ship B. The generalized mass matrix for two-body has the form of

\[
M_{ij} = \begin{bmatrix}
M_A & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & M_A & 0 & -M_{Zc} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & M_A & M_{Yc} & M_{Xc} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
-12 & 0 & 0 & M_{Zc} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & M_{Xc} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & M_{Yc} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & M_{Zc} & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & M_{Xc} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & M_{Yc} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & M_{Zc} & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & M_{Xc} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & M_{Yc} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 
\end{bmatrix}
\]  

(26)

\[
X_{cA} = X_{gA} - X_{mA}, \quad Y_{cA} = Y_{gA} - Y_{mA}, \quad Z_{cA} = Z_{gA} - Z_{mA} \\
X_{cB} = X_{gB} - X_{mB}, \quad Y_{cB} = Y_{gB} - Y_{mB}, \quad Z_{cB} = Z_{gB} - Z_{mB} 
\]

where \( M_A, M_B \) are the masses, \( I_{iiA}, I_{iiB} \) the moments of inertia in i-th modes and, \( I_{ijA}, I_{ijB} \) the products of inertia, \((X_g, Y_g, Z_g)\) the center of gravity, \((X_m, Y_m, Z_m)\) the center of motion of the ship A and ship B, respectively.

The added mass and damping coefficients matrices are represented by including fully coupled effects for ship A and ship B even they have one longitudinal plane of symmetry.

\[
A_{ij} = B_{ij} \neq 0 \quad \text{for } i = 2, 4, 6, 8, 10, 12 \text{ and } j = 1, 3, 5, 7, 9, 11
\]  

(27)

\[
A_{ij} = B_{ij} \neq 0 \quad \text{for } j = 1, 3, 5, 7, 9, 11 \text{ and } i = 2, 4, 6, 8, 10, 12
\]  

(28)

Furthermore, for a two-body in the free surface the linear hydrostatic restoring force coefficients matrices are uncoupled while the added mass and damping matrices are coupled;

\[
C_{33} = \rho g A_{WA}, \quad C_{99} = \rho g A_{WB} \]  

(29a)

\[
C_{44} = \rho g V_A \bar{G} M_{TA}, \quad C_{1010} = \rho g V_B \bar{G} M_{TB} \]  

(29b)

\[
C_{55} = \rho g V_A \bar{G} M_{LA}, \quad C_{1111} = \rho g V_B \bar{G} M_{LB} \]  

(29c)

\[
C_{35} = C_{53} = -\rho g M_{WA}, \quad C_{911} = C_{119} = -\rho g M_{WB} \]  

(29d)

where \( A_{WA}, A_{WB} \) are the water plane area, \( M_{WA}, M_{WB} \) the moment of the water plane, \( V_A, V_B \) the displacement, \( \bar{G} M_{TA}, \bar{G} M_{TB} \) the transverse meta centric height, \( \bar{G} M_{LA}, \bar{G} M_{LB} \) the longitudinal meta centric height for ship A and ship B, respectively.
5. Relative motion between two floating bodies

The longitudinal, horizontal and vertical relative motion between ship A and ship B at any position can be expressed as three components by Fang and Kim [5],

\[
\frac{L_R}{\zeta_a} = \frac{1}{\zeta_a} \left[ (\xi_1 + z_A \xi_5 - y_A \xi_6) - (\xi_7 + z_B \xi_{11} - y_B \xi_{12}) \right] \\
\frac{H_R}{\zeta_a} = \frac{1}{\zeta_a} \left[ (\xi_2 + x_A \xi_6 - z_A \xi_4) - (\xi_8 + x_B \xi_{12} + z_B \xi_{10}) \right] \\
\frac{V_R}{\zeta_a} = \frac{1}{\zeta_a} \left[ (\xi_3 - x_A \xi_5 + y_A \xi_4) - (\xi_9 - x_B \xi_{11} + y_B \xi_{10}) \right]
\]

where, \((x_A, y_A, z_A)\) and \((x_B, y_B, z_B)\) are the coordinates of the position with respect to each body frame system; see Fig. 1.

6. Wave induced structural loads for two floating bodies

The wave induced structural loads, such as shear forces and bending moment between two floating bodies in waves at a particular cross section arise from the difference between the mass inertia forces and the sum of wave induced external forces acting on the portion of the hull forward of the section in question of 2-dimensional strip theory by Salvesen et al [25] and 3-dimensional theory by Chan [26]. The ship can be considered rigid body in the determination of the wave loads. The mass inertia forces are due to the accelerations of the each ship while wave induced external forces are due to the static restoring forces, the wave exciting forces as well as the motion induced hydrodynamic forces.

The wave loads between two floating bodies consist of the compression force \(F_1, F_7\), the horizontal shear force \(F_2, F_9\), and the vertical shear force \(F_3, F_9\), the torsional moment \(F_4, F_{10}\), the vertical bending moment \(F_5, F_{11}\), and the horizontal bending moment \(F_6, F_{12}\) for ship A and ship B, respectively.

\[
F_A = F_1 \hat{i} + F_2 \hat{j} + F_3 \hat{k}, \quad M_A = F_4 \hat{i} + F_5 \hat{j} + F_6 \hat{k} \quad \text{for ship A} \\
F_B = F_7 \hat{i} + F_8 \hat{j} + F_9 \hat{k}, \quad M_B = F_{10} \hat{i} + F_{11} \hat{j} + F_{12} \hat{k} \quad \text{for ship B}
\]

The definition of sign conventions for wave induced structural loads are presented in Fig. 2 and expressed in the form including speed terms as follows,
Fig. 2. Definition of sign conventions for wave induced structural loads

\[ F_1 = \int m_A (\dddot{x}_1 + z_{gA} \dddot{z}_5) \, dx_A - \rho \iiint (\phi_t - U \phi_x) \, n_1 \, ds_A \]  
\[ F_2 = \int m_A (\dddot{z}_2 + x_{A} \dddot{z}_6 - z_{gA} \dddot{z}_4) \, dx_A - \rho \iiint (\phi_t - U \phi_x) \, n_2 \, ds_A \]  
\[ F_3 = \int m_A (\dddot{x}_3 + x_{A} \dddot{z}_5) \, dx_A - \rho \iiint (\phi_t - U \phi_x) \, n_3 \, ds_A \]  
\[ + 2\rho g \int y_A (\dddot{z}_3 - x_{A} \dddot{z}_5) \, dx_A \]  
\[ F_4 = \iiint [u_{44} \dddot{z}_4 - m_A z_{gA} (\dddot{z}_2 + x_{A} \dddot{z}_6)] \, dx_A - \rho \iiint (\phi_t - U \phi_x) n_4 ds_A \]  
\[ + \rho \dddot{z}_4 \left[ 2\rho \int y_A^2 \, dx_A - \int m_A z_{gA} \, dx_A + \rho \int A_A z_{gA} \, dx_A \right] \]  
\[ F_5 = -(x_A - x_{pA}) F_3 \]  
\[ F_6 = (x_A - x_{pA}) F_2 \]  
\[ F_7 = \int m_B (\dddot{z}_7 + z_{gB} \dddot{z}_{11}) \, dx_B - \rho \iiint (\phi_t - U \phi_x) n_7 \, ds_B \]  
\[ F_8 = \int m_B (\dddot{x}_8 + x_{B} \dddot{z}_{12} - z_{gB} \dddot{z}_{10}) \, dx_B - \rho \iiint (\phi_t - U \phi_x) n_8 \, ds_B \]  
\[ F_9 = \int m_B (\dddot{z}_9 + x_{B} \dddot{z}_{11}) \, dx_B - \rho \iiint (\phi_t - U \phi_x) n_9 \, ds_B \]
\[ + 2 \rho g \int y_B (\xi_9 - x_B \xi_{11}) dx_B \]  
\[ F_{10} = \int [t_{1010} \dddot{\xi}_{10} - m_B z_{gB} (\dddot{\xi}_B + x_B \dddot{\xi}_{12})] dx_B - \rho \iint (\phi_t - U \phi_x) n_{10} ds_B \]
\[ + g \xi_{10} \left[ 2 \rho \int y_B^3 dx_B - \int m_B z_{gB} dx_B + \rho \int A_B z_{BB} dx_B \right] \]
\[ F_{11} = -(x_B - x_{pB}) F_9 \]
\[ F_{12} = (x_B - x_{pB}) F_8 \]

where \( m_A, m_B \) are the sectional mass per unit length of the ship, \( z_{gA}, z_{gB} \) the vertical position of center of gravity of the sectional mass, \( z_{pA}, z_{pB} \) the vertical position of center of submerged cross section, \( A_A, A_B \) the area of the submerged cross section, \( i_{44}, i_{1010} \) the sectional mass moment of inertia about the x-axis. The \( \int \) and \( \iint \) are the surface integration over the mean wetted hull surface forward of specific cross section and the line integration along the hull forward of specific section, respectively.

7. **Numerical Results and Discussion**

The numerical calculations of hydrodynamic pressure, wave exciting force, motion response, relative motion and wave loads between two floating structures have been carried out for Kodan [2]'s rectangular barge and conventional ship model with zero speed. The principal particulars of a barge and a ship model are given in Table 1. The distance between barge and ship from each body's center of gravity is 1.2m, i.e., \( P=1.2 \text{m} \) (see Fig.3). The longitudinal centers of barge and ship are positioned same.

In order to validate the developed motion prediction program, the comparisons are performed for Kodan [2]'s experimental and 2-D strip theory numerical results, Fang and Kim [5]'s 2-D strip theory numerical results and Fang and Chen [6, 27]'s 3-D potential theory numerical results, respectively.

The numerical calculated results of hydrodynamic pressure distribution, wave exciting force, motion response, relative motion and wave loads in waves are presented in Fig. 5 through Fig. 22.

**Table 1. Principal particulars of two models (Kodan, [2])**

<table>
<thead>
<tr>
<th>Items</th>
<th>Unit</th>
<th>Barge(Left)</th>
<th>Ship(Right)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>m</td>
<td>3.125</td>
<td>2.088</td>
</tr>
<tr>
<td>Breadth</td>
<td>m</td>
<td>0.600</td>
<td>0.369</td>
</tr>
<tr>
<td>Draft</td>
<td>m</td>
<td>0.113</td>
<td>0.131</td>
</tr>
<tr>
<td>Displacement</td>
<td>m³</td>
<td>0.203</td>
<td>0.081</td>
</tr>
<tr>
<td>Water plane area</td>
<td>m²</td>
<td>1.875</td>
<td>0.685</td>
</tr>
<tr>
<td>( GM_T )</td>
<td>m</td>
<td>0.223</td>
<td>0.074</td>
</tr>
<tr>
<td>( GM_L )</td>
<td>m</td>
<td>7.184</td>
<td>2.523</td>
</tr>
<tr>
<td>KG</td>
<td>m</td>
<td>0.106</td>
<td>0.080</td>
</tr>
<tr>
<td>( K_{xx} )</td>
<td>m</td>
<td>0.118</td>
<td>0.097</td>
</tr>
<tr>
<td>( K_{yy} )</td>
<td>m</td>
<td>0.751</td>
<td>0.506</td>
</tr>
</tbody>
</table>
The calculations of second order wave drift force between two floating structures have been carried out for same size barge model. The principal particulars of barge are presented in Table 2. The distance between two barges, from the wall of each barge, are 10.0m for side-by-side offloading arrangement (see Fig. 4).

The comparisons are performed to validate the program for Pinkster [20]'s experimental results. The numerical results of motion response and drift forces are presented in Fig. 23 through Fig. 25.

**Table 2. Principal particulars of two barge models (Pinkster, [20])**

<table>
<thead>
<tr>
<th>Items</th>
<th>Unit</th>
<th>A Barge (Left)</th>
<th>B Barge (Right)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length m</td>
<td></td>
<td>150.0</td>
<td>150.0</td>
</tr>
<tr>
<td>Breadth m</td>
<td></td>
<td>50.0</td>
<td>50.0</td>
</tr>
<tr>
<td>Draft m</td>
<td></td>
<td>10.0</td>
<td>10.0</td>
</tr>
<tr>
<td>Displacement m(^3)</td>
<td></td>
<td>73,750</td>
<td>73,750</td>
</tr>
<tr>
<td>Water plane area m(^2)</td>
<td></td>
<td>7,500</td>
<td>7,500</td>
</tr>
<tr>
<td>GM(_T) m</td>
<td></td>
<td>16.23</td>
<td>16.23</td>
</tr>
<tr>
<td>GM(_L) m</td>
<td></td>
<td>186.0</td>
<td>186.0</td>
</tr>
<tr>
<td>KG m</td>
<td></td>
<td>10.0</td>
<td>10.0</td>
</tr>
</tbody>
</table>

The distance between the centers of two models, P=10.0m
7.1 Hydrodynamic pressure distribution between barge and ship

The hydrodynamic pressure distributions on the ship hull surface with and without barge are shown in Fig. 5 through Fig. 7. The hydrodynamic pressure is non-dimensionalized by the density of sea water, gravitational constant, wave amplitude.

Fig. 5 shows the hydrodynamic pressure distribution for ship with and without barge in following seas ($\beta = 0^\circ$) which wavelength is 2.038m and has same length as ship length. In case of ship floating alone without barge, the hydrodynamic pressure distribution has symmetric behavior along the x-direction, so there is no any horizontal mode force and motion. However, in case of ship with barge, the hydrodynamic pressure has asymmetric behavior along the x-direction due to hydrodynamic interaction, so it is expected that the horizontal mode force and motion in following and head seas. Fig. 6 and 7 show the hydrodynamic pressure distributions on the ship at $45^\circ$ and $-45^\circ$ heading waves. If the ship is floating alone, there is no hydrodynamic pressure difference irrelevant to wave heading angle. However, in case of ship with barge, the hydrodynamic pressure has under the influence of the barge. The hydrodynamic pressure of the ship with barge is smaller on the leeside than on the weather side due to the hydrodynamic interaction by the sheltering effect.
Fig. 5. Hydrodynamic pressure for ship with and without barge at $\beta=0^\circ (\lambda=2.038\text{m})$

Fig. 6. Hydrodynamic pressure for ship with and without barge at $\beta=45^\circ (\lambda=2.038\text{m})$

Fig. 7. Hydrodynamic pressure for ship with and without barge at $\beta=-45^\circ (\lambda=2.038\text{m})$
7.2 Wave exciting force and moment between barge and ship

The wave exciting forces and moments are shown in Fig. 8 through Fig. 10. The dashed line shows the results of the ship without barge. Fig. 8 and 9 show the sway and heave exciting force for ship at 45° and -45° heading waves. The roll exciting moment for a ship at 45° and -45° heading waves are shown in Fig. 10.

Fig. 11 show the sway and heave exciting force for ship at 0° heading waves. Due to the hydrodynamic interaction effect between barge and ship, the ship exciting forces and moments with barge are quite different from the exciting forces and moments of a single ship without barge. The one of the interesting phenomena on two-body wave exciting forces that the interaction effect was found in following sea (β= 0°), where no sway exciting forces occur if the ship was floating alone. Generally, theoretical results give a good correlation with the Kodan [2]'s experimental results except the roll exciting moment at β=-45°.
Fig. 10. Roll exciting moment for ship with and without barge at $\beta=45^\circ$, $\beta=-45^\circ$

Fig. 11. Sway and Heave exciting force for ship with and without barge at $\beta=0^\circ$
7.3 Motion behaviors between barge and ship

Fig. 12 show the sway amplitudes for ship at 45° and -45° heading waves. The heave amplitudes for a ship at 45° and -45° heading waves are shown in Fig. 13. Due to the hydrodynamic interaction effect between barge and ship, the coupled resonance occur in the motion responses and the ship motion responses with barge are quite different from the responses of a single ship without barge. All the motion responses of the ship with barge are smaller on the leeside than on the weather side because of the sheltering effect.

Fig. 14 show the sway and the heave amplitudes for barge at 45° and -45° heading waves, respectively. Unlike the ship's case, the motion responses of barge with ship are the same as the motion responses of single barge without ship except several frequencies, which causes the hydrodynamic interaction effect between barge and ship. In general, the present 3-D results are much closer to the experimental result than other numerical results specially heave and sway motion response.
7.4 Relative motions between barge and ship

Fig. 15 shows the horizontal relative motion amplitudes at 45° and -45° heading waves. The calculation positions for the relative motion are each body's center of gravity, i.e., \((x_A, y_A, z_A)=(0, 0, 0)\), \((x_B, y_B, z_B)=(0, 0, 0)\).

Fig. 16 shows the vertical relative motion amplitude at 45° and -45° heading waves. For comparison of the present 3-D relative motion results, Fang and Kim [5]'s 2-D results and Fang and Chen [6]'s 3-D results are used. Generally, the large relative motions between two bodies occur around low frequency region and coupled resonance frequency region. As we can see, the present 3-D results have the same behavior as Fang and Chen [6]'s 3-D results except the high frequency, which caused the difference of heave motion responses in coupled resonance regions. Like the hydrodynamic pressure, wave exciting force and motion responses, all the relative motion responses of the ship with barge are smaller on the leeside than on the weather side due to the sheltering effect.

Fig. 15. Horizontal relative motion between barge and ship at \(\beta=45^\circ\), \(\beta=-45^\circ\)

Fig. 14. Sway and Heave amplitude for barge with and without ship at \(\beta=45^\circ\), \(\beta=-45^\circ\)
7.5 Wave induced structural loads between barge and ship

Once we know the wave exciting force and the motion responses between two bodies, we can predict the wave induced structural loads at mid-ship section of the ship by using equation (35) through (46). The wave loads for ship at mid-ship section are shown in Fig. 17 through Fig. 22. Horizontal bending moments for ship with and without barge at 45° and -45° heading waves are shown in Fig. 17, Fig. 18 and Fig. 19 show the vertical bending moments and torsional moments at 45° and -45° heading waves, respectively. Unfortunately, the experimental data was not available to compare for the calculated wave loads. However, it is a reasonable result that because of the equation of wave loads can be solved numerically by known wave exciting force and motion responses for correspond wave frequency and heading angle.

Like the hydrodynamic pressure distributions, wave exciting forces and motion responses, the hydrodynamic interaction effects between two bodies are also more affect the wave loads of ship with barge than that of ship floating alone without barge, specially resonance frequency region. Due to the sheltering effect, the wave loads on the lee side are generally smaller than the case on the weather side. As shown in Fig. 20 through Fig. 22, the one of the interesting phenomena on two-body wave loads that the interaction effect was found in head and following sea (β=180°, β=0°), where no horizontal bending moment and torsional moment occur if the ship was floating alone.

Fig. 16. Vertical relative motion between barge and ship at β=45°, β=-45°

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**Fig. 17.** Horizontal Bending Moment for ship with and without barge at $\beta=45^\circ, \beta=-45^\circ$

**Fig. 18.** Vertical Bending Moment for ship with and without barge at $\beta=45^\circ, \beta=-45^\circ$

**Fig. 19.** Torsional Moment for ship with and without barge at $\beta=45^\circ, \beta=-45^\circ$
Fig. 20. Horizontal Bending Moment for ship with and without barge at $\beta=180^\circ$, $\beta=0^\circ$

Fig. 21. Vertical Bending Moment for ship with and without barge at $\beta=180^\circ$, $\beta=0^\circ$

Fig. 22. Torsional Moment for ship with and without barge at $\beta=180^\circ$, $\beta=0^\circ$
7.6 Wave induced second order force and moment between two barges

The comparisons are performed to validate the program for Pinkster [20]'s experimental results. The numerical results of motion response and drift forces are presented in Fig. 23 through Fig. 25. The solid line and circle symbol show the results of the barge alone without another barge.

Fig. 23 shows the surge and sway amplitude at Head and Bow quartering seas, respectively. Because the two barges have same size and dimension, the surge amplitude in head seas have same behavior and hydrodynamic interaction is small.

However, the hydrodynamic interaction effect was shown in sway amplitude at Bow quartering seas despite two barges have same condition. Due to the hydrodynamic interaction between two barges, the motion responses with barge are quite different from the responses of a single barge alone. The motion response of the barge is smaller on the leeside than on the weather side because barge B obstructs the wave exciting force of barge A.

Fig. 24 and 25 show the surge, sway and yaw drift force and moment in Head and Bow quartering seas, respectively. One of the interesting phenomena on two-body motion problem that interaction effect was found in head sea, where no sway and yaw drift forces and moment occur if the barge is floating alone. Because the two barges have same size and dimension, the surge drift force in head seas is same behavior and hydrodynamic interaction is small like the motion response. From the results, we find that two barges drift away against each other, that are barge A moves to the positive y-axis direction and barge B moves to the negative y-axis, and the drift force for two barges is same. This implies that the results seem to be physically reasonable. If the LNG-FPSO contains turret system to control the bow-heading angle against the incident waves, the dominant waves are head and bow quartering seas since LNG-FPSO weather vane along wave heading. However, the high stiffened mooring lines between two floating structures to endure the wave drift force and moment in head and bow quartering seas will solve this problem.

![Fig. 23. Surge motion amplitude at $\beta=180^\circ$ and Sway motion amplitude at $\beta=135^\circ$](image_url)

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8. Conclusions

In this present paper, the numerical predictions are described on the hydrodynamic pressure distributions, wave exciting forces, coupled motion responses, wave induced structural loads and second order drift forces between two offshore floating structures including the hydrodynamic interaction effect in various heading waves.

In order to validate the developed motion program between two floating structures, we performed the comparison study for the well-known rectangular barge and conventional ship model. Generally, the numerical calculation results give a good correlation with the experimental results and also with other numerical results.

According to the present numerical study, the motion response, wave loads including second order drift forces between two floating structures due to the hydrodynamic interaction effect can be quite significant phenomena during offloading operation. This hydrodynamic interaction effect between two floating structures comes from the incident waves scattering and radiate waves due to the presence of neighborhood floating structures.

Therefore, the motion analysis including the hydrodynamic interaction effect between two floating structures during offloading operations is needed for the more safe offloading operability and also should be applied to the offloading system design to avoid the undesirable large motions at the connected loading arm.
Also, the optimum separation distance of these multiple floating structures during offloading should be carefully designed (Kim and Ha, [13]) to avoid the unnecessary large motions between two floating bodies.

As a result, the present developed numerical calculation program can be used effectively to predict the wave exciting forces, motion responses, relative motions, first and second order wave loads for multiple floating structures in various heading waves.

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UDC 629.5.016: 629.5.017.3: 629.561.1
Professional paper

Summary

Selection of a tugboat to be used in a port according to the operations to be carried out is a difficult problem that requires many criteria to be evaluated at the same time. This selection requires high experience as well as technical knowledge about tugboats and operations. In this study, a numerical analysis was carried out within the framework of design, operational and financial criteria to develop a method to select a tugboat. The propulsion/manoeuvring systems used in the tugboats were investigated and different criteria related to the tugboats with different propulsion systems were evaluated by subject matter experts through a survey including technical and financial data. The responses were interpreted with the fuzzy analytic hierarchy process to select a suitable tugboat alternative based on the type of propulsion system.

Key words: Tugboat; fuzzy analytic hierarchy process; propulsion system; manoeuvring

1. Introduction

The nature of transportation is directly influenced by international trends and global developments. Maritime economy has always an important place in world trade and transportation. Main arteries of maritime import-export economy are ports. In order to provide faster and safer traffic flow in the ports, besides good organization and coordination, it is necessary to have advanced suitable equipment and vessels that serve the respective port. Among them, tugboats enable faster and safer performance of manoeuvring in ports and consequently speed up the flow of goods through the port concerned.

A tugboat is a type of vessel that manoeuvres other vessels by pushing or pulling them either by direct contact or by means of a tow line [1]. Tugs typically move vessels that either are restricted in their ability to manoeuvre on their own such as ships in a crowded harbour or a narrow canal or those that cannot move by themselves such as barges, disabled ships, log rafts or oil platforms. Tugboats are powerful in terms of their size and strongly built, and some are ocean-going. A typical profile drawing of a tugboat is given in Fig. 1.
Tugboats are designed to perform one or more very specific functions. Of course, many tugboats also tend to get used to perform more than one of these duties and thus become more “multi-purpose”. The primarily required specifications for tugboats are high manoeuvrability and power. However, it is important to understand that tugboats with different design features have different handling characteristics [2]. These could be, but not limited to, a combination of hull profile, engine and/or rudder type and thruster’s configuration and towing winch design, power and location. Selection of the tugboats according to the port and operations to be used is of great importance in terms of efficiency and operating cost. Typically, tugboats are categorized according to the type of work they do, and then by the configuration or type of propulsion system used [3]. Main propulsion systems of the tugboats vary according to the operational requirements and capabilities of the tugboat. In general, there are three basic types of harbour tugboats according to the propulsion systems: Conventional, azimuth stern drive and tractor. The main differences between these 3 types are the equipment used in the propulsion system and the locations of these equipment. Details will be given in Table 5, third section [4, 5].

Numerous researches have been carried out related to the selection of propulsion and manoeuvring systems. The sophisticated techniques about the selection of waterjet propulsion systems for hydrofoil craft were examined by Hatte and Davis [6]. Barr, R. A. and Etter, R. J. [7] suggested a problem of matching ship performance and geometric requirements to propulsion system characteristics to select the best propulsion system for a specific application to be discussed. Stevens et al. [8] brought into question the superconducting electric propulsion machinery systems for a ship and examined 3000 hp feasibility models of full scale systems. Unal et al. [9] discussed the Taguchi method for a propulsion system design optimization and proposed a systematic and efficient approach for transportation vehicle. Olcer and Odabasi [10] suggested a generalised fuzzy multiple attributive group decision making methodology about propulsion/manoeuvring system selection to warship requirements definition. Chang et al. [11] elaborated the safety requirements of dual-fuel gas turbine electric (DFGE) propulsion and dual-fuel diesel mechanical (DFDM) propulsion system. Castles et al. [12] assessed propulsion system inefficiency for destroyers and other marine vessels and proposed a hybrid drive operation system. The development of electric propulsion motors for naval and commercial ships was investigated by McCoy and Amy [13]. Mizythras et al. [14] advanced a new tool for simulation of a ship propulsion system performance during manoeuvring in shallow waters by coupling the propulsion system and seakeeping models. Martelli, M., and Figari, M. [15] examined the methodology and the
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Abit Balın

Simulation models required to design the propulsion control logics for an innovative combined diesel-electric and gas (CODLAG) propulsion plant. Altosole, M., and Martelli, M. [16] did propose a simulation based design methodology adopted to improve and check new control schemes for ship propulsion control strategies for emergency manoeuvres. Geertsma, et al. [17] put forward a propulsion model with a Mean Value First Principle (MVFP) diesel engine model that is likely to be parameterised with publicly available manufacturer data and further calibrated with obligatory Factory Acceptance Tests (FAT) and Sea Acceptance Tests (SAT) measurements. Geertsma et al. [18] presented a simulation model of a hybrid propulsion system to investigate two parallel control strategies for diesel mechanical and electrical propulsion on multifunction ships. Geertsma et al. [19] analysed the benefits and drawbacks, and trends in application of propulsion and power supply technologies, and they review the applicability and benefits of promising advanced control strategies. Mu et al. [20] assessed the podded propulsion unmanned surface vehicle model identification based on field experiments and verified by simulation data. Pugi et al. [21] proposed redundant and reconfigurable propulsion systems to improve motion capability of underwater vehicles and layout exhibit superior manoeuvring performances that are expected to be useful for the inspection of offshore plants and more generally for harsh operational conditions.

In this paper, a new multi-criteria decision making (MCDM) problem has been proposed in a fuzzy environment about ranking and selection of alternatives. A new methodology was presented with an application to the propulsion system selection problem. Thus, the research is devoted to find a useful and rational decision making model that enables to handle the already-mentioned problems. The main objective of this research is to make contribution to the development of an MCDM method with multiple decision makers, which are capable of working in a fuzzy environment. In spite of the fact that the method has already identified in some scientific articles for different areas, few studies are available for the importance ratings on propulsion system selection in the maritime industry.

This paper has been organized as follows: Section 2 is intended to outline the general concept of analytic hierarchy process problem, Section 3 propulsion system selection is used as a real case study to show the applicability of the proposed approach, and results of the evaluations and concluding remarks are presented in the last sections.

2. Concept of analytic hierarchy process

Proposed by Saaty [22, 23], Analytic Hierarchy Process (AHP) is traditionally a powerful decision-making methodology so as to determine the priorities among different criteria, enabling to compare alternatives for each criterion and to determine an overall ranking of the alternatives. It has also been used in various fields in the shipbuilding and maritime industry. Zamarin et al. [24] developed a novel three-stage methodology for selection of optimal mast and standing rigging using AHP in the first two stages and FEM analysis in the final stage. Balin et al. [25] implemented fuzzy AHP and VIKOR techniques to the expert failure detection of marine diesel engine and auxiliary systems. Kafali et al [26] used AHP to evaluate different pipe cutting methods used in shipyards in accordance with the criteria which consist of three main factors and fourteen sub-factors, and determined the most appropriate method. Stanić et al. [27] investigated modern production concepts to improve productivity and decrease operational costs in shipbuilding process. AHP was used after a three-phase methodology to select the optimal solution. Their methodology could also be used to detect the topics that need improvements and some other crucial points to overall reduction in costs.

The conventional AHP is hardly sufficient to handle the imprecise or vague nature of linguistic assessment. Possible to be represented by the triangular fuzzy numbers (TFNs) [28]
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in fuzzy AHP, common sense linguistic statements have been utilized in the pair-wise comparison. Then, the step of combining the pair-wise comparison and the synthesis of the priorities to decide on the overall priorities of the decision alternatives will be done [29].

The cause of TFNs being used to capture the vagueness of the linguistic assessments is that TFN is open to use both in intuitive and easy way. The TFNs in the pair-wise comparison are defined with the help of three real numbers expressed as a triple \((l, m, u)\). These values are defined on the basis of the smallest possible value, the most promising value, and the largest possible value that describes a fuzzy event. The one that looks to better match the preferences scale of the Fuzzy AHP is summarized in Table 1.

**Table 1** Triangular fuzzy conversion scale

<table>
<thead>
<tr>
<th>Linguistic Scale</th>
<th>TFNs/ Reciprocal TFNs</th>
</tr>
</thead>
<tbody>
<tr>
<td>AS-Absolutely Strong</td>
<td>(3.50, 4.00, 4.50)</td>
</tr>
<tr>
<td>VS-Very strong</td>
<td>(2.50, 3.00, 3.50)</td>
</tr>
<tr>
<td>FS-Fairly strong</td>
<td>(1.50, 2.00, 2.50)</td>
</tr>
<tr>
<td>SS-Slightly strong</td>
<td>(0.50, 1.00, 1.50)</td>
</tr>
<tr>
<td>E-Equal</td>
<td>(1.00, 1.00, 1.00)</td>
</tr>
<tr>
<td>SW-Slightly weak</td>
<td>(0.67, 1.00, 2.00)</td>
</tr>
<tr>
<td>FW-Fairly weak</td>
<td>(0.40, 0.50, 0.67)</td>
</tr>
<tr>
<td>VW-Very weak</td>
<td>(0.29, 0.33, 0.40)</td>
</tr>
<tr>
<td>AW-Absolutely weak</td>
<td>(0.22, 0.25, 0.29)</td>
</tr>
</tbody>
</table>

The operations on TFNs can be addition, multiplication, and inverse. Suppose \(M_1\) and \(M_2\) are TFNs where \(M_1 = (l_1, m_1, u_1)\) and \(M_2 = (l_2, m_2, u_2)\); then,

**Addition:** \[ M_1 \oplus M_2 = (l_1 + l_2, m_1 + m_2, u_1 + u_2) \] (1)

**Multiplication:** \[ M_1 \otimes M_2 = (l_1 \cdot l_2, m_1 \cdot m_2, u_1 \cdot u_2) \] (2)

**Inverse:** \[ M_1^{-1} = (l_1, m_1, u_1)^{-1} \cdot (1/u_1, 1/m_1, 1/l_1) \] (3)

**Step 1: Construct the fuzzy pair-wise comparison matrix**

To construct the fuzzy judgment matrix \(\hat{A} = \tilde{a}_{ij}\) of \(n\) criteria or alternatives via pair-wise comparison by asking which is the more important of each two criteria, the TFNs are used as follows by equation (4).

\[
\hat{A} = \begin{bmatrix}
1 & \tilde{a}_{12} & \ldots & \tilde{a}_{1n} \\
\tilde{a}_{21} & 1 & \ldots & \tilde{a}_{2n} \\
\ldots & \ldots & \ldots & \ldots \\
\tilde{a}_{ni} & \tilde{a}_{n1} & \ldots & 1
\end{bmatrix}
\] (4)

where \(\tilde{a}_{ij}\) is a fuzzy triangular number, \(\tilde{a}_{ij} = l_{ij}, m_{ij}, u_{ij}\), and \(\tilde{a}_{ji} = 1/\tilde{a}_{ij}\). For each TFN, \(\tilde{a}_i\) or \(M = (l, m, u)\), its membership function \(\mu_{\tilde{a}_i}(x)\) or \(\mu_M(x)\) is a continuous mapping from real number \(-\infty \leq x \leq \infty\) to the closed interval \([0, 1]\) and can be defined by equation (5).
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\[
\mu_x(x) = \begin{cases} 
(x-l)/(m-l), & l \leq x \leq m \\
(u-x)/(u-m), & m \leq x \leq u \\
0, & \text{otherwise} 
\end{cases} 
\]  

(5)

Step 2: Aggregate the group decisions

Subsequent to the collection of the fuzzy judgment matrices from all decision makers, these matrices can be assembled by dint of the fuzzy geometric mean method (6) of Buckley [30, 31]. The total TFN of n decision makers’ judgment in a certain case \( \bar{u}_{ij} = (l_{ij}, m_{ij}, u_{ij}) \) is:

\[
\bar{u}_{ij} = (\prod_{k=1}^{n} \bar{a}_{yk})^{1/n} 
\]  

(6)

where \( \bar{a}_{yk} \) is the relative importance in form of TFN of the \( k_{th} \) decision maker’s view, and n is the total number of decision makers.

Step 3: Compute the value of fuzzy synthetic extent:

Based on the aggregated pair-wise comparison matrix, \( \bar{U} = \bar{u}_{ij} \), the value of fuzzy synthetic extent \( S_i \) with respect to the \( i_{th} \) criterion can be computed as (7) by making use of the algebraic operations on TFNs as described in (1)–(3).

\[
W_i = \sum_{j=1}^{m} \bar{u}_{ij} \otimes \left[ \sum_{j=1}^{n} \sum_{j=1}^{m} \bar{u}_{ij} \right]^{-1} 
\]  

(7)

where

\[
\sum_{j=1}^{m} \bar{u}_{ij} = \left( \sum_{j=1}^{m} l_{ij}, \sum_{j=1}^{m} m_{ij}, \sum_{j=1}^{m} u_{ij} \right) \quad \text{and} \quad \sum_{i=1}^{n} \sum_{j=1}^{m} \bar{u}_{ij} = \left( \sum_{i=1}^{n} \sum_{j=1}^{m} l_{ij}, \sum_{i=1}^{n} \sum_{j=1}^{m} m_{ij}, \sum_{i=1}^{n} \sum_{j=1}^{m} u_{ij} \right)
\]

Step 4: Calculate best non-fuzzy performance (BNP) value

Utilize Center of Area (COA) method to find out the best non-fuzzy performance (BNP) value (crisp weights) of each criterion by the Equation (8).

\[
\text{BNP} \bar{w}_i = \frac{(lw_i + mw_i + lw_i)}{3 + lw_i} 
\]  

(8)

In accordance with the value of the derived BNP for each of the alternatives, the ranking of the each alternative can then proceed.

Step 5: Consistency test of the comparison matrix

To ensure a certain quality level of a decision, we are required to analyze the consistency of an evaluation. With the aim of testing the value of consistency of the comparison matrix depended on n, the consistency rate (CR) needs to be computed. The CR is described in (9) as a ration between the consistency of a consistency index (CI) and the consistency of a random consistency index (RI). Its value should not surpassed 0.1 for a matrix larger than 4x4. For pair-wise comparison matrix being compatible, upper-bound of CR should be as shown in Table 2 [22, 23].

\[
\text{CR} = \frac{\text{CI}}{\text{RI}} 
\]  

(9)
Table 2: Upper bound for pair-wise comparison matrix to be compatible

<table>
<thead>
<tr>
<th>n</th>
<th>3x3</th>
<th>4x4</th>
<th>n&gt;4</th>
</tr>
</thead>
<tbody>
<tr>
<td>CR</td>
<td>0.58</td>
<td>0.90</td>
<td>1.12</td>
</tr>
</tbody>
</table>

The CI is used to measure the inconsistency pair-wise comparison as shown in (10) where the eigenvalue $\lambda_{\text{max}}$ can be computed by averaging all eigenvalues of the pair-wise comparison matrix (11). Table 3 shows values of RI in different values of n.

$$\text{CI} = (\lambda_{\text{max}} - n) / (n - 1)$$

$$\lambda_{\text{max}} = \sum_{j=1}^{n} \frac{a_{ij} W_j}{W_i} = n \quad i, j = 1, 2, ..., n$$

Table 3: Values of random consistency index (RI) per different number of criteria [32]

<table>
<thead>
<tr>
<th>n</th>
<th>RI</th>
<th>n</th>
<th>RI</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>0.58</td>
<td>10</td>
<td>1.49</td>
</tr>
<tr>
<td>4</td>
<td>0.90</td>
<td>11</td>
<td>1.51</td>
</tr>
<tr>
<td>5</td>
<td>1.12</td>
<td>12</td>
<td>1.48</td>
</tr>
<tr>
<td>6</td>
<td>1.24</td>
<td>13</td>
<td>1.56</td>
</tr>
<tr>
<td>7</td>
<td>1.32</td>
<td>14</td>
<td>1.57</td>
</tr>
<tr>
<td>8</td>
<td>1.41</td>
<td>15</td>
<td>1.59</td>
</tr>
<tr>
<td>9</td>
<td>1.45</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

3. Case study: Selection of suitable tugboat according to propulsion system for ports of Turkey

Selection of a suitable tugboat for a port is a very complex task that requires many criteria to be evaluated at the same time. In this study, it is aimed to apply fuzzy analytic hierarchy process to this selection problem. Fig. 2 displays hierarchical structure designed in compliance with Buckly’s fuzzy AHP method that includes criteria and tugboat alternatives. Short definitions of the criteria that have effect on the selection of the tugboat are given in Table 4.

Fig. 2 Hierarchical structure for tugboat selection according to propulsion system type
Applying fuzzy analytic hierarchy process to selection of suitable tugboat according to propulsion/manoeuvring system type for ports in Turkey

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Table 4 Definitions of selected criteria and tugboat alternatives

<table>
<thead>
<tr>
<th>No</th>
<th>Criteria</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>Bollard pull</td>
<td>It is a conventional measure of the pulling (or towing) power of a tugboat. It is defined as the force exerted by a tugboat under full power and used for measuring the strength of tugboats.</td>
</tr>
<tr>
<td>C2</td>
<td>Speed</td>
<td>Maximum and/or service speed of a tugboat</td>
</tr>
<tr>
<td>C3</td>
<td>Tank capacity</td>
<td>Capacity of the fuel tank and the other tanks</td>
</tr>
<tr>
<td>C4</td>
<td>Seakeeping</td>
<td>The seakeeping performance accounts for the performance of a tugboat in wind, waves and current. The performance can be expressed in terms of comfort, crew workability, damage to ship and cargo due to ship motions.</td>
</tr>
<tr>
<td>C5</td>
<td>Deck arrangement</td>
<td>The size of working area on deck and the arrangement of the equipment e.g. winches, windlass, and hook</td>
</tr>
<tr>
<td>C6</td>
<td>Hull form</td>
<td>The underwater design of the tugboat and the characteristic of the hull lines</td>
</tr>
<tr>
<td>C7</td>
<td>Working environment</td>
<td>The environmental conditions in which the tugboat will work</td>
</tr>
<tr>
<td>C8</td>
<td>Safety</td>
<td>Vessel stability in towing operation and critical equipment installation on deck and engine room</td>
</tr>
<tr>
<td>C9</td>
<td>Maintenance</td>
<td>Short-time, easy and cheaper maintenance</td>
</tr>
<tr>
<td>C10</td>
<td>Operation</td>
<td>Low operation costs, e.g. low fuel consumption, low crew cost</td>
</tr>
<tr>
<td>C11</td>
<td>Functionality</td>
<td>Easy line handling and best manoeuvring in limited areas at the port</td>
</tr>
<tr>
<td>C12</td>
<td>Price</td>
<td>Required capital and operational expenses</td>
</tr>
<tr>
<td>C13</td>
<td>Delivery time</td>
<td>Short-term construction after order confirmation</td>
</tr>
<tr>
<td>C14</td>
<td>Maturity possibility</td>
<td>Support to the customer for financing issues</td>
</tr>
</tbody>
</table>

As mentioned in introduction section, tugboats are generally categorized according to the configuration or type of propulsion system used. In general, there are three basic types of harbour tugboats: Conventional, azimuth stern drive (ASD) and tractor. Tractor type tugboats are also divide into 2 main types. The main difference between these types is the equipment used in the propulsion system and the locations of these equipment. Short definitions and informative profile silhouettes of alternative tugboats used in study are given in Table 5.
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Applying fuzzy analytic hierarchy process to selection of suitable tugboat according to propulsion/manoeuvring system type for ports in Turkey

Table 5 Alternative tugboat types according to the propulsion system

<table>
<thead>
<tr>
<th>No</th>
<th>Alternative</th>
<th>Profile silhouette</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>ASD - Azimuth stern drive tug</td>
<td><img src="image1" alt="Image" /></td>
<td>A tug fitted with two azimuth thrusters in nozzles at the stern.</td>
</tr>
<tr>
<td>A2</td>
<td>Conventional tug</td>
<td><img src="image2" alt="Image" /></td>
<td>A tug fitted with fixed propellers, single or twin screw (left or right handed) and single rudders with fixed nozzles.</td>
</tr>
<tr>
<td>A3</td>
<td>ASD Tractor tug</td>
<td><img src="image3" alt="Image" /></td>
<td>A tug with azimuth thrusters located generally forward off the midship and a rudder-shaped fin at aft side.</td>
</tr>
<tr>
<td>A4</td>
<td>VSP (Voith-Schneider Propeller) Tractor tug</td>
<td><img src="image4" alt="Image" /></td>
<td>A tug with Voith-schneider propellers generally located forward off the midship and a rudder-shaped fin at aft side.</td>
</tr>
</tbody>
</table>

According to the hierarchical structure created, pairwise comparisons to the tugboats with different propulsion systems by subject-matter-experts have been employed for geometric average proposed by Buckley Fuzzy AHP. Triangular Fuzzy Numbers (TFN) in pairwise comparison have been utilized instead of real numbers so as to identify the triple \((l, m, n)\) where \(l \leq m \leq u\) is.

4. Results and discussion

All views of subject-matter-experts have been evaluated as stated above. A common view has been obtained by taking the geometric average of all pairwise comparisons. This common view was evaluated by Buckley Fuzzy AHP and the results are shown in Table 6.

The results obtained according to the criteria by examining the solutions in detail are shown in Table 7. As shown, according to the evaluations of subject-matter-experts, it is obvious that criteria C8 \((0.130)\) and C1 \((0.118)\) affects the selection of tugboat type mostly. The C14, C5, C6, and C3 criteria are in the last place with \((0.050)\), \((0.052)\), \((0.053)\), and \((0.054)\), respectively. In this case, it can be seen that C8 and C1 criteria are the most important factors affecting the selection of the tugboat regarding the propulsion system. On the other hand, it has been observed that the C14, C5, C6 and C3 criteria less influence the selection of the tugboat than the other criteria.
<table>
<thead>
<tr>
<th>Criteria</th>
<th>C1</th>
<th>C2</th>
<th>C3</th>
<th>C4</th>
<th>C5</th>
<th>C6</th>
<th>C7</th>
<th>C8</th>
<th>C9</th>
<th>C10</th>
<th>C11</th>
<th>C12</th>
<th>C13</th>
<th>C14</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>1.00</td>
<td>0.33</td>
<td>0.31</td>
<td>0.39</td>
<td>0.36</td>
<td>0.42</td>
<td>0.51</td>
<td>0.66</td>
<td>0.43</td>
<td>0.42</td>
<td>0.51</td>
<td>0.66</td>
<td>0.41</td>
<td>0.37</td>
</tr>
<tr>
<td>C2</td>
<td>2.04</td>
<td>1.00</td>
<td>0.61</td>
<td>0.74</td>
<td>0.65</td>
<td>0.60</td>
<td>0.76</td>
<td>1.40</td>
<td>0.60</td>
<td>0.83</td>
<td>0.63</td>
<td>0.76</td>
<td>0.72</td>
<td>0.47</td>
</tr>
<tr>
<td>C3</td>
<td>2.55</td>
<td>1.00</td>
<td>0.64</td>
<td>1.00</td>
<td>0.87</td>
<td>0.76</td>
<td>1.00</td>
<td>2.05</td>
<td>0.76</td>
<td>1.00</td>
<td>0.87</td>
<td>1.08</td>
<td>0.92</td>
<td>0.61</td>
</tr>
<tr>
<td>C4</td>
<td>2.18</td>
<td>1.44</td>
<td>1.00</td>
<td>0.92</td>
<td>0.85</td>
<td>0.67</td>
<td>0.91</td>
<td>2.23</td>
<td>0.82</td>
<td>0.82</td>
<td>0.85</td>
<td>0.79</td>
<td>0.80</td>
<td>0.53</td>
</tr>
<tr>
<td>C5</td>
<td>1.64</td>
<td>0.72</td>
<td>0.55</td>
<td>0.72</td>
<td>1.00</td>
<td>0.80</td>
<td>1.28</td>
<td>2.26</td>
<td>0.72</td>
<td>0.89</td>
<td>1.18</td>
<td>1.53</td>
<td>0.87</td>
<td>0.74</td>
</tr>
<tr>
<td>C6</td>
<td>2.38</td>
<td>1.66</td>
<td>1.48</td>
<td>1.66</td>
<td>1.25</td>
<td>1.00</td>
<td>1.75</td>
<td>2.43</td>
<td>1.41</td>
<td>1.30</td>
<td>2.13</td>
<td>2.05</td>
<td>1.56</td>
<td>1.33</td>
</tr>
<tr>
<td>C7</td>
<td>1.00</td>
<td>0.33</td>
<td>0.31</td>
<td>0.39</td>
<td>0.36</td>
<td>0.42</td>
<td>0.51</td>
<td>0.66</td>
<td>0.43</td>
<td>0.42</td>
<td>0.51</td>
<td>0.66</td>
<td>0.41</td>
<td>0.37</td>
</tr>
<tr>
<td>C8</td>
<td>1.00</td>
<td>0.39</td>
<td>0.45</td>
<td>0.50</td>
<td>0.70</td>
<td>1.00</td>
<td>0.56</td>
<td>0.49</td>
<td>0.70</td>
<td>0.80</td>
<td>0.49</td>
<td>0.44</td>
<td>0.44</td>
<td>0.44</td>
</tr>
<tr>
<td>C9</td>
<td>1.00</td>
<td>0.49</td>
<td>0.46</td>
<td>0.68</td>
<td>0.61</td>
<td>0.64</td>
<td>1.16</td>
<td>1.28</td>
<td>0.84</td>
<td>0.59</td>
<td>1.16</td>
<td>1.10</td>
<td>0.62</td>
<td>0.54</td>
</tr>
<tr>
<td>C10</td>
<td>1.00</td>
<td>0.49</td>
<td>0.46</td>
<td>0.68</td>
<td>0.61</td>
<td>0.64</td>
<td>1.16</td>
<td>1.28</td>
<td>0.84</td>
<td>0.59</td>
<td>1.16</td>
<td>1.10</td>
<td>0.62</td>
<td>0.54</td>
</tr>
<tr>
<td>C11</td>
<td>2.04</td>
<td>1.00</td>
<td>0.61</td>
<td>0.74</td>
<td>0.65</td>
<td>0.60</td>
<td>0.76</td>
<td>1.40</td>
<td>0.60</td>
<td>0.83</td>
<td>0.63</td>
<td>0.76</td>
<td>0.72</td>
<td>0.47</td>
</tr>
<tr>
<td>C12</td>
<td>2.55</td>
<td>1.00</td>
<td>0.64</td>
<td>1.00</td>
<td>0.87</td>
<td>0.76</td>
<td>1.00</td>
<td>2.05</td>
<td>0.76</td>
<td>1.00</td>
<td>0.87</td>
<td>1.08</td>
<td>0.92</td>
<td>0.61</td>
</tr>
<tr>
<td>C13</td>
<td>2.18</td>
<td>1.44</td>
<td>1.00</td>
<td>0.92</td>
<td>0.85</td>
<td>0.67</td>
<td>0.91</td>
<td>2.23</td>
<td>0.82</td>
<td>0.82</td>
<td>0.85</td>
<td>0.79</td>
<td>0.80</td>
<td>0.53</td>
</tr>
<tr>
<td>C14</td>
<td>1.64</td>
<td>0.72</td>
<td>0.55</td>
<td>0.72</td>
<td>1.00</td>
<td>0.80</td>
<td>1.28</td>
<td>2.26</td>
<td>0.72</td>
<td>0.89</td>
<td>1.18</td>
<td>1.53</td>
<td>0.87</td>
<td>0.74</td>
</tr>
</tbody>
</table>

Table 6 Criteria-criteria pairwise comparison

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According to the evaluations of subject-matter-experts, as a result of pairwise comparisons, the most effective criteria on selection of tugboat according to the type of propulsion system is safety with a value of 0.130 BNP. Then, bollard pull with 0.118 BNP value, price with 0.081 BNP value, functionality with 0.074 BNP value, speed with 0.069 BNP value, working environment with 0.069 BNP value, operation with 0.065 BNP value, maintenance with 0.064 BNP value, seakeeping with 0.061 BNP, delivery time with 0.060 BNP value, tank capacity with 0.054 BNP value, hull form with 0.053 BNP value, deck arrangement with 0.052 BNP value and maturity possibility with 0.050 BNP value are listed, respectively. Concisely, in this case, it can be seen that C8 and C1 criteria are the most effective factors in the selection of the Tugboat according to the propulsion system. On the other hand, it has been observed that the C14, C5, C6 and C3 criteria less influence on the selection of the tugboat than the other criteria.

Table 7  Criteria evaluation solution results

<table>
<thead>
<tr>
<th>C</th>
<th>W (Weight)</th>
<th>BNP- Best nonfuzzy performance</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(0.117, 0.120, 0.116)</td>
<td>(0.118)</td>
</tr>
<tr>
<td>C2</td>
<td>(0.071, 0.069, 0.068)</td>
<td>(0.069)</td>
</tr>
<tr>
<td>C3</td>
<td>(0.054, 0.053, 0.055)</td>
<td>(0.054)</td>
</tr>
<tr>
<td>C4</td>
<td>(0.062, 0.060, 0.062)</td>
<td>(0.061)</td>
</tr>
<tr>
<td>C5</td>
<td>(0.050, 0.050, 0.054)</td>
<td>(0.052)</td>
</tr>
<tr>
<td>C6</td>
<td>(0.054, 0.053, 0.052)</td>
<td>(0.053)</td>
</tr>
<tr>
<td>C7</td>
<td>(0.069, 0.068, 0.068)</td>
<td>(0.069)</td>
</tr>
<tr>
<td>C8</td>
<td>(0.131, 0.133, 0.126)</td>
<td>(0.130)</td>
</tr>
<tr>
<td>C9</td>
<td>(0.064, 0.064, 0.065)</td>
<td>(0.064)</td>
</tr>
<tr>
<td>C10</td>
<td>(0.067, 0.066, 0.064)</td>
<td>(0.065)</td>
</tr>
<tr>
<td>C11</td>
<td>(0.073, 0.074, 0.075)</td>
<td>(0.074)</td>
</tr>
<tr>
<td>C12</td>
<td>(0.081, 0.082, 0.08)</td>
<td>(0.081)</td>
</tr>
<tr>
<td>C13</td>
<td>(0.057, 0.059, 0.064)</td>
<td>(0.060)</td>
</tr>
<tr>
<td>C14</td>
<td>(0.050, 0.049, 0.051)</td>
<td>(0.050)</td>
</tr>
</tbody>
</table>

As shown in Table 8, among the alternative tugboat types, A1: ASD Type has been identified as the most suitable alternative with 7.255 BNP value and A4: Voith Tractor Type as the most unsuitable with 7.069 BNP value as a common opinion of all experts. The other alternatives were prioritized as Conventional Type as the second preferred alternative with 7.094 BNP value and ASD Tractor as the third preferred alternative with 7.075 BNP value.

Table 8  BNP values according to fuzzy AHP Method

<table>
<thead>
<tr>
<th>Alternative</th>
<th>BNP</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1: ASD type propulsion system</td>
<td>7.255</td>
</tr>
<tr>
<td>A2: Conventional type propulsion system</td>
<td>7.094</td>
</tr>
<tr>
<td>A3: ASD Tractor type propulsion system</td>
<td>7.075</td>
</tr>
<tr>
<td>A4: Voith Tractor type propulsion system</td>
<td>7.069</td>
</tr>
</tbody>
</table>
Applying fuzzy analytic hierarchy process to selection of suitable tugboat according to propulsion/manoeuvring system type for ports in Turkey

It should be noted that, although A1 should have been clearly distinguished from others, BNP values which indicate the order of the selection for other 3 alternatives are very close to each other. In other words, in this case, although determining the best alternative is easy with the BNP value, it is hard to determine the worst, or the last due to the fact that the BNP values of other three alternatives are very close to each other.

5. Conclusion

In this study, Fuzzy AHP methods, type of multi criteria decision making methods, have been applied to determine suitable tugboat according to the type of propulsion system. For this purpose, four alternative tugboat types were considered according to the fourteen criteria by taking into account subject-matter-experts opinions. Created decision models were modelled and evaluated by the technical experts of tugboat operators serving at Turkish ports, the engineers of international tugboat design firms and the expert academicians.

The fuzzy AHP model generated using the obtained data was analysed and the ranking among alternatives and criteria was given. According to the fuzzy AHP method results and expert opinions, it can be seen that C8 and C1 criteria are the most important factors affecting the selection of the tugboat type according to the propulsion system based on BNP values. Besides, C14, C5, C6 and C3 criteria have less effect on the selection than the other criteria.

Considering the impact of weight ratings, C8 and C1 criteria are the most effective factors and C14, C5, C6 and C3 criteria have less influence on the selection of the alternatives, A1: ASD Type has been determined as the most appropriate alternative with the 7.555 BNP value as the common opinion of all subject-matter-experts. The A4: Voith Tractor Type has been selected as the last choice to be preferred with the 7.069 BNP value among four selected alternatives. It should be taken into account that the results of these assessments may vary if the weight of the criteria or the expert changes.

In the future studies, it is possible to develop a new method by changing or increasing / decreasing the criteria used in the evaluations. These methods will help companies that operate in national and international ports decide the type of tugboat they will invest. Similar applications can also be developed for different ship types.

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IMPACT OF WIND LOADS ON LONG-TERM FUEL CONSUMPTION AND EMISSIONS IN TRANS-OCEANIC SHIPPING

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Summary

The influence of weather conditions on fuel consumption and CO₂ emissions for a containership is assessed by calculating the total added resistance, for various sea-states and relative wave directions, through a time domain procedure. The present article extended a formerly published research presented during the IMAM2015 conference [1] providing a deeper insight of the methodology adopted furtherly discussing the results achieved. In particular, the present paper studies the impact of wind velocity and directionality on the ship speed and its relative importance when compared with the wave added resistance. Moreover, considering the most commonly sailed routes in the North Atlantic, the long-term rate of increment of fuel consumption and CO₂ emissions is estimated comparing the resulting ship performance with the wave climate expected for these specific routes.

Key words: fuel consumption; emissions; wind loads; speed loss; containership.

1. Introduction

The demand of international trades is expected to continuously increase in the next years, and so will the relevance of maritime shipping for the global economy and its burden on the ecosystems. In the last decade, the International Maritime Organization (IMO) promoted the exploration of solutions aimed at limiting the air pollution caused by ships in terms of a reduction of emissions of greenhouse gases (GHGs). Possibilities of effective interventions have been identified both in the design and in the operational phases. As a consequence two indices have been developed to serve as a reference for monitoring the emissions and stimulating to pursue innovative solutions. The Energy Efficiency Design Index (EEDI, [2]) is a measure of the ship’s energy efficiency and relates the CO₂ emissions to a nominal transportation work rate depending on the ship type and technical characteristics. In the Energy Efficiency Operational Indicator (EEOI, [2]), instead, the efficiency of a specific voyage is calculated by comparing the emissions to the effective cargo carried and the length of the journey.

At the design phase, the energy efficiency can be improved through the optimization of speed and cargo capacity, propulsion system and the form of the hull and the superstructure, leading to reductions between 10% to 50%. A similar improvement can be obtained in the operational phase by making weather routing, fleet management and efficient logistics be part
of the daily activity of any ship operator. Overall emissions could be lowered of a rate that ranges between 25% and 75% if the two approaches are combined [3]. To ensure future maritime transportations to be cleaner, but also more competitive, a key role has to be played by the shipping industry, placing energy efficiency and environmental protection at a first place since the design phase.

Environmental loads are among the most important factors influencing the fuel consumption of a vessel in navigation. This is clearly shown in [4] where a fuel efficiency model is proposed from the analyzes of log data of container-ships. Not only ship resistance is increased due the effect of waves and wind on the hull and superstructures. Indeed, the efficiency of the propulsion system is also reduced, generally with the effect of requiring more power, decreasing the attainable ship speed and increasing the fuel consumption and emissions. Moreover, ship motions may induce the Shipmaster to voluntary reduce the speed, increasing the complexity of an accurate assessment of ship performance in seaways.

In the past ship design was often focused on meeting the requirements of the trial tests, thus the hull resulted optimized to navigate at the design speed in still water and at full load draft. Such a condition has an extremely low probability to be encountered in real operations. Instead, the ship will frequently be partially loaded, in some cases sailing at different speeds and, as shown in [5], in an environment far from the ideal still water.

A reliable estimation of the attainable ship speed in the actual environment that the ship will encounter is a critical task to support the decision making processes in the maritime economy. It is fundamental, among the others, for the selection of the most favorable route [6] and logistic issues, but also when comparing different designs. All the factors influencing the added resistance must be taken into account for an accurate estimation of ship efficiency on a seaway, as depicted in figure 1 where a scheme of the program is shown.

---

**Fig. 1** Schematic code architecture
The irregular sea and wind loads are simulated in the time domain for the specific sea state. While the effect of waves has been discussed in previous articles of the authors [7], [8], this work focuses on the effect of wind on the involuntary speed reduction, and on fuel consumption. Wind loads on marine structure have been largely studied in the literature. A review of the methods can be found in [9]. In most of the cases, wind is studied for its negative effect on safety and maneuverability [10], [11], especially on ships with large exposed surface when sailing in congested areas or in ports. The combined effect of wind and waves on attainable speed has been studied by [12] with an application to a bulk carrier. In [13] added resistance due to both wind and waves is considered to develop a semi-empirical performance prediction model. An estimation of the extreme wind loads can be found in [14], where four methods have been used to compute ahead force, a side force and the yawing moment. The statistical analysis of noon reports of ships with large exposed area (pure track or car carrier), has been performed by [15]. The fuel consumption and speed curves corresponding to different Beaufort have been estimated, showing an increase of the resistance of 1.6% in head wind compared to side wind. [16] showed that speed loss due to wind may be comparable to the one due to waves for high ship speeds and heavy weather conditions. [17] assessed the resistance and consumption due to wind to account for up to one third of the total ones in challenging weather conditions.

In this paper, the impact of wind loads on the speed loss and fuel consumption is estimated in the time domain considering a pre-defined initial cruise speed and the forces acting on the superstructures due to a simulated realistic wind profile. The attainable ship speed in a given condition is then assessed as the mean values of the resulting ship speed time-series. In section 2 methodology adopted for the simulation of a realistic wind time-series is presented. Section 3 describes the computation of the wind load on the hull above the sea level and the superstructure, and of the effect on the propulsion system. Section 4 underlines the method used to assess the long-term effect of wind in a seaway. The application of the method to a containership is shown in section 5, providing a quantitative assessment of the relevance of wind in the speed drop when compared to waves. Conclusions and final remarks are discussed in section 6.

2. Wind time series simulation

The program allows to compute the ship speed in time affected by a fluctuating wind profile. This is obtained by superposing a turbulent (gust) component to the mean speed, aiming at realistically reproducing the actual condition encountered by a ship. Thus, the generation of the wind trace is fundamental for a reliable estimation of attainable speed and, consequently, fuel consumption.

The horizontal component is described by the mean direction, the mean speed, and its fluctuation aiming at capturing the stochastic nature of wind, continuously varying in time and space. Instead, the vertical component of wind velocity is usually neglected in practical applications.

For a given mean speed, the fluctuation can be represented as a random process with zero mean, distributed in frequency according to a spectrum called the gust spectrum. The description of such a turbulent process is very complex task, which has been largely studied in the literature in order to represent the fluctuating component in terms of a gust spectrum. Proposed formulations include among the others: the Harris (1971) spectrum [18], the Davenport spectrum [19] that derives from measurements on land, [20], [21] and NPD Wind [22]. The program, initially developed in [7] has been extended to include the effect of wind according to the previous considerations. The code supports the calculation of the gust component by applying the formulations of Harris or Davenport.
The Harris formulation has been chosen for the simulation of the wind gust, as recommended by the registers for the design of offshore structures [23]:

\[ f \cdot S(f) = 4\kappa \bar{U}^2 \frac{f'}{\left(2 + f'^2\right)^{5/6}} \]  

(1)

where \( S \) indicates the power spectral density [m^2/s], \( f \) is the frequency [Hz], \( \kappa \) is the surface roughness parameter and \( f' \) is the non-dimensional frequency given by:

\[ f' = \frac{1800f}{\bar{U}} \]  

(2)

The Harris wind gust spectra for different mean wind speed from 5m/s to 50m/s is shown in figure 2. The time series is then computed assigning a random phase to each frequency and the amplitude computed as:

\[ z(f) = \sqrt{2S(f)} \Delta f \]  

(3)

Figure 3 shows three samples of simulated wind trace of the duration of 10 minutes. Mean speeds of 6.9m/s, 15.3m/s and 26.5m/s respectively are considered, corresponding to the significant wave heights of 1m, 5m, and 15m when the Pierson-Moskowitz relation is assumed.

3. Wind loads

Among the environmental loads, wind may give a not negligible contribution, especially in particular conditions of speed and relative direction and for ships with great area above the waterline. From the safety point of view, criticalities are in most of the cases limited to maneuvering or mooring in locations characterized by strong winds and narrow passages. However, a reliable assessment of the attainable speed may also require to take into account the influence of wind.

In regular operations, its influence is generally constrained to an added resistance, that may be positive in case of following wind, and a drift component. Although the drift component may require to adjust the route with an appropriate rudder angle, resulting in increased consumptions, hereafter only the direct influence on added resistance will be taken into account.
A thorough mathematical model can be found, for instance, in [10].

Equation 4 can be applied for the calculation of the longitudinal force imposed by wind:

\[ F_{\text{wind}} = \frac{1}{2} C_{XW} A_T \rho_{\text{air}} U_{\text{rel}}^2 \]  

(4)

where \( C_{XW} \) is the aerodynamic drag force coefficient, \( A_T \) is the frontal wind exposed area, \( \rho_{\text{air}} \) is the air density and \( U_{\text{rel}} \) is the relative wind speed. The increased ship resistance due to the rudder correction necessary to keep the course has been neglected.

The area above the sea-level is typically not considered when tank tests or wave added resistance numerical software which provide information on hull resistance. Nevertheless this exposed area is affected by presence of the wind that, due to the forward ship speed, exist even in the mildest weather conditions. Due to this self-component, both speed and direction of the natural wind must be adjusted when calculating the wind as seen by the ship. In figure 4 a schematic representation of absolute and relative wind direction is provided.

**Fig. 3** 10 minutes wind time-series corresponding to mean speeds of 6.9 m/s, 15.3 m/s and 26.5 m/s

**Fig. 4** Schematic representation of the relative wind
For the computation of the relative wind speed and direction (by convention zero corresponds to following wind) the following equations can be applied:

\[ U_{rel} = \sqrt{u_{rel}^2 + v_{rel}^2} \]  
\[ \psi_{rel} = \arctan2(|u_{rel}|, |v_{rel}|) \]

where \( u_{rel} \) and \( v_{rel} \) are the longitudinal and lateral component respectively given by the equations:

\[ |u_{rel}| = -U_{wind} \cos(\psi - \pi) - V_{ship} \]  
\[ |v_{rel}| = U_{wind} \sin(\psi) \]

Equation 4 moves the problem of calculating aerodynamic force to the critical estimation of the drag force coefficient \( C_{XW} \). Albeit wind tunnel tests (e.g. [24]) are recommended for accurate results, such trials are typically expensive and time consuming. CFD calculations are nowadays possible alternatives [25], nevertheless factors such as the relative direction, the ship draft and the cargo distribution (e.g. in a containership), as well as the relative direction, have an influence on \( C_{XW} \), as shown in [26]. Thus, the applicability of these methodologies to operative problems is limited.

For practical application one can opt to a faster numerical formulation for the estimation of the wind loads. In literature numerous proposals can be found:

- Gould [27], [28] studied effect of wind on the superstructures considering a logarithmic wind profile, aiming at the development a procedure for the estimation of the resulting forces and moments.
- Isherwood [29] based his analysis on several results published in literature relative to a wide range of ships and relative directions and introduced regressions providing lateral and side wind forces and yaw moment.
- Blendermann [30], [31] performed various wind tunnel tests on scale model systematically collecting the results. The statistical analysis of such a database permitted the formulation of expressions to relate wind drag force coefficients (for both forces and moments) to the frontal and lateral projected areas depending on the angle of attack of wind.
- The OCIMF method [32] is suitable for very large crude carriers (VLCCs) and has been developed from statistics on a database of wind tunnel tests.

It is necessary to remember that, while the previous methods can provide an indication for standard cases, in critical situations it is recommended to verify them with experiments. Indeed, often the outcomes are not in agreement as they derive from different assumptions and databases, as shown in [9] where a comparative study is performed.

Isherwood method is used in the this work to estimate the drag force coefficient for the wind directions from 0° to 180° with a step of 30° (see table 2). A spline interpolation is then applied to take into account the actual relative direction at each step.

According to the method proposed by [29], the drag force coefficient can be readily estimated when knowing a number of dimensions of the ship: the overall length \( L_{O\!A} \), the beam \( B \), the lateral and transverse projected areas \( A_L \) and \( A_T \), the length \( S \) of the perimeter of the lateral projection excluding waterline and slender bodies, the distance from bow of the centroid of the lateral projected area \( C \) and the number of distinct groups of masts or kingposts. Thus, the coefficient is given by the equation:

\[ C_{XW} = A_0 + A_1 \frac{2A_L}{L_{O\!A}^2} + A_2 \frac{2A_T}{B^2} + A_3 \frac{L_{O\!A}}{B} + A_4 \frac{S}{L_{O\!A}} + A_5 \frac{C}{L_{O\!A}} + A_6 M \]  

(7)
The actual speed that the ship is able to sustain is computed at each time-step comparing the thrust provided by the propeller and the total resistance including the variating wind force. A B-series propeller [33] is assumed and the required number of revolution (RPM) is calculated according to the relative torque characteristics and the wake fraction. A constant engine torque is considered by accounting for the mass inertia of the ship and engine performance.

Depending on the difference between the initial and the attainable speed, the system may take some cycles to reach a stationary condition, from where the mean speed can be considered as representative of the attainable speed. In [7] more details on the model are provided.

4. Effect on a seaway

The model for the calculation of the forces that wind imposes to the ship has to be matched with appropriate weather information depending on the purpose. If the estimation of wind effect on a near future route is required, output of a numerical weather forecast has to be used. Differently, considering a historical weather database allows assessing the long-term effect (see [34]). In the following the second approach is adopted.

A wave database contains climatological data in a spatial grid covering a specific area, which can be obtained from simulation models, measurements or observations. The state of the art wave models [35], [36] allow to solve the spectral energy equation and assess how the directional wave spectrum propagates in space and time. Due to the continuous evolution of both numerical models and data assimilation schemes differences in the long-term results may in some cases be related to these changes rather than to the natural variability of weather. In order to allow comparisons and significant statistics for a specific long period of time, weather databases are generated by running the same model over the desired period as to overcome the numerical inhomogeneities. Such a process is called reanalysis [37]. It must be noted that the issue of differences in measured data coverage over time cannot be solved.

The state-of-the-art product ERA-interim reanalysis [38] from the European Centre for Medium-Range Weather Forecast (ECMWF) is used in this work. The database has been generated from 1979 and is updated in real time, covering the entire globe. Besides atmospheric variables, the ERA-Interim also includes wave parameters, produced using the ECMWF Integrated Forecasting System by a two-way coupled atmosphere-wave model system. The wave model used is the WAM model with a four-dimensional Variational Data Assimilation (4DVAR) scheme. Observations of ocean wind speeds from VOS, buoys, satellite scatterometer and (when available) satellite altimeter SWHs were assimilated in the analysis process. The database consists of 6-hourly global fields with a 1°x1° grid resolution for the wave parameters. Figure 7 shows the global mean SWH computed from the ERA-interim database.

The statistical multivariate distribution of all variables involved in the process is calculated from the ERA-interim database. When only the wind is considered it reduces to the bivariate distribution of mean wind speed and direction. Instead, when the effect of waves is also included, it becomes a five-variate distribution of significant wave height (Hs), wave peak period (Tp), mean wave direction θ, mean wind speed (Ws) and direction (WD). In this work a different probability distribution has been calculated for each one of the routes analysed considering the effective climate on their path, making the results more detailed.

When both wind and waves are taken into account, the long term influence on the speed reduction is computed assuming the superposition of the effects, such as:

\[
\Delta V = \sum_{H_s,T_p,\theta,W_s,W_D} p_i \left( \Delta V_{H_s,T_p,\theta} + \Delta V_{W_s,W_D} \right)
\]

where \(\Delta V_{H_s,T_p,\theta}\) and \(\Delta V_{W_s,W_D}\) are the speed drops due to waves and wind respectively and \(p_i\) is the probability of encountering the specific weather condition.
Knowing the attainable speed of the ship, the average fuel consumption on the seaway is given by:

$$\text{FOC} = C_{\text{conv}} \cdot \frac{\text{SFOC} \cdot \text{BP}}{\nu} \cdot L_{\text{route}}$$  \hspace{1cm} (9)$$

where SFOC is the specific fuel oil consumption in g/kWh, BP is the engine brake power in kW, \(\nu\) is the ship speed in knots, \(L_{\text{route}}\) is the length of the route in nautical miles and \(C_{\text{conv}} = 10^{-6}\) is a conversion coefficient to obtain the FOC in metric tonnes.

Strictly, when the ship drag is increased, also the propeller load is increased, thus, if the engine torque is constant, the engine speed \(\nu\) will drop meaning the engine power would also drop. Furthermore, every change would also result in the change of the SFOC. For slight changes, however, the engine working point does not undergo large variations, thus constant power condition may be assumed and the SFOC may be considered constant as well.

Being the present work focused on the long-term effect on fuel consumption and emissions rather than on operational or extreme conditions, the inaccuracy introduced do not have a significant impact on the results. Heavy weather conditions, in fact, are encountered very seldom due to the tendency of ship to avoid storms. As an example, in the most severe route considered (denoted as Ch_VA in figure 7), a ship has about 8% probability to encounter a significant wave height greater than 5m.

5. Numerical example

The S175 containership has been used to test the developed code. The cruise speed set as initial condition is 21 knots and requires from the main engine 27.5MW to be sustained in calm water. This condition corresponds to a fuel oil consumption (FOC) of 224kg/nmi. In table 1 the main dimensions of the ship are listed, while the drag force coefficients calculated according to Isherwood method are shown in table 2.

In figure 5 the attainable ship speed for different relative direction is plotted in function of the absolute wind speed. When wind speed is moderate, namely lower than 6-8 m/s the influence of the self-wind component caused by the ship speed is predominant. The impact of natural wind can be appreciated for stronger wind speeds, although in the considered conditions a decrease of 0.5 knots at most can be found.

**Table 1** Main dimensions of the S175 containership

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars</td>
<td>175.0 m</td>
</tr>
<tr>
<td>Breadth moulded</td>
<td>25.4 m</td>
</tr>
<tr>
<td>Design draft</td>
<td>9.5 m</td>
</tr>
<tr>
<td>Lateral projected area</td>
<td>3710 m(^2)</td>
</tr>
<tr>
<td>Transverse projected area</td>
<td>756 m(^2)</td>
</tr>
</tbody>
</table>

**Table 2** Aerodynamic drag force coefficient for different relative wind directions

<table>
<thead>
<tr>
<th>Wind dir.</th>
<th>(C_{\text{WX}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-0.136</td>
</tr>
<tr>
<td>30</td>
<td>-0.230</td>
</tr>
<tr>
<td>60</td>
<td>-0.146</td>
</tr>
<tr>
<td>90</td>
<td>-0.025</td>
</tr>
<tr>
<td>120</td>
<td>0.192</td>
</tr>
<tr>
<td>150</td>
<td>0.313</td>
</tr>
<tr>
<td>180</td>
<td>0.146</td>
</tr>
</tbody>
</table>
The wind direction of 150° is selected, being the direction from where wind has the most important effect on speed loss, to compare it with the effect of waves. The results are shown in figure 6 for wind and wave separately, as well as their combined effect [39].

The case of fully developed sea-states is considered, that is wind speed and SWH are considered correlated through the Pierson-Moskowitz relation, such as:

\[ \text{SWH} = 0.22 \left( \frac{u_{\text{wind}}^2}{g} \right) \]

(10)

It is thus assumed that wind and waves are aligned, which is generally true for developing seas, but may be not the case in the open ocean on a swell dominated sea-state.
The red and blue curves show a stabilization and small increase of the ship speed when $H_s$ exceeds 8 meters. The reason can be attributed to the use of a one parameter spectral model. When wave heights increase, wave periods does the same, eventually resulting in lower motions and loads on the hull.

Except for very low SWH, the wind effect on reducing ship speed in operations is in general minor when compared to the total environmental loads. This effect settles on values of about 6% for sea-states characterized by SWH higher than 5m.

Fig. 8 Rate of increment of fuel consumption and CO$_2$ emissions due to environmental factors in the main North Atlantic routes.
A database derived from [40] has been here used for the estimation of the long-term effect on fuel consumption and CO\textsubscript{2} emissions. It describes the real weather experienced by ships navigating on the routes identified in [5] and showed in figure 7. Figure 8 shows the results of such analysis in terms of percentage of increment, with respect to calm weather, caused by the wind. The sole effect of waves and the combination of both environmental factors are also given for comparison. The portion due to wind has an overall contribution of about 0.4\% of increment of FOC, which, although small can be not negligible when detailed analysis is performed. Moreover, this rate refers to open waters, characterised by a significant predominance of swell or fully-developed sea-states. It can be expected to increase when routes on marginal seas are considered, such as Mediterranean or Baltic.

As it was expectable, northern routes, which include the areas where extratropical storms generate, are those more conditioned by environmental factors. In these cases, an increase of 6\% in the FOC can be accounted in the long term. This is not the case for the southern routes and especially the one between the Strait of Gibraltar and Miami where the influence of weather is lower than 2\%.

6. Conclusions

The impact of wind load on ship speed loss, fuel consumption and CO\textsubscript{2} emissions has been studied for different weather that ships may encounter. Furthermore, the long-term effect for some important trans-oceanic routes has been estimated.

For a given weather condition a realistic wind time-series is simulated by considering appropriate wind gust spectra. Then the ship speed is calculated in the time domain considering step by step the actual wind load affected by the relative speed and direction. Finally, the mean of the resulting speed profile is considered representative of the mean attainable speed. The long-term influence of environmental loads on speed reduction, fuel consumption and emissions is studied by considering the actual weather that a ship faces while sailing a specific route. For this purpose, the join probability of encountering determined weather conditions has been taken into account through the analysis of the climate which characterises the main North Atlantic routes.

Within 0° and 60°, wind pushes the vessel and, in hypothetical calm waters, allows the conservation of the cruise speed and its minor increment for stronger winds (indicatively with speeds higher than 14m/s). The worst conditions are for wind from 120° and 150° due to the loading of containers above the deck which extends the area exposed to wind, resulting in an increase of the drag force coefficient. Even in these cases, however, the speed drop is limited to less than one knot.

The increment in the fuel consumption and CO\textsubscript{2} emissions only due to wind appears to be small. On the other hand, when it is neglected it leads to an underestimation of roughly 0.6\%. As a regerence, it means as much as 5 tonnes of fuel in a trans-oceanic passage as an average.

The results show that long-term effect of wind is much lower than the one of waves, and its low percentages may follow within the general uncertainties of environmental loads when considering the lifetime of a ship. Different results could come if specific operations are considered. For instance, for ships sailing in storms at low speed, in congested or restricted waters or in developing sea-states typical of marginal seas, a greater impact of wind can be expected.

In the present study wind speed and SWH are correlated according to the Pierson-Moskowitz relation, that is fully developed sea-states are considered. Due to this assumption, the effect of wind in marginal seas or sheltered areas, where the occurrence of developing seas is higher, can be underestimated. To take into account these conditions, further studies
excluding the strict dependency of wind and waves are necessary. Nevertheless, due to the high computational demand of such analysis with the time-domain program used, they have not been carried out in the present work.

The results confirm the expectations on the effect of wind loads and its proportion with respect to wave loads. Nevertheless, due to the minor relative influence of wind on speed loss, the differences between the approach presented in this paper and traditional methodologies is expected to be negligible. Taking into account the higher complexity of the technique here proposed, this is not expectable to supersede traditional methods which can be considered sufficiently reliable for the scope of ship operations.

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Impact of wind loads on long-term fuel consumption
and emissions along the main North Atlantic routes

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ON SELF-PROPULSION ASSESSMENT OF MARINE VEHICLES

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Original scientific paper

Summary

Estimation of ship self-propulsion is important for the selection of the propulsion system and the main engine so that the ship can move forward with the required speed. Resistance characteristics of the vessel or the open-water performance of a propeller only are not usually enough to assess the working conditions of the ship. Both in numerical simulations and in experiments; there is a need to treat the propulsion system and the hull as a whole for a better estimation of the self-propulsion parameters. In this study, the self-propulsion points of one submarine (DARPA Suboff) and two surface piercing vessels (KCS and DTC) were obtained with methods based on computational fluid dynamics (CFD) approach. The self-propulsion points were also calculated by a classical engineering approach that makes use of the empirical relations that may be found in the literature. The results were evaluated with respect to the experiments and numerical results generated by other researchers in this field. It was found that the self-propulsion points of traditional ship forms can be very closely approximated with a classical engineering approach, given the basic geometric and the hydrostatic properties of the hull and the propeller.

Key words: self-propulsion point; DARPA; KCS; DTC; virtual disk; moving reference frame

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>B</td>
<td>Ship breadth</td>
</tr>
<tr>
<td>C_b</td>
<td>Ship block coefficient</td>
</tr>
<tr>
<td>C_M</td>
<td>Ship midsection coefficient</td>
</tr>
<tr>
<td>C_F</td>
<td>Skin friction resistance coefficient</td>
</tr>
<tr>
<td>C_P</td>
<td>Pressure resistance coefficient</td>
</tr>
<tr>
<td>C_T</td>
<td>Total resistance coefficient</td>
</tr>
<tr>
<td>D</td>
<td>Propeller diameter</td>
</tr>
<tr>
<td>F_r</td>
<td>Froude number</td>
</tr>
<tr>
<td>J</td>
<td>Propeller advance ratio</td>
</tr>
<tr>
<td>K_T</td>
<td>Propeller thrust coefficient</td>
</tr>
<tr>
<td>K_Q</td>
<td>Propeller torque coefficient</td>
</tr>
<tr>
<td>L_pp</td>
<td>Length between perpendiculars</td>
</tr>
<tr>
<td>n</td>
<td>Propeller rotation rate</td>
</tr>
<tr>
<td>R_T</td>
<td>Total resistance of the bare hull</td>
</tr>
<tr>
<td>R_e</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>S</td>
<td>Wetted area of the hull</td>
</tr>
<tr>
<td>t</td>
<td>Thrust deduction factor</td>
</tr>
<tr>
<td>T</td>
<td>Thrust generated by the propeller</td>
</tr>
<tr>
<td>T_m</td>
<td>Ship mean draft</td>
</tr>
<tr>
<td>U_g</td>
<td>Grid uncertainty</td>
</tr>
<tr>
<td>U_N</td>
<td>Total numerical uncertainty</td>
</tr>
<tr>
<td>U_V</td>
<td>Validation uncertainty</td>
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</table>
1. Introduction

Predicting the self-propulsion point of a marine craft is a challenging issue because of its geometry and complex hydrodynamic interaction between the ship and the propeller. Computational Fluid Dynamics (CFD) approach gives an opportunity to simulate the flow around a ship and propeller in a more economical way compared to the experimental methods. In the last three decades, researchers became capable of simulating ship flow with higher mesh numbers and smaller time steps thanks to the rapid development of computing technology. Several works are present about hull-propeller interaction and self-propulsion calculations using CFD approach. Potential flow based CFD methods are still widely used to numerically simulate the flow around ships and propellers. On the other hand, recent research shows that Reynolds Averaged Navier-Stokes Equations (RANSE) based methods started becoming a spearhead for ship resistance and propeller flow simulations. Though not very common, coupled methods utilizing the boundary element method (BEM) and RANSE also get a foothold in recent studies. Although CFD is the most popular method for propulsion simulations lately, experiments which are the workhorse of the industry are still touted as the most trusted option. In a recent study, the effect of the propeller on the stern region was experimentally examined by Pecoraro et al. using Laser-Doppler Velocimetry (LDV) [1].

Experiments are also used for validating numerical approaches. Bugalski and Hoffmann found accordance in ship self-propulsion results generated by RANSE and experiments using sliding grids [2]. Theoretical derivations are also useful to assess the efficiency of numerical methods. Actuator disk method has proved its worth in time for ship propulsion simulations. A comparison of RANSE and actuator disk methods with respect to the experimental data was made by Gao et al. and advantages of each method were demonstrated [3]. A coupled approach joining field methods (RANSE) and panel methods (BEM) were used to reduce high computational costs of ship self-propulsion simulations. Forces acting on the ship hull should be calculated with RANSE to include viscosity while potential flow approach implementing BEM is enough to calculate the thrust generated by the propeller. This model was first proposed by Stern et al. and various applications are present in the literature such as the study of Berger et al. [4, 5]. The model was based on a body force approach where additional terms were added to the Navier-Stokes equations to model the interaction. Gaggero et al. performed a successful ship self-propulsion prediction by utilizing a coupled approach which was developed by Villa et al. [6, 7]. Starke and Bosschers also made use of this coupled approach and discussed the scale effects in ship resistance and propulsion [8].
Model scale CFD simulations are more common due to the difficulty of high Reynolds numbered full scale simulations. Castro et al. performed CFD simulations for predicting the full scale KRISO Container Ships (KCS) self-propulsion [9]. Grid structure has an intense effect on propulsion simulations especially in resolving the flow field; however, Da-Qing stated that computing the forces acting on the propeller can even be found with a coarse mesh structure [10]. Self-propulsion is also a critical issue for submarines. Chase and Carrica studied on DARPA Suboff and its propeller E1619. RANSE, DES (Detached Eddy Simulation), DDES (Delayed Detached Eddy Simulation) and NTM (No Turbulence Method) approaches were used and investigated for comparison [11].

Recently, overlapping grids for the propeller section were started to be used effectively in RANSE propeller simulations. Chao et al. simulated an ice flow field in front of a propeller and investigated the effects of gap between the ice and the propeller on the propeller hydrodynamic coefficients by implementing an overlapping grid structure around the propeller [12]. Another approach to numerically simulate a rotating propeller during a ship’s surge motion is using the sliding mesh. Wang et al. dealt with the influences of the skew angle variance on propulsion performance and the trailing vortex wake [13]. They utilized RANSE based CFD method with sliding mesh technique for open water propeller simulations of a series of DTMB propeller models and achieved compatible results with the experiments. They concluded that increment of skew angle has benefits on hydrodynamic performance of DTMB series. Go et al. numerically examined the effects of a duct using KP505 propeller [14]. Different duct diameter and angle of attack conditions were simulated after a validation study. Detailed post-process illustrations were presented about the hydrodynamic effects of a duct on the propeller.

The present study deals with the estimation of self-propulsion points of vessels using numerical simulations and empirical relations. Numerical simulations can be made for single vessel and single propeller cases to model the interaction via propeller wake and thrust deduction factor. In that case, simulations are conducted for isolated hull and isolated propeller. Propeller wake and thrust deduction factor may be obtained computationally, numerically or via empirical relations and statistical regressions etc. Numerical simulations can also be made for self-propelled case, where the vessel and the propeller are modelled together to predict the self-propulsion of the ship. Another option is to use a basic engineering approach and making use of some empirical relations that may be found in the open literature for the estimation of the self-propulsion point. These methods were used to predict the propulsion performances of the DARPA Suboff, KRISO Container Ship (KCS) and the Duisburg Test Case (DTC). Experimental and numerical results of other researchers for these benchmark ships exist in the literature and comparisons were made with them where applicable.

2. Numerical estimation methods of ship self-propulsion point

Ship self-propulsion point can be estimated by experimental or numerical self-propulsion tests. The self-propulsion point of a ship can also be predicted if bare hull resistance and open-water propeller performance are known. The communication between the total bare hull resistance and open-water propeller performance can be set up via propeller wake and thrust deduction factor.

A working propeller will change the flow, especially at the stern region of the hull. The effect of the propeller on the total resistance will be accounted to the thrust deduction factor. The existence of a hull in front of the propeller will change the incoming flow on the propeller disk. This will be accounted to the propeller wake. Therefore it may be said that the hull-propeller interaction is governed by two non-dimensional parameters, namely the propeller
wake, \( w \) and the thrust deduction factor, \( t \). If experimentally or numerically the self-propelled case is not tested, then the interaction should be investigated using these two parameters. In this section, the different methods used in this study to predict the self-propulsion of ships were presented.

2.1 Estimation of self-propulsion point by a classical engineering approach

Given the geometric and hydrostatic properties of a hull and a propeller; an engineer can specify the self-propulsion point of that hull-propeller system with a basic engineering approach. The engineer also requires the following to assess the propulsive characteristics of the ship:

- Total towed resistance (with no propeller), \( R_T \): The total ship resistance can be obtained from model experiments, computational fluid dynamics approach or using empirical relations.
- Wake fraction, \( w \): Wake fraction of a hull can be obtained from model experiments. It can also be obtained by calculating the axial velocity \( V_A \) that the propeller receives by CFD and using equation (1). There are also some empirical equations provided in the literature such as the one suggested by IMO [15].
- Open-water thrust coefficient, \( K_T \): Thrust coefficient of a propeller can be obtained experimentally from open-water propeller tests. Open-water propeller tests can also be numerically simulated using field (finite volume) or panel (boundary element) methods.
- Thrust deduction factor, \( t \): The thrust deduction factor can experimentally be determined. Experiments should be conducted for the hull with and without the propeller both and they should be used in equation (4) (or equation (5) if the experiments are made for the model scale). Same procedure can also be followed numerically to obtain \( t \). Numerical approach can either be carried out by modelling the propeller itself or using a virtual disk to represent it. Empirical relations such as the one proposed by IMO serve another option [15].

As briefly explained above, these values can be empirically calculated if available in the literature. Using the results of experiments or computational fluid dynamics approach are other options. With these parameters in hand, the methodology would be as follows:

a. Using wake fraction \( w \), calculate the axial velocity \( V_A \) that the propeller receives using the equation:

\[
    w = \frac{V_S - V_A}{V_S} \tag{1}
\]

It must be noted here that the wake fraction provided in equation (1) is the \textit{Taylor wake fraction in general}. The \textit{Taylor wake fraction in thrust identity} defined by the ITTC is different and defined in ITTC [16]. Axial velocity \( V_A \) can also be directly calculated if experimental or numerical methods are followed.

b. Select an arbitrary value of propeller rotation rate \( n \) and calculate the advance ratio \( J \), using \( V_A \) obtained from equation (1) with the equation:

\[
    J = \frac{V_A}{nD} \tag{2}
\]

c. Obtain thrust coefficient \( K_T \) at the advance ratio \( J \) (obtained from equation (2)) using the open water propeller performance curve. Intermediate values of \( J \) can be calculated by interpolation or by fitting an equation.
d. Determine the thrust $T$ using the thrust coefficient $K_T$ from the open-water results using the equation:

$$K_T = \frac{T}{\rho n^2 D^4} \quad (3)$$

e. If the calculations are carried for the full scale ship, the total towed resistance $R_T$ should be equal to the thrust $T$ generated by the propeller times $1 - t$. This relationship is more widely known with the equation:

$$t = \frac{T - R_T}{T} \quad (4)$$

However, if the calculations are carried out for a model ship, then a skin friction correction (SFC) must be made. This is due to the fact that the frictional resistance coefficients of the model and the full scale ship are not equal. This is explained in greater detail in ITTC [16]. Equation (4) in this case becomes:

$$t = \frac{T - R_T + SFC}{T} \quad (5)$$

f. If $R_T - SFC \neq T(1 - t)$, return to point b to select another propeller rotation rate $n$. It should be noted that $SFC = 0$ if the experiments are made for the full scale ship. Skin friction correction equation is given as:

$$SFC = \{(1 + k)(C_{F0M} - C_{F0S}) - \Delta C_F\} \cdot \frac{1}{2} \rho S V_S^2 \quad (6)$$

The flow diagram of the procedure explained above is given in figure 1.

---

**Fig. 1.** Flow diagram of the classical engineering approach adopted in this study.
The flow diagram presented in figure 1 reflects the authors’ choice to obtain self-propulsion parameters at the initial design stage. There are many other methods to approach the problem such as making use of the auxiliary quantity $K_T/J^2$ as in reference [6].

2.2 Self-propulsion approximation using “virtual disk”

The self-propulsion point of a marine vehicle can computationally be estimated without directly modeling the propeller itself. A virtual disk that totally covers the propeller geometry is created at the position of the propeller and it uses the propeller’s open-water performance characteristics. Figure 2 summarizes how a virtual disk is created in numerical simulations.

It must be mentioned that the virtual disk in figure 2 (right) does not contain the propeller geometry. It is only a cylinder that is “considered” to model the propeller. Implementation of virtual disk is handy at the preliminary design stage of a ship due to the following:

- handling the geometry is relatively easier at the CAD stage and
- reduces the number of elements needed to physically model the propeller.

However it must be noted that the open-water propeller performance of the propeller is a prerequisite to use the virtual disk. All self-propulsion simulations with virtual disk in this study are based on the body force propeller method. The method does not take propeller swirl into account and therefore it is advised to be used when there is no need to solve the flow in the vicinity of the propeller. The method is especially handy when one needs to understand hull-propeller interactions instead of resolving the flow field in the wake. A recent study utilized this method to investigate hull-propeller interactions through simulations of self-propulsion tests [17].

![Fig. 2. Hull without a propeller (left). Hull with a propeller (middle). Hull with a virtual disk at the position of the propeller (right). The propeller is not present in the virtual disk.](image)

2.3 Numerical simulation of self-propulsion tests

The hull-propeller system can be directly simulated numerically to determine the self-propulsion point of a ship. This method is of course more realistic as the real geometry of the propeller is included in the simulations. There are various ways to directly simulate the hull-propeller system and the initial conditions of the simulation should be selected accordingly:

- Propeller starts its rotation when the ship has zero forward speed.
  
  If this approach is selected, then the simulation must be ran in transient mode. Initially, the propeller rotation rate is given but the ship is stationary. The ship starts moving and after a while it reaches a steady forward speed with the thrust generated by the propeller.

- Propeller starts its rotation when the ship is moving with a specified forward speed.
  
  In this approach, force balance should be closely investigated. The thrust generated by the propeller should be equal to that produced by the total resistance of the hull. A time-independent approach may be implemented using a moving reference frame to model the propeller rotation.
In this study, the second option is selected to obtain the self-propulsion point directly using CFD. Therefore, the force balance was closely observed and the propeller rotation rate was iteratively found for a specified velocity of the ship.

3. Numerical implementation

A commercial software, Star CCM+ was used to numerically simulate all the cases involved in this study. Due to flows having high Reynolds numbers, a turbulent flow approach implementing the $k-\epsilon$ turbulence model was selected. $y^+$ values were checked and were in accordance with the requirements of the selected turbulence model. The free surface boundary was tracked using the Volume of Fluid (VOF) approach where applicable. The DARPA suboff was considered to be totally submerged and far from free surface; therefore, a single phase (containing water as the only fluid in the domain) numerical approach was adopted.

To assess the propulsive characteristics of ships easily, a virtual disk surrounding (but not including) the propeller is created and open-water propeller performance was imposed on the solver. Virtual disk provides easier generation of the model and uses lower computer memory due to the lesser number of elements used. For the cases of ships including the propeller, a moving reference frame (MRF) method was used. MRF allows solving the ship-propeller interaction problem via a quasi-transient approach. The propeller remains fixed in the flow but the domain just surrounding the propeller is given a rotation instead, which accelerates the fluid particles in the selected region. A sample grid system around the DARPA Suboff with a propeller is given in figure 3. The propeller is inside the cylindrical domain at the stern part of the hull. Some more details about the moving reference frame is explained in the related section.

![Fig. 3. Grid system on the DARPA Suboff surface.](image)

![Fig. 4. Grid structure around the underwater hull of KCS. Stern (left), bow (right).](image)

To correctly capture the waves generated by the hull for surface piercing ships, a Kelvin-wave refinement is made. This refinement can clearly be seen in figure 4 (left), starting from around the midship and extending $2L$ to the wake region. The reason of not extending the Kelvin wave refinement region up to the bow region of the ship was to save from computational time and memory. The aim of this study was to focus on the self-propulsion characteristics of ships;
therefore, the number of elements used in simulations was tried to be optimized. Only the places where the propeller would get affected were refined. Refinements over the hulls were also made to places where pressure gradients were expected to be high like the position of the bulbous bow in figure 4 (right). For better approximation of viscous forces, at least 4 prism layers were used on the hulls.

Moving reference frame

Moving reference frame offers a time-averaged solution rather than a time accurate one. The propeller is held stationary while the surrounding block is given a rotation. This rotation of the block is not a real one, as the grid elements are always stationary throughout the simulation. However, the flow is being rotated in the opposite direction of the actual direction of propeller rotation. The flow rotation defines relative velocities and generates flux for each grid in the block surrounding the propeller. An increase in the propeller rotation rate is reflected as an increase in the flux at each grid. The communication with the outer fluid domain is made by an interface in between the two regions. For a diagrammatic explanation of the moving reference frame, figure 5 may be referred.

A good advantage of the moving reference frame is the flexibility of this method to be used in steady state approaches. In this study, all self-propelled CFD simulations were performed using moving reference frame in steady state. The relative positioning of the propeller would not be dominant in results if one is to obtain quantitative data of the propulsion system of a ship such as thrust or torque coefficients, thrust deduction factor, propeller efficiency etc. The transient behavior of ship-propeller interaction would be significant in cases such as cavitation phenomenon or noise generation. The propeller rotation rates were all low in the cases investigated in this study; therefore, cavitation is left out of the simulations. Moving reference frame provides a faster and an efficient way to simulate thrusters such as ship propellers. The method is widely used in numerical open-water propeller simulations [18]. In a recent article, moving reference frame is used extensively to estimate the self-propulsion parameters for a bulk carrier [19]. However, a major drawback of this method is that it can give different results for different relative positions of the propeller blades.

![Fig. 5. Working principle of the moving reference frame.](image)

4. Uncertainty of numerical simulations

An uncertainty estimation of numerical simulations was made using the appended form of the DARPA Suboff and using the CFD verification and validation methodology of Stern et al. [20]. To get a deeper insight on all the uncertainty parameters presented in this section, please refer to the reference article [20]. The total resistance coefficient was taken as the integral variable. The propeller was not present in the simulations and the speed of the vessel was taken
as 10knots. The experimental results of resistance are present in Liu and Huang [21]. The numerical simulations were made using the steady state time assumption and the iterative uncertainties were very low compared to the grid uncertainty. Therefore, the total numerical uncertainty was roughly taken in this study as $U_N \approx U_G$. The total resistance coefficient was obtained using three grid systems and the results are presented in table 1 along with the experimental results.

### Table 1. Total resistance coefficients obtained with different grids.

<table>
<thead>
<tr>
<th></th>
<th>Experiment</th>
<th>GRID 1</th>
<th>GRID 2</th>
<th>GRID 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of elements</td>
<td>-</td>
<td>145k</td>
<td>411k</td>
<td>1062k</td>
</tr>
<tr>
<td>$C_T$</td>
<td>3.297*10^{-3}</td>
<td>3.511*10^{-3}</td>
<td>3.192*10^{-3}</td>
<td>3.050*10^{-3}</td>
</tr>
</tbody>
</table>

The estimation of numerical uncertainty was made using grid 2 which has a total number of 411k grid elements. The grid refinement ratio was taken as $r_G = \sqrt{2}$. Following the total resistance coefficient results given in table 1, the parameters of uncertainty for the verification part were calculated as presented in table 2.

### Table 2. Verification parameters for numerical simulations.

<table>
<thead>
<tr>
<th>$\varepsilon_{32}$</th>
<th>$\varepsilon_{21}$</th>
<th>$R_{G_2}$</th>
<th>$r_G$</th>
<th>$p_G$</th>
<th>$\delta_{RE}$</th>
<th>$C_G$</th>
<th>$U_G$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.418*10^{-4}</td>
<td>3.192*10^{-4}</td>
<td>0.444</td>
<td>1.414</td>
<td>2.341</td>
<td>2.551*10^{-4}</td>
<td>1.251</td>
<td>3.832*10^{-4}</td>
</tr>
</tbody>
</table>

From table 2, it can be seen that $U_G = 12\% S_{G_2}$. The parameters for the validation part of the uncertainty study are presented in table 3.

### Table 3. Validation parameters for numerical simulations.

<table>
<thead>
<tr>
<th></th>
<th>$S_{G_2}$</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiment</td>
<td>3.297*10^{-3}</td>
<td>3.192*10^{-3}</td>
</tr>
</tbody>
</table>

Fig. 6. Wall y+ on DARPA Suboff at 10knots speed.
The error for grid 2 as compared with the experimental result was found to be $E = 3.17\%$ $D$. The experimental uncertainty was not provided in the reference study; therefore, the validation uncertainty was taken as $U_e \approx U_N$. The numerical uncertainty was $U_N = 11.62\%$ $D$. The error was smaller than the validation uncertainty $E < U_e$ and the numerical simulation results were validated. Wall $y+$ values for grid 2 at a vessel speed of 10knots is given in figure 6.

5. DARPA Suboff self-propulsion results

The self-propulsion points of DARPA Suboff at different velocities were determined using the three different methods explained in this paper. For the self-propelled CFD results, INSEAN E1619 propeller was fitted to the DARPA Suboff similar to [11] and the results were compared where applicable. The geometric properties of the DARPA Suboff and the INSEAN E1619 propeller are given in table 4 and table 5 respectively [11].

<table>
<thead>
<tr>
<th>Table 4. Geometric properties of the full scale DARPA Suboff.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hull Length</td>
</tr>
<tr>
<td>Hull Diameter</td>
</tr>
<tr>
<td>Propeller Diameter</td>
</tr>
</tbody>
</table>

The propeller diameter for the DARPA Suboff was 0.262m as stated in table 4, whereas the original E1619 propeller diameter was 0.485m. Therefore, a $\lambda = 1/1.8512$ scaled model of the E1619 propeller was fitted at the stern part of the hull to numerically solve for self-propulsion directly by CFD. For the case with the virtual disk, a cylinder having a diameter of 0.262m was placed at the location of the propeller instead. Calculations were first carried out for no propeller case for two purposes. The first was for the validation of the results with the reference experiments and the second was to gather data (such as the wake fraction $w$ and total towed resistance $R_T$) for the classical engineering approach (CEA) to estimate the self-propulsion point of the DARPA Suboff. The vessel was considered to be totally submerged in water; therefore, free water surface effects were not included in the numerical simulations.

<table>
<thead>
<tr>
<th>Table 5. Geometric properties of the E1619 propeller fitted to DARPA Suboff.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Blades</td>
</tr>
<tr>
<td>Propeller Diameter</td>
</tr>
<tr>
<td>Hub Diameter</td>
</tr>
<tr>
<td>Pitch at r=0.7R</td>
</tr>
<tr>
<td>Chord at r=0.7R</td>
</tr>
</tbody>
</table>

5.1 No propeller case

Numerical simulations were carried out to assess the total resistance of the DARPA Suboff and the results were compared with the experiments of Liu and Huang [21]. The comparisons were made for both the bare hull (configuration 1) and fully appended (configuration 8) cases. The results are given in figure 7.

Prediction of total resistance for both configurations of the DARPA Suboff is satisfactory compared to the experiments. There is a small deviation in results for the fully appended case but the results are nearly on top of each other for the bare hull case. The resistance components
are given in figure 7 (right) for the fully appended case. Pressure resistance is dominant over frictional resistance for all velocities.

**Fig. 7.** Total resistance (left) for both configurations and resistance components (right) of DARPA Suboff for configuration 8 at different speeds.

**Fig. 8.** Wake fraction with respect to changing hull velocity.

Changes in the wake fraction with respect to speed are given in figure 8. The wake fraction was calculated by equation (1). The wake fraction showed a decreasing trend with respect to increasing hull velocity. Conventional self-propulsion calculations assume wake fraction to be constant but in fact there is a slight decrease in $w$ with increasing ship speed. Changes in $w$ might become important if the ship speed increases (or decreases) dramatically. The contours of the nominal wake for two speeds are given in figure 9. Although it is very hard to distinguish between wake contours in this figure, the wake fraction was remarkably lower at higher speed.
5.2 The case with the propeller

The case with the propeller was numerically carried out to assess the thrust deduction factor \( t \) and the propeller rotation rate \( n \). The assessment of the propeller rotation rate \( n \) is made in the self-propulsion section. The thrust deduction factor given in figure 10 was calculated by equation (4) using two different methods. The thrust \( T \) in equation (4) can numerically be obtained either by using a virtual disk to represent the propeller or modeling the propeller directly. The thrust deduction factor obtained from the virtual disk method was consistently higher than the self-propelled CFD method in all the speeds covered in this study. The underlying physical issue for this miscalculation cannot be foreseen; however, the ship reaches the self-propulsion equilibrium (thrust – resistance equilibrium) later than expected. This is probably based on the actuator disk theory where it is assumed that the disk has zero thickness. In the actual case, propeller blades have a non-negligible thickness which reduce the thrust achieved by the propeller. The self-propelled CFD method showed an increasing trend with increasing velocity whereas the results generated by the virtual disk method were nearly constant.

![Nominal propeller wake at different velocities of DARPA Suboff](image)

**Fig. 9.** Nominal propeller wake at different velocities of DARPA Suboff.

![Thrust deduction factor obtained from different numerical approaches](image)

**Fig. 10.** Thrust deduction factor obtained from different numerical approaches.
5.3 Estimation of self-propulsion point

The estimation of self-propulsion point of the DARPA Suboff was made using three different approaches. The first one was by using the CEA while the second and the third were numerical. The second approach was solving the self-propulsion problem by using a virtual disk to represent the propeller and the third was by direct modeling of the propeller with CFD approach. The obtained results using the three different methods were compared with the results of Chase and Carrica, Ozden and Celik [11, 22] and Sezen, Delen and Bal [23]. They are given in table 6.

In table 6, OWC denotes the open-water curve. All the calculations carried out to estimate the self-propulsion points given in table 6 were made using the experimental OWC. The thrust deduction factor $t$ that was essential to calculate the self-propulsion via the CEA was obtained from the numerical simulations using a virtual disk. Other options were to use the $t$ from self-propelled CFD or empirical equations. Considering that large deviations in the thrust deduction factor reflect as a minor change in the advance coefficient $J$, carrying out extra calculations were not found to be necessary (a 50% change in $t$ only reflects as a 3% change in $J$).

**Table 6.** Self-propulsion assessment of the DARPA Suboff with E1619 propeller at $V = 5.35\text{knots}$.

<table>
<thead>
<tr>
<th></th>
<th>$J$</th>
<th>$K_T$</th>
<th>$K_Q$</th>
<th>$\eta_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chase and Carrica [11] - Self-propelled CFD</td>
<td>-</td>
<td>0.2342</td>
<td>0.0471</td>
<td>0.5927</td>
</tr>
<tr>
<td>Chase and Carrica [11] - Using CFD OWC</td>
<td>0.7498</td>
<td>0.2342</td>
<td>0.0458</td>
<td>0.6115</td>
</tr>
<tr>
<td>Chase and Carrica [11] - Using Experimental OWC</td>
<td>0.7659</td>
<td>0.2342</td>
<td>0.0435</td>
<td>0.6602</td>
</tr>
<tr>
<td>Ozden and Celik [22] - Self-propelled CFD</td>
<td>0.728</td>
<td>0.2416</td>
<td>0.0464</td>
<td>0.6033</td>
</tr>
<tr>
<td>Sezen et al. [23] - Self-propelled CFD</td>
<td>-</td>
<td>0.2363</td>
<td>0.4556</td>
<td>-</td>
</tr>
<tr>
<td>Present - Self-propelled CFD</td>
<td>0.7774</td>
<td>0.2312</td>
<td>0.0461</td>
<td>0.6202</td>
</tr>
<tr>
<td>Present - CEA</td>
<td>0.7731</td>
<td>0.2336</td>
<td>0.0473</td>
<td>0.6079</td>
</tr>
<tr>
<td>Present - Virtual disk</td>
<td>0.7271</td>
<td>0.2584</td>
<td>0.0550</td>
<td>0.5437</td>
</tr>
</tbody>
</table>

**Fig. 11.** Propeller rotation rates at self-propulsion point for different speeds of DARPA Suboff.
All results were found to be in accordance except the virtual disk approximation that predicted lower propeller efficiency $\eta_0$ at the self-propulsion point. It was also found out that the results generated by the CEA were promising. CEA predicted similar results with the other time-costly methods and is much more practical than the others.

The propeller rotation rates (rounds per second) obtained by each method are given in figure 11. It was found that all three methods generated close results; although, the discrepancy between results were slightly higher at higher hull velocities.

6. KRISO Container Ship (KCS) self-propulsion results

The self-propulsion points of KCS at its service speed ($Fr = 0.26$) was calculated using the CEA and direct CFD approach as explained in previous sections. The propeller used was the KP505 propeller and a scaled model of $\lambda = 1/31.599$ was solved in numerical simulations. The geometric properties of the KCS and the KP505 propeller are given in table 7 and table 8 respectively.

<table>
<thead>
<tr>
<th>Table 7. Geometric properties of the scaled KCS.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars $L_{pp}$ m</td>
</tr>
<tr>
<td>Beam at waterline $B$ m</td>
</tr>
<tr>
<td>Draft $T_m$ m</td>
</tr>
<tr>
<td>Wetted surface area w/o rudder $S$ m</td>
</tr>
<tr>
<td>Block coefficient $C_B$</td>
</tr>
<tr>
<td>Midship section coefficient $C_M$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 8. Geometric properties of the scaled KP505 propeller.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Blades -</td>
</tr>
<tr>
<td>Propeller Diameter m</td>
</tr>
<tr>
<td>Hub Diameter m</td>
</tr>
</tbody>
</table>

6.1 No propeller case

Calculations were first carried out to obtain the towed resistance of KCS without the propeller. Then, the self-propelled results were presented using the three mentioned methods. There are many results on the towed resistance values of KCS in the literature. Along with the results obtained in the present study, they are presented in table 9. Experiments published in Tokyo 2005 CFD Workshop were taken as reference from Carrica et al. and various numerical results are provided in table 9. Carrica et al. have provided a broad list in their study and in this work it was extended to cover some more results from the literature [24]. The second and third rows in the table are the numerical results carried out by Carrica et al. while the fourth (Hamburg Ship Model Basin), fifth (Potsdam Model Basin), sixth (Korean Maritime and Ocean Engineering Research Institute, now named as MOERI) and seventh (Osaka Prefecture University) rows are numerical simulations provided by various institutes or universities from all around the world. All results provided in table 9 were graphed in figure 12 to provide a better insight.
The total resistance coefficient obtained in the present study was slightly higher than the other results found in the literature. This was due to high pressure resistance predicted in our simulations. Our results were parallel with the numerical results of HSVA. Their results also suggest high pressure resistance which led to higher total resistance. The frictional resistance values of our study seemed close to other results found in the literature with only a 1.7% difference with experiments. All numerical results predicted high nominal wake values for KCS including our study. They were in between the range $0.72 < W_n < 0.75$ except the numerical results provided by OPU where they have calculated the nominal wake to be $W_n = 0.634$.

![Graphical comparison of resistance coefficients and nominal wakes in the literature that are provided in table 9.](image-url)
6.2 Estimation of self-propulsion point

The self-propulsion estimation for KCS was performed using the self-propelled CFD and the CEA methods. Virtual disk results were not presented for surface-piercing ships such as KCS due to very high predictions of thrust deduction factor and the hull efficiency. This was also verified by Dogrul et al. [30]. They have concluded their study by stating that the existence of the free water surface changes the thrust deduction factor dramatically and suggested using other methods for hull-propeller interactions. Due to the absence of virtual disk method in this section, \(w\) and \(t\) was calculated by the empirical relations suggested by the IMO [15]. The self-propelled CFD results were compared with the vast amount of data found in the literature. All results are listed in table 10.

| Table 10. Resistance coefficients and propulsion estimates for self-propelled KCS. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | \(C_T\*10^3\) | \(C_p\*10^3\) | \(C_r\*10^3\) | \(K_f\) | \(K_o\) | \(l-t\) | \(l-W_t\) | \(\eta_m\) | \(\eta_p\) | \(J\) | \(n\) | \(\eta_m\) | \(\eta\) |
| Experiments | 3.966 | 1.134 | 2.832 | 0.17 | 0.0288 | 0.853 | 0.792 | 0.682 | 1.011 | 0.728 | 9.5 | 1.077 | 0.74 |
| Carrica et al. [24] - DES | 4.011 | 1.172 | 2.847 | 0.1689 | 0.0296 | 0.8725 | 0.803 | 0.683 | 0.976 | 0.733 | 9.62 | 1.0866 | 0.724 |
| Carrica et al. [24] - HSV | 3.942 | 1.261 | 2.681 | 0.1702 | 0.03 | 0.865 | 0.789 | 0.667 | 0.981 | 0.725 | 9.56 | 1.0963 | 0.717 |
| Carrica et al. [24] - SVA | 3.878 | 1.024 | 2.854 | 0.163 | 0.0297 | 0.91 | 0.765 | 0.614 | 1.0065 | 0.705 | 9.5 | 1.1895 | 0.735 |
| Carrica et al. [24] - KRISO | 3.973 | 1.194 | 2.779 | 0.17 | 0.0228 | 0.857 | - | - | - | - | - | - | - |
| Carrica et al. [24] - OPU | 3.933 | 1.221 | 2.712 | 0.167 | 0.0282 | 0.8515 | 0.7888 | 0.631 | 1.074 | 0.7178 | 9.528 | 1.0795 | 0.7315 |
| Starke [28] | 3.96 | 0.99 | 2.97 | 0.176 | 0.0305 | - | - | - | - | - | 9.328 | - | - |
| Kim et al. [25] | - | - | - | 0.168 | 0.0288 | 0.843 | 0.802 | 0.664 | 1.014 | 0.726 | 9.7 | 1.0511 | 0.708 |
| Bugalski and Hoffman [2] | 3.804 | - | - | 0.1502 | 0.0263 | - | - | - | - | - | 9.8 | - | - |
| Gao et al. [3] | 3.964 | - | - | 0.165 | 0.029 | 0.852 | 0.772 | - | - | 0.714 | - | 1.1036 | 0.715 |
| Gaggero et al. [26] | 3.754 | - | - | 0.1694 | - | 0.8914 | 0.7471 | - | 1.021 | - | 9.2 | 1.1931 | - |
| Shen et al. [27] | 3.84 | - | - | 0.1682 | 0.029 | 0.8857 | 0.8721 | 0.6785 | 0.9811 | 0.7363 | 9.3231 | 1.0156 | 0.7429 |
| Gaggero et al. [6] | - | - | - | - | - | 0.856 | 0.769 | - | - | - | 9.6 | 1.1131 | - |
| Gaggero et al. [31] | - | - | - | - | - | 0.8688 | 0.7618 | - | - | - | 9.656 | 1.1405 | - |
| Present study - Self Propelled CFD | 3.983 | 1.284 | 2.699 | 0.167 | 0.028 | 0.891 | 0.7945 | 0.6818 | 1.0347 | 0.721 | 9.68 | 1.122 | 0.7915 |
| Present study - CEA | - | - | - | 0.1952 | 0.032 | 0.8165 | 0.7378 | 0.637 | - | 0.6613 | 9.8 | 1.1066 | - |

Table 10 covers a broad range of latest results for the self-propelled case of KCS at 1/31.599 model scale. Experimental results taken from Carrica et al. were again taken as reference for the numerical studies [24]. The frictional resistance coefficient of the experiment was calculated by the ITTC 1957 friction line given as \(C_F = 0.075/(\log Re - 2)^2\). The pressure resistance coefficient was not measured but calculated by \(C_P = C_T - C_F\). The table also covers two results carried out in the present study. The self-propelled CFD results were
produced with respect to the 1978 ITTC performance prediction method and the practical guidelines for ship self-propulsion CFD [16, 32]. For the CEA; $R_T$ was taken from numerical simulations of KCS without the propeller and $K_T$ was taken from experimental open-water propeller performance results. The other two inputs, $w$ and $t$, were taken from empirical relations recommended by IMO [15]. IMO recommends wake fractions for block coefficients of 0.5, 0.6, 0.7 and 0.8+, not providing any data for intermediate values. Therefore, a third order polynomial was fitted to the recommended values of wake fraction for one propeller ships and using the equation of this polynomial, $w$ for KCS was calculated using block coefficient value from table 7, $C_B = 0.6505$. IMO recommends using $t = 0.7w$ [15], therefore, thrust deduction factor $t$ of KCS was calculated using this equation.

A close observation of table 10 indicates that the self-propelled CFD results generated in this study are in accordance with the experiments and the other numerical results that may be found in the literature. On the other hand, the propulsive estimates generated by the CEA were found to be fair predictions. The results of CEA were not as good as the numerical studies implementing the finite volume method. However; considering the amount of time spent, CEA was found to be very practical and a good method to refer to at the initial design stages of a ship. The results presented in table 10 are visualized in figure 13.
Table 11. Propulsion factor estimates using different inputs for the CEA.

<table>
<thead>
<tr>
<th></th>
<th>$K_T$</th>
<th>$K_Q$</th>
<th>$1-t$</th>
<th>$1-W_T$</th>
<th>$\eta_0$</th>
<th>$J$</th>
<th>$n$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiments</td>
<td>0.17</td>
<td>0.0288</td>
<td>0.853</td>
<td>0.792</td>
<td>0.682</td>
<td>0.728</td>
<td>9.5</td>
</tr>
<tr>
<td>CEA with CFD + IMO</td>
<td>0.1867</td>
<td>0.0312</td>
<td>0.8165</td>
<td>0.7378</td>
<td>0.6464</td>
<td>0.6779</td>
<td>9.56</td>
</tr>
<tr>
<td>CEA with experiments</td>
<td>0.1701</td>
<td>0.0291</td>
<td>0.853*</td>
<td>0.792*</td>
<td>0.6621</td>
<td>0.7106</td>
<td>9.79</td>
</tr>
</tbody>
</table>

* Values taken from the experimental results published in (Carrica et al., 2010) [24].

The predictions of the CEA were better for the DARPA Suboff which was presented in the previous section. This is due to using empirical relations provided by the IMO which were not very accurate. IMO’s recommendation generates $1 - W_T = 0.7378$ and $1 - t = 0.8165$ and these inputs are not very good estimates compared to the experimental values of $1 - W_T = 0.792$ and $1 - t = 0.853$. Using the experimental values for the inputs $w$, $t$ and $R_T$, the estimation of the propulsion factors with the CEA become significantly enhanced as given in table 11 [15]. If CEA is supported by right interaction parameter values ($w$ and $t$) the results become very close to experimental results. The slight difference compared to experiments arise due to the difference in total resistance and the errors made during the interpolation in the thrust identity method.

7. Duisburg Test Case (DTC) self-propulsion results

The self-propulsion point of DTC at a Froude number of $Fr = 0.218$ was calculated in this section. A scaled model of $\lambda = 1/59.407$ was used in the calculations in accordance with the experimental results published in the reference study [33]. The geometric properties of DTC and its propeller are given in table 12 and table 13 respectively.

Table 12. Geometric properties of the scaled DTC.

<table>
<thead>
<tr>
<th></th>
<th>$L_{pp}$</th>
<th>m</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculas</td>
<td></td>
<td></td>
<td>5.976</td>
</tr>
<tr>
<td>Beam at waterline</td>
<td>$B$</td>
<td>m</td>
<td>0.859</td>
</tr>
<tr>
<td>Draft</td>
<td>$T_m$</td>
<td>m</td>
<td>0.244</td>
</tr>
<tr>
<td>Wetted surface area</td>
<td>$S$</td>
<td>m$^2$</td>
<td>6.243</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>$C_B$</td>
<td></td>
<td>0.661</td>
</tr>
<tr>
<td>Displacement</td>
<td>$\nabla$</td>
<td>m$^3$</td>
<td>0.827</td>
</tr>
<tr>
<td>Speed</td>
<td>$V$</td>
<td>m/s</td>
<td>1.668</td>
</tr>
</tbody>
</table>

Table 13. Geometric properties of the scaled propeller.

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades</td>
<td>-</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Propeller diameter</td>
<td>m</td>
<td>0.15</td>
<td></td>
</tr>
<tr>
<td>Hub diameter</td>
<td>m</td>
<td>0.0264</td>
<td></td>
</tr>
</tbody>
</table>

7.1 No propeller case

Experimental results can be found in the reference study for the resistance characteristics of the vessel without the propeller [33]. The CFD results for the no propeller case in comparison with the experiments are given in figure 14.
Although the CFD generated results in this study were lower, it can be said that the results were still in accordance with the experiments. It was found that the general trend of the two $C_T$ curves agreed well and the difference in the total resistance originated from the frictional resistance. This was possibly due to the vessel being forced to be held stationary in our numerical simulations while it was free to sink and trim in the experiments. Still, numerical simulations in this study suggested closer results to the experiments when compared with the results published by Kinaci and Gokce [34]. There are some other numerical studies mentioning the no propeller case of DTC and approaching the resistance problem by some other methods. The readers are referred to [35] to compare RANSE + VOF based numerical results of this study with only RANSE based results (double body approach) and [36] with RANSE + empirical approach based results.
The wake at the propeller disk at \( Fr = 0.218 \) is given in figure 15. Digital values of resistance and wake are summarized in table 14. These values were used in CEA to estimate the self-propulsion point of DTC at \( Fr = 0.218 \).

**Table 14.** Numerical results for no propeller case at \( Fr = 0.218 \).

| DTC - Model Scale: 1/59.407 |  |
|---|---|---|---|---|
| \( C_T \times 10^3 \) | \( C_P \times 10^3 \) | \( C_F \times 10^3 \) | \( W_n \) |
| 3.296 | 0.444 | 2.852 | 0.725 |

7.2 Estimation of self-propulsion point

Estimation of self-propulsion point of DTC was made at \( Fr = 0.218 \) using the CEA and the self-propelled CFD. The four parameters as inputs to the CEA were derived from:

- \( K_T \): value obtained from the open-water propeller results found in the reference study [33].
- \( R_T \): calculated using table 12 and table 14.
- \( w \): value calculated from the table provided by IMO [15].
- \( t \): calculated by using the equation \( t = 0.7 \times w \) provided by IMO [15].

Using these four inputs and calculating the self-propulsion estimates, table 15 was obtained for DTC. A close observation of this table points to the fact that the propulsion estimates calculated with both methods were in accordance with each other.

**Table 15.** Propulsion estimates for self-propelled DTC.

<table>
<thead>
<tr>
<th></th>
<th>( C_T \times 10^3 )</th>
<th>( C_P \times 10^3 )</th>
<th>( I-T )</th>
<th>( L-W_T )</th>
<th>( \eta_H )</th>
<th>( \eta_R )</th>
<th>( J )</th>
<th>( n )</th>
<th>( \eta_U )</th>
<th>( \eta )</th>
</tr>
</thead>
<tbody>
<tr>
<td>el Moctar et al. [33]</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.91</td>
<td>0.725</td>
<td>0.592</td>
<td>0.993</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Present study Self propelled CFD</td>
<td>3.608</td>
<td>0.788</td>
<td>2.820</td>
<td>0.195</td>
<td>0.035</td>
<td>0.831</td>
<td>0.762</td>
<td>0.589</td>
<td>0.914</td>
<td>0.657</td>
</tr>
<tr>
<td>Present study CEA</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.205</td>
<td>0.033</td>
<td>0.812</td>
<td>0.732</td>
<td>0.633</td>
<td>-</td>
</tr>
</tbody>
</table>

There are some experimental data in el Moctar et al. which are also presented in table 15 and our results generated by two different methods in comparison with experiments were found to be satisfactory [33]. The biggest discrepancy was in thrust deduction factor \( t \) which was also reflected on the hull efficiency \( \eta_H \). DTC is a relatively new benchmark ship and due to this reason, studies focusing on this ship are limited in the literature. There is a recent study on DTC which also covers self-propulsion estimation by Sigmund and el Moctar but it was made for a different ship scale \( (\lambda = 1/63.65) \) and a lower Froude number \( (Fr = 0.087) \) [37].

8. Conclusions

In this study, estimation of self-propulsion point was presented for three ships in comparison with other results found in the literature. The self-propelled CFD approach implemented in this study was found to be in accordance with the experiments and other numerical studies. Using a virtual disk to represent the propeller is also a method to estimate the self-propulsion point and generated close results for the DARPA Suboff when compared with the results of other researchers in the field. However, virtual disk numerical model can be applied on self-propulsion simulations which only requires the open-water propeller
performance as an input. Results generated by the virtual disk approach were not satisfactory in numerical simulations where free water surface was present. Although satisfactory accordance was found for the DARPA Suboff, the results of self-propulsion for KCS and DTC were not compatible.

It was one of the main goals of this study to show the robustness of the classical engineering approach on predicting the self-propulsion points of marine vehicles. Using some empirical relations and open-water propeller results, the basic approach returns quite compatible results with experiments and numerical simulations. Considering the practicality of this engineering approach, it is believed that it can be used at least during the pre-design stages of a ship.

It is believed that it would be interesting to challenge the classical engineering approach with planing hulls or hulls with multiple propellers. In this respect, future studies are expected to be made for different types of ships such as planing hulls, catamarans and twin-propeller ships.

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LEAN TOOLS APPLIED TO A SHIPBUILDING PANEL LINE ASSEMBLING PROCESS

UDC 629.5.081  
Professional paper

Summary

To improve the shipbuilding process and obtain a more cost-effective production, several approaches are used today. The state of the art approach is based on newer production technologies, mainly steel cutting and welding technologies, which allows to improve the quality of the products and reduce the production time. Other approach, not so applied, is based on the re-organization some shipbuilding stages, applying some production improvement tools, like the Lean approach. The panel assembly line stage of the shipbuilding process is a key phase of the production sequence due to its relation with the remaining shipbuilding stages, essentially framed in the ship’s block construction sequence, and its characteristics make it an accessible shipbuilding stage to apply the Lean approach tools. The Lean tools can be applied with a wide range of minuteness and on variable settings. The present paper analyses a rather simple modification on the panel line assembling process sequence, achieved through the application of some key concepts of the Lean approach. For the verification of the times and costs benefits due to the application of the proposed modification, a graphic interface tool was developed, allowing to analyse the costs and times results of the panel line production process.

Key words: shipbuilding; panel line; lean

1. Introduction

The production competition imposed on the European shipbuilding industries, due to better developed production processes of the Far-East shipbuilding companies, force new solutions to be studied and applied in the European shipyards. The state of the art solution to reach a cost-effective ship production tends to emphasize new technical solutions, mainly new and better cutting and welding technologies, as quoted by Gordo et al. [1]. However, a parallel approach to improve the production process concerns the use of process improvement methodologies, as the Lean methodology. The Lean methodology application covers a wide range of minuteness and can be applied on variable settings and production stages. The present paper concerns a rather simple analysis and application of some Lean methodology key concepts on one of the most important production stage of the shipbuilding process: the panel assembly line sequence. The panel assembly line is one of the most straightforward stages, framed in the complex shipbuilding process, to apply such production improvement tools, due to its simplicity and well-established sequence of actions and workstations.
2. Background

The Lean methodology was developed in Japan after 2nd World War, and was also known as Toyota Production System (TPS). As stated by Taichii Ohno [2], one of the key TPS’s goals is to identify and eliminate the seven major wastes typically found in mass production: overproduction, waiting, motion, transport, over processing, inventory and defects. During the 90’s the Lean principles were imported to the shipbuilding studies and applied in the industry. The importance and benefits of the application of Lean methodology in the shipbuilding industry was stressed by Liker & Lamb [3] and Storch & Lim [4]. Kolich et al. [5, 6] developed a significant set of studies on the subject of application of Lean methodologies in the shipbuilding process, some of them inspecting with more detail such methodology application on the panel assembly line.

Motivated by these works, the present paper aims to deepen the demonstration of economic advantages due to the application of Lean methodology on the panel assembly line, by developing a graphic interface tool to compute the times and costs of the panel assembly line manufacturing process before and after the application of rather simple modifications in the production sequence.

Regarding shipbuilding times and costs analysis models and the panel assembly line manufacturing, which had significant importance for the present paper, some investigations were recently carried out, respectively, by Leal & Gordo [7] and by Oliveira & Gordo [8].

3. Panel assembly line analysis tool

The understanding of the times and costs effects of modifications on the panel assembly line manufacturing stages, a graphical interface tool (PALIAN, Panel Assembly Line ANalysis) [9] was created, as shown in Figure 1.

PALIAN allows the user to specify the times and costs settings for each main activity as presented in Figure 2, of the four panel assembly line stages considered: Plate cutting; butt-welding; stiffeners assembling; and stiffeners welding. The user is also free to detail the set of flat panels which are to be considered during the computations.

PALIAN generates an output file with the times and costs values of the classical approach and Lean improved proposed approach allowing for comparison.

4. LEAN methodology on the panel assembly line

The improvement of the manufacturing process can only be done by the understanding and interpretation of the original sequential flow of the production process, and then apply manufacturing improvement methodologies in the production sequence.

4.1 Original panel assembly line manufacturing process

The original panel assembly line manufacturing process is based on Oliveira & Gordo [8], where the production sequence is quite well demonstrated. Figure 3 illustrates the original panel assembly line sequence flow production.

The 1st workstation consists on the plate blanket fabrication, through the one side submerged arc welding technology of the panel’s plates, hence not needing to turn over the plates. This workstation can be divided in 5 main stages: align and tack welding of the plates; SAW (Submerged Arc Welding) preparation; SAW of the butt weld; weld quality control; possible re-welding of some weld fragment. That butt welding process’ stages are repeated for as many plate joins that the panel presents.
The 2nd workstation is responsible for the automatic oxy-fuel cutting and marking of the previously assembled plate blanket. The workstation can be summarized in the following stages: automatic cutting preparation; automatic marking; automatic cutting; manual cutting; dimensional control.

The 3rd workstation consists of the manual assembling of the stiffeners on the plate blanket, and can be summarized in the following stages: Distribution of the stiffeners, tag-welding of the stiffener and angular control.
The 4th workstation due the semi-automated GMAW-MAG (Gas Metal Arc Welding – Metal Active Gas) welding of the stiffeners, on both sides. In this workstation the welding process can be divided in the following phases: preparation of the welding equipment; preparation of the first side weld; welding of the first side of the stiffener; preparation of the second side weld, i.e., deburring of the tag-welds realized in the third workstation; welding of the second side of the stiffeners; quality control of the weld; possible re-welding of some fragments of the fillet weld.

4.2 Flat panels to simulate manufacturing

PALIAN allows to undertake the manufacturing process simulation of the panel assembly line by computing the times and costs, using a set of values defined by the user in the software’s graphical interface. The flat panels used in the present paper were part of a new ship built in the Portuguese “WESTSEA Shipyard”, and the correspondent panel line construction stages times were obtained by the direct monitorization of the manufacturing process by the authors. The four flat panels considered present the following characteristics:

The first flat panel, with a 7.7 m width and 7.0 m length, illustrated in Figure 4, comprehends four plates of 7 mm thickness, totalizing a cutting length of 34.6 m, and ten stiffeners with thickness of 8 mm and 10 mm, totalizing 63.0 m of stiffeners to assemble and weld.

The second flat panel, with 11.1m width and 8.6 m length, presents five plates of 5 mm thickness, totalizing 49.8 m of cutting length, and twenty-three stiffeners of 6 mm thickness, totalizing 129.1 m of stiffeners to assemble and weld, as shown in Figure 5. The third flat panel, with 12.5 m width and 10.0 m length, is made up also with five plates of 5 mm, totalizing 43.8 m of cutting length, and 23 stiffeners of 6 mm, totalizing 141.3 m, as illustrated in Figure 6.

The fourth flat panel considered, with 12.6 m width and 11.5 m length, presents five plates of 5 mm thickness, totalizing 62.3 m of cutting length, and forty-one stiffeners totaling 237.9 m, Figure 7.
5. Original sequence times results

PALIAN generated the flow sequence time chart of the manufacturing process, as in Figure 8. Each colour of the time chart is representative of a different flat panel, the first row illustrates the time for the butt welding, the second row illustrates the cutting of the plate blanket, the third one represents the stiffeners assembly, and the last row shows the times for the stiffeners welding.

Through the interpretation of the time values obtained, it is possible to generate a hand-made value stream map of the original panel assembly line sequence, as shown in Figure 9. The value stream mapping analysis of the production process is a key tool of the Lean methodology and Figure 10 displays the representative meanings of the symbols and notations of the value stream map.
As demonstrated in the value stream map for the original sequence production flow, we obtain the following ratio of the Lead Time (LT), which translates the time between the beginning of the activity and its completion, over the TAV (Time of Added Value):

$$\frac{LT_1}{TAV_1} = \frac{35.25}{20.13} = 1.75$$

In order to explain the way some of the values of Figure 9 were computed, it is possible to exemplify that, for the 1st workstation, the Cycle Time (T/C) was obtained by dividing the total time, from the moment of the initiation of the first flat panel until the moment of completion of the last flat panel (15.96 h), by the number of flat panels (4 panels); the Changeover Time (T/R) was computed by dividing the activities that cannot be categorized as added time values (9.04 h), not including waiting times, by the number of flat panels; and the Time of Added Value Activities was obtained by dividing the activities that can be categorized as source of added value (6.92 h) by the number of flat panels.

It is key to stress that considerations of what should be assumed as “added value activities” in the panel assembly line were greatly loose assumptions. Otherwise, the ratio would be expected to be much higher if one would be more precise in such “added value activities” concept. The main goal of the Lean methodology application is to decrease as far as possible the ratio, by reducing the non-value-added activities and eliminating queues in the flow production.
6. Change of sequence of the panel assembly line

By analysing the time chart and respective value stream map of the original sequence manufacturing flow is evident some possible adverse effect of the queue instances on all the stages. Hence, it will be studied the implementation of different manufacturing sequence flow of the panel assembly line in way to reduce or eliminate the flow queues.

6.1 First suggested sequence modification

The first alternative on the panel assembly line sequence to be study aims to test the reliability of the “one-piece flow” concept on the several plates that form the panel, as suggested by Liker & Lamb [3] and Kolich et al. [6], i.e., instead of assuming the panel as an individual product, consider it as a sum of multiple single products. Hence, the panel assembly line sequence would suffer a profound modification on its nature, thus displaying the following arrangement: Cutting of the correct perimeter of the steel plate, assembling of the stiffeners on the steel plate, welding of the stiffeners, and, finally, butt welding of the individual sets “plate + stiffeners”, as shown in Figure 11.

<table>
<thead>
<tr>
<th>Modified panel's line sequence</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cut the perimeter on each</td>
</tr>
<tr>
<td>panel's plate, individually</td>
</tr>
<tr>
<td>Assembling the stiffeners</td>
</tr>
<tr>
<td>of each plate</td>
</tr>
<tr>
<td>Welding the stiffeners</td>
</tr>
<tr>
<td>of each plate</td>
</tr>
<tr>
<td>Butt welding of the plates</td>
</tr>
</tbody>
</table>

Figure 11 First alternative on the panel assembly line sequence

PALIAN also includes the consideration of such panel assembly line sequence, and the respective obtained time chart of such manufacturing approach is illustrated in Figure 12. To correctly interpret the time chart is key to stress that each row and colour is associated with one given workstation: the first row, in red, represents the times of the plates cutting; the second row, in blue, represents the times of the stiffeners assembling on the plate; the third one, in green, illustrates the welding of the stiffeners; the last row, in yellow, exhibit the times of the butt welding of the panel’s plates. Each colour’s tint illustrates different panels, i.e., for example, in the third row, representing the stiffeners welding stage, the first four columns, with similar green tint, illustrates the four plates/sets of the first panel, and the 5th, 6th, 7th, 8th and 9th columns, with a brighter tint of green illustrates the five plates/sets of the second panel.

From the resultant hand-made value stream map illustrated in Figure 13, we get the following ratio:

\[
\frac{LT_2}{TAV_2} = \frac{48.16}{20.13} = 2.39
\]

Although we have eliminated the queue before the 2nd and 3rd workstations, the conditional constrains of the 4th workstation, where to butt weld the sets they need to be fully manufactured, generates vast gaps between the successive butt welds, thus increasing significantly the cycle time of the butt weld workstation, and thereafter increasing the lead time (LT) of the complete process.
A fast and less correct analysis of the time chart obtained could lead us to consider that this alternative panel assembly line manufacturing sequence is better due to the 8% decrease of the total time needed to complete the four panels (from 81.4 h to 74.9 h). The 37% increase of the ratio testified in eq. (2) strongly suggests that such solution would not be beneficial to the panel assembly line manufacturing process.

Figure 12 First sequence modification time chart

Figure 13 First sequence modification value stream map

6.2 Second suggested sequence modification

Analysis of Figure 13 suggests to join workstations 3 and 4. State of the art of actual equipments allows this merge between the butt welding of the profiles and its immediate fillet welding, as referred, for example, by Kolich et al. [5]. PALIAN supports such panel assembly line sequence possibility. The resultant time chart is presented in Figure 14, and, through its values, one can draw and compute the values of the respective value stream map, as shown in Figure 15. By merging the 3rd and 4th workstations, the queue times are eliminated, Figure 14, and the value stream map become:
The ratio obtained is slightly smaller than the one of the original panel assembly line situation presented in formula (2), however the time needed to accomplish the complete manufacturing of all the four flat panels is 10% larger (89.8 h) than the time needed on the original panel assembly line, thus is interesting to realize a costs analysis comparison. As it was stated earlier, the developed program also computes the cost analysis of each panel assembly line workstation, which times are presented in Table 1 and 2.

The modified sequence situation presents no inactive labour cost, i.e., there is no flow queue in any workstation and thus no workers waiting for the product from the previous workstation. Using the concept of “plate + stiffeners” as main product instead of the panel as such, the changeover activities of the steel cutting station will be repeated more times.
Instead of performing the cutting preparation, dimensional control and post cutting work activities only for the 4 panels, they will be repeated for the 19 plates of the four flat panels which generates an increase of the labour cost. Such supplementary activities generate a significant increase of the man-hours needed, thus increasing the labour cost on the steel cutting workstation. The savings of the modified sequence are, although positive, negligible, Table 1 and Table 2. The time needed to manufacturing of the four panels is 10% greater. One can conclude that the costs savings do not support the greater lead time of the manufacturing process.

<table>
<thead>
<tr>
<th>Workstations</th>
<th>Consumables costs</th>
<th>Active labor costs</th>
<th>Inactive labor costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Butt welding</td>
<td>31.6</td>
<td>66.3</td>
<td>-</td>
</tr>
<tr>
<td>Plate blanket cutting</td>
<td>14.7</td>
<td>50.7</td>
<td>30.0</td>
</tr>
<tr>
<td>Stiffeners assembling</td>
<td>86.6</td>
<td>340.6</td>
<td>14.7</td>
</tr>
<tr>
<td>Stiffeners Welding</td>
<td>234.5</td>
<td>431.2</td>
<td>7.1</td>
</tr>
</tbody>
</table>

Table 2 Second sequence modification cost analysis [€]

<table>
<thead>
<tr>
<th>Workstations</th>
<th>Consumables costs</th>
<th>Active labor costs</th>
<th>Inactive labor costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate cutting</td>
<td>14.7</td>
<td>84.4</td>
<td>-</td>
</tr>
<tr>
<td>Stiffeners assembling</td>
<td>86.6</td>
<td>340.6</td>
<td>-</td>
</tr>
<tr>
<td>Stiffeners welding</td>
<td>234.5</td>
<td>431.2</td>
<td>-</td>
</tr>
<tr>
<td>Butt welds</td>
<td>31.6</td>
<td>66.3</td>
<td>-</td>
</tr>
</tbody>
</table>

7. Overall Equipment Efficiency concept


Some of those wastes can be identified in the equipment-related activities, thus allow to list a set of other particular wastes related to the Overall Equipment Efficiency (OEE) as shown in Table 3 [10]. From the wide set of input data which the user should define in PALIAN, Figure 16, one can identify the OEE wastes listed in Table 4.

Considering the original panel assembly line process as the start reference point, by decreasing or eliminating the time values of some OEE wastes in an acceptable way (decrease, by half, the equipment setup time activities and considering that no re-welds are needed) Table 4 sums up the obtained values testifying the cost advantages of the implementation of production politics that lead to a decrease of the OEE wastes, such like the setup times or the re-works. The case study indicates a 10% reduction of the production cost and, simultaneously, the time needed to complete the four panels manufacturing process decrease by 19%, from 81.4 h to 66.2 h.
Lean Tools Applied to a Shipbuilding Panel Line Assembling Process

A. Oliveira, J. M. Gordo

Figure 16 OEE analysis wastes [10]

Table 3 OEE wastes specified in the input interfaces

Butt welding workstation
- Welding preparation
- Tag-welding preparation
- Re-welding

Plate blanket cutting workstation
- Automatic cutting equipment setup

Stiffeners welding workstation
- First side welding preparation
- Second side welding preparation
- Welding equipment preparation
- Re-welding

Table 4- OEE wastes improvement costs analysis

<table>
<thead>
<tr>
<th>Workstations</th>
<th>Consumables costs</th>
<th>Active labor costs</th>
<th>Inactive labor costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Butt welding</td>
<td>28.7</td>
<td>52.4</td>
<td>-</td>
</tr>
<tr>
<td>Plate blanket cutting</td>
<td>13.7</td>
<td>48.2</td>
<td>18.1</td>
</tr>
<tr>
<td>Stiffeners assembling</td>
<td>86.6</td>
<td>340.6</td>
<td>5.3</td>
</tr>
<tr>
<td>Stiffeners Welding</td>
<td>227.3</td>
<td>341.5</td>
<td>10.4</td>
</tr>
</tbody>
</table>

8. CONCLUSIONS

The results obtained and analysed allowed to testify that the appropriated product to consider in the “one-piece flow” methodology is the flat panel, instead of the “plate + stiffeners” sets that compose the flat panel. For such analysis, the value stream mapping of the manufacturing process was a key element to interpret the production efficiency, proving the reliability of Lean methodology analysis tool.
By considering the “7 deadly wastes” described by the Lean methodology, and applying them to the OEE, thus suggesting the equipment-related activities which times should be decreased or eliminated, allowed to obtain significant cost and time advantages on the panel assembly line manufacturing process. Such improvements could be obtained by installing newer equipment and higher quality welding technologies.

9. Acknowledgements

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REFERENCES

APPLICATION AND ANALYSIS OF SOLID OXIDE FUEL CELLS IN SHIP ENERGY SYSTEMS

UDC 629.5.038:629.545
Original scientific paper

Summary

This study identifies and analyses the overall energy consumption required for converting LNG (liquefied natural gas) from liquid to gaseous state in high pressure vaporizers at various cargo discharge rates on a liquid natural gas regasification vessel (LNGRV). The actual measured data were collected on an available vessel, to calculate the overall energy consumption of converting natural gas from liquid to gaseous state by three built-in turbo generators. Next, the study considers their replacement with the newest innovative technology including three solid oxide fuel cell (SOFC) power plants of the same power. This paper provides a simplified analysis of the proposed implementation of the SOFC power plant which was performed for the first time in the existing literature. The results show a significant increase in the achievable electrical efficiency of 40.6%, with respect to 32.9% of a system with turbo generators. The research has also shown that waste heat from the SOFCs can be used to produce thermal energy, resulting in further savings of 2.6% in natural gas consumption. Future research could be done on other regasification terminals in the world, which use different main propulsion technology, using an open, closed or combined cycle during the regasification operations.

Key words: LNG (liquid natural gas); LNGRV (liquid natural gas regasification vessel); Regasification unit; High pressure manifold; SOFC (solid oxide fuel cell)

1. Introduction

A total of 452 LNG tankers are currently working around the world. The difference is based on the propulsion of the LNG tankers. According to the latest data from [1, 2], 262 LNG tankers can be found with a steam turbine propulsion, 112 LNG tankers with a dual fuel diesel electric propulsion, 50 LNG tankers with a diesel engine propulsion and a re-liquefaction plant, 25 tankers with a three fuel diesel electric propulsion, 3 LNG tankers with a marine engine gas injection propulsion and 1 LNG icebreaking class tanker with a diesel engine propulsion. 74 new LNG tankers were ordered with different propulsions till 01.04.2020.

The advent of LNG as a marine fuel is triggered by the efforts of policy makers to abate environmental burdens from the exhaust gases of ships. A thorough outline of the regulatory
environment triggering the use of alternative fuels in ships and, therefore, maritime networks, is presented in literature [3]. It seems evident that air emission regulations will continue to demand that ships reduce emission and increase operational efficiency. Hence, policy and technology evolution and market initiatives are expected, which is a promising set of conditions for the market acceptance of LNG-fuelled ships [3]. The growing demand for energy sources in the global framework, particularly for LNG, has led to the development and construction of Liquefied Natural Gas Regasification Vessels (LNGRVs). This type of an LNG carrier is specific for its compatibility with all the existing terminals (loading and discharging) in the world. Furthermore, along with its conventional purpose of loading, transporting, and discharging LNG (in liquid state), it is fitted with a built-in regasification unit at the bow area, i.e. a unit which converts LNG from liquid to gaseous state, hence distributing compressed gas ashore through a high pressure terminal pipeline (connecting to it through a high pressure manifold on the carrier deck). So far, eight carriers using this technology have been built in the world. They are named LNGRVs, and are mostly owned by the US company Excelerate Energy. Each of the eight LNGRV carriers has its own propulsion plant (steam turbine) which enables them to sail, but they can also be used as ships for cargo storage and regasification [4]. Figure 1 shows a LNGRV tanker connected to the discharging terminal [5].

The paper describes the functioning of a regasification system, particularly the regasification system used on an LNGRV, and the means of discharging compressed gas through an open loop high pressure manifold, in which relatively warm seawater is used as the heat source. The seawater passes through a high pressure vaporizer, causing the vaporization of LNG.

LNGRVs are capable of providing a continuous compressed gas discharge amount to the consumers, ranging from 50 to 800 MMSCFD (Million Standard Cubic Feet per Day) at atmospheric pressure and the temperature of 15.56 °C. One million standard cubic feet per day equals 1,116.28 m³/h (the volume, in m³, is given at 0 °C and 100 kPa). For the purpose of
calculating a natural gas volume, necessary for calculating and determining the gas energy value, standard conditions for temperature and pressure are used. Accordingly, it follows that the compressed gas discharge flow at LNGRVs ranges from 1.34 million m$^3$/d to 21.43 million m$^3$/d.

The technology of LNG vaporization on LNGRVs implies the usage of water (sea or fresh water) in high pressure vaporizers aiming to change the aggregate state of LNG. A schematic diagram of the regasification process is shown in Fig. 2. The main component parts of the regasification unit system on an LNGRV are: 3 feed cargo pumps, a suction drum, 2 small high pressure pumps, 6 high pressure pumps, 6 high pressure vaporizers, a metering unit, 2 control valves, and a high pressure pipeline. All these components are used for the LNG regasification. In addition, the auxiliary system comprises 3 steam heaters, 3 ballast pumps, and 3 circulating pumps. LNG is stored in the cargo tanks at a pressure slightly above atmospheric pressure and is pumped by small high pressure feed pumps to the suction drum. From the suction drum the liquid pressure is increased through pumping by the high pressure pumps to the vaporizers. The regasification is achieved by passing the liquid through the water heated shell and tube vaporizers, the gas then being sent to the distribution network onshore through a high pressure manifold. The high pressure pumps increase the pressure of the LNG from the suction drum to the vaporizers, to a pressure in excess of 100 bars. The gas is then metered and passed via a back pressure control valve, through an emergency shutdown valve and is in turn directed to the high pressure manifold discharge connection. The purpose of the auxiliary heating water system is to supply heating water to the vaporizers and, if required, to heat up the heating water up to the required temperature in case of operation in a closed or combined mode. An LNG vaporizer, a shell and tube type, has been designed to cope with the high thermal shrinkage due to the cold LNG and corrosion against sea water. The piping system connected to the inlet and outlet of the vaporizer has been designed to have sufficient flexibilities so that excessive nozzle forces are not exerted to the vaporizer. The vaporizer has been designed to operate in the supercritical pressure of the natural gas. In general, the heating water outlet from the LNG sides is at a lower temperature than the natural gas side [4].

![Fig. 2. Schematic presentation of regasification process [4]](image)

The ship’s main boilers produce the superheated steam necessary for driving 3 turbo generators that have to ensure enough electrical energy for running the ship machinery and for driving the engines and devices that partake in the regasification process. The ship’s main
boilers (steam generators) use exclusively natural gas as fuel, which is brought to the boilers by a low pressure compressor through the steam heater located in the ship’s compressor room.

Liquefied natural gas (LNG) fuelled shipping systems adopt the boil-off gas (BOG) reliquefaction process to maintain the pressure of the storage tank and to minimize methane loss [6].

Fuel cells may provide a suitable solution to reduce the environmental impact of the ship’s operations, since they are fuel efficient emitting few hazardous compounds. The expanding infrastructure of liquefied natural gas and development state of natural gas-fuelled fuel cell systems can facilitate the introduction of fuel cells on ships. Fuel cell combined cycles, hybridisation with auxiliary electricity storage systems and redundancy improvements are identified as topics for further studies [7].

The aim of this paper is to analyse energy efficiency when replacing the existing turbo generators by the Solid Oxide Fuel Cells (SOFCs). Instead of three turbo generators (each of the power of 3,700 kW), three SOFCs (each of the power of 3,700 kW) would be installed in the steam plant for producing electrical energy.

Yousri et al., analysed a combined system consisting of a solid oxide fuel cell (SOFC) and a steam turbine system which utilizes the energy transported with the exhaust gas leaving the fuel cell operated in ship power plants [8]. It is found that a high overall efficiency approaching 60% may be achieved with an optimum configuration using SOFC system. In another study [9], steam and SOFC based reforming options of natural gas for PEM fuel cells are proposed as an attractive option to limit the environmental impact of the marine sector. It is found that a high overall efficiency approaching 60% may be achieved using SOFC based reforming systems which are significantly better than a reformed PEM system or an SOFC only system.

A conceptual design of the SOFC for a 250 kW CHP (Combine heat and power) application [10], has spurred the group of authors to develop a hybrid system with diesel engines and two SOFCs to produce electricity on supplier vessels [11].

Energy analysis of the SOFC, fuelled by methane [12,13], then the distribution of electric power generated by SOFCs [14], and an analysis of energy and exergy reformation of methane using steam [15-17], provide the first concept for the use of SOFCs for production of electricity on an LNG tanker with steam propulsion. On an LNG tanker transporting natural gas, a main turbine uses superheated steam to run. These facts are advantages and therefore have a positive reflection on the research of combined steam propulsion systems with SOFCs. The steam produced from the ship’s boilers is used for natural gas reformation to produce hydrogen for the SOFC.

A thermodynamic analysis of a Brayton cycle and a Rankine cycle with the fuel cell experimental data [18,19] and exergetic analysis applied to a combined cycle power plant has an inclination [20,21] for saving energy if it is electricity generated by the SOFC. According to [22-24], the integration of a hybrid SOFC with gas turbines was investigated, where an analysis of energy, exergy and exergo-economic of cogeneration of heat was completed and a similar one could implement a combination of the SOFC with the steam turbine plant.

This research did not include a combined and a closed cycle for both commercial reasons and the impossibility to use steam heaters during the regasification operation due to the continuous high temperature of the sea water in the location where the measurements were taken.

Diesel-electric propulsion systems with heat utilization can achieve even higher efficiency but our goal is to improve efficiency on the steam turbine propulsion ships. The benefits of the proposed power system configuration is achieved on an LNG tanker transporting natural gas with combined steam propulsion systems that include SOFCs.
2. Description of an open loop regasification system

A similar issue is published in [25] where aim of the study has been to determine the energy consumption required to liquefied natural gas from liquid into a gas at high pressure evaporators at different instalments of discharging cargo. This research was conducted for the gas terminal Mina Al Ahmadi Gas Port in Kuwait, and the results were monitored during the discharge in May 2011. The paper describes the functioning of the regasification system. The basic principle of the system for gasification of tankers to transport liquefied natural gas has been described too. The results of this study can be used to compare the energy consumption in the conversion of gas from a liquid to a gaseous state to the other terminals that use the services of ships for transporting liquefied natural gas plant with integrated gasification, and to analyse the energy consumption of conventional unloading of liquefied natural gas terminal. During this research, the consumed energy obtained data were only measured in the open cycle because the system was limited by sea temperature of around 25 °C during May 2011.

This paper deals with new SOFC technology that can be used instead of a turbo generator to produce the electricity needed for the regasification process on LNGRV carrier. It should be noted that the analysis was performed with the data collected from the existing LNGRV carrier. Also, another analysis should be carried out in the future with different LNG carriers using new propulsion technologies.

The functioning of a regasification system was monitored on the LNGRV Exquisite carrier during the discharge period from early January to late April 2014 at the Guanabara Bay LNG Import Terminal just outside of Rio de Janeiro in Brazil. Throughout the period in question, the average seawater temperature ranged from 22 °C to 26 °C. Such seawater temperature enabled the regasification process to be carried out in an open loop, i.e. the cargo in the high pressure vaporizers was heated by seawater, without using the steam heaters. The placement of the pumps and pipelines on the carrier allowed for the achievement of the desired compressed gas output pressure, which was dependant on the current demands of the consumers ashore. The regasification and the compressed gas discharge process were conducted without suspension while the liquefied natural gas from another LNG carrier was being loaded to the LNGRV Exquisite carrier.

For the LNGRV Exquisite’s regasification process and cargo discharge, commonly 2 seawater ballast pumps, 2 seawater circulating pumps, 2 feed cargo pumps, 5 high pressure cargo pumps and 5 high pressure vaporizers were used.

The discharge flow ranged from 111,628 m³/h to 893,024 m³/h. The seawater was passing through 6 high pressure vaporizers with the minimal flow of 1,800 m³/h in order to avoid the high pressure vaporizers’ freezing, i.e. the vaporizers were ready for use at any instance. During the maximum discharge flow of 893,024 m³/h, it was necessary to turn on 3 seawater ballast pumps, 3 seawater circulating pumps, 3 feed cargo pumps, 6 high pressure cargo pumps, 6 high pressure vaporizers, and the diesel motor (that uses natural gas for fuel), which significantly increased the natural gas consumption [4].

In order for a regasification unit system to start working, the first task would be to cool down the pipelines and other units that partake in the process. LNG is supplied to the pipeline leading from the cargo tank to the suction drum, and inside the suction drum itself, through a
small spray nozzle installed in the cargo tank. The part of the unit leading from the suction drum to the vaporizer has to be cooled down and brought to a working pressure by using a small high pressure pump with a design capacity of 20 m$^3$/h and discharge pressure of 237 bars.

A feed pump pressures the LNG from the cargo tank towards the suction drum. Each feed pump has a design capacity of 620 m$^3$/h, and discharge pressure of 15.5 bars. The suction drum volume is 17 m$^3$. At normal system activity, the suction drum working pressure is 0.35 MPa. The LNG takes about 55% of the suction drum total capacity. By free falling, the LNG reaches the suction side of a high pressure pump with a design capacity of 205 m$^3$/h, and discharge pressure of 237 bars. The high pressure pump brings the LNG to the pressure of 10 MPa and pressures it towards the high pressure vaporizer, where the aggregate state of the LNG changes from liquid to gaseous.

Water temperature is a very important factor in the process of cargo regasification in high pressure vaporizers. The water for vaporizing LNG circulates through a high pressure vaporizer in an open loop, without being heated with steam heaters. With such open loop systems, the seawater is fed from the pumps (with a design capacity of 5,000 m$^3$/h) located in the ship’s engine room, and transported towards the circulating pumps (with a design capacity of 5,000 m$^3$/h) that force the water towards the high pressure vaporizers, and finally to an overboard discharge. Throughout the process, the water temperature decreases by about 7 °C. The input water temperature in a high pressure vaporizer must not be lower than 5 °C, with a minimum water flow of 1,800 m$^3$/h for each high pressure vaporizer.

Throughout the process, the water temperature lowers by about 7 °C. The input water temperature in a high pressure vaporizer must not be lower than 14.7 °C, with a minimum water flow of 1,800 m$^3$/h for each high pressure vaporizer. In an open loop the minimum heating water flow to each vaporizer is designed to avoid any possibility of the loss of the heating water flow and the resulting greater risk of icing up a vaporizer.

The compressed gas travels from the vaporizers towards the metering unit, which measures the compressed gas flow, temperature, pressure, and composition. Following the metering unit, there is a control valve built in the high pressure pipeline, which regulates the compressed gas output pressure to somewhere between 7 and 10 MPa. The compressed gas travels towards a high pressure terminal ashore. The control over the entire system is based on the following 3 parameters: discharge rate (demanded cargo quantity), maximum pressure, and minimum temperature of the compressed gas to be discharged [4].

3. Natural gas consumption during regasification and discharging process

During this research on energy consumption, the obtained data were exclusively measured at the open loop, as the system was limited by the seawater temperature of 22 °C to 26 °C as stated above. All the data measured were taken from sophisticated, calibrated and certified devices installed on this type of carrier. The Fuel Gas Flow Turbine Meter devices were used for measuring the compressed gas discharge flow and the consumption of natural gas in the main boilers, and the obtained data were sent every 15 seconds to the data collection unit, Flow Counter M3000. This device shows the data on the screen immediately, and prints them out hourly every day. The data printed by the device were used in this research. The data gathered (15-samples) did not differ significantly from one another, so the arithmetic mean was used for calculating the average value. It has all the necessary properties which characterize measures of central tendency, as well as additional properties significant for its application.

The graph in Figure 3 shows the need to produce electrical energy and the overall gas consumption of the steam-turbine plant in dependency on the value of natural gas discharge flow. The need to produce electrical energy proportionally increases to the rate of 669,768 m$^3$/h
and starts rapidly growing when it exceeds this value. The reason behind this is the turning on of another seawater ballast pump, seawater circulatory pump, feed cargo pump, high pressure cargo pump, and the diesel motor. Accordingly, the increased production of electrical energy in the ship’s engine room leads to an increased consumption of natural gas which is combusted in the ship’s main boilers, but also, at the discharge rate of 669,768 m$^3$/h in the diesel engine which uses natural gas for fuel. The ship’s main boilers produce the superheated steam needed to run the turbo generators, which is connected to an alternator through a reduction gear box, and is used for producing the needed electrical energy. The ship has a total of 3 built-in turbo generators and 3 alternators for producing electrical energy of a total of 11.1 MW. For producing additional energy, the diesel generator with the power of 3.5 MW is used. Only the use of a diesel generator significantly increases the consumption of natural gas during the discharge rate greater than 669,768 m$^3$/h[4].

![Fig. 3. The amount of electricity generated and total gas consumption as a function of discharging flow of natural gas](image)

4. Potential application of solid oxide fuel cell on LNG carriers

Figure 4 displays a schematic representation of a hybrid system in which, instead of three turbo generators [26], three SOFCs with the same power would be installed in the steam plant for producing electrical energy, and a part of the heat produced during the fuel cell activity would be used for producing steam in the LPSG (low pressure steam generator), of up to 0.8 MPa, in order to achieve as great system efficiency as possible. A part of the anode exhaust gases would be re-used and mixed with the steam for reforming fuel (natural gas), before entering the reformer. The remaining part of the anode exhaust gases would travel further to the combustion chamber of the main boilers, where it would combust completely. However, as this reserve would be negligible, it was not taken into consideration in this study. The combustion of anode exhaust gases is vital for reducing exhaust emission, i.e. for environmental protection.

A fuel cell power system comprises a SOFC stack [27], its main component, a fan, a compressor, a reformer, a fuel (gas) heater, and an air heater. Taking into consideration the heat
requirements for the reactions in the reformer to occur, the gas heater raises the temperature of
the fuel before it reaches the reformer itself. Before entering the fuel cell stack, the gas is
reformed partly to hydrogen, and partly to carbon monoxide. The reactions in the reformer are
ones of the system optimization parameters, and they have enormous influence on the heat
balance and overall efficiency. The share of carbon that enters into a fuel cell (external reaction)
and the minimum flow of oxygen for cooling down the fuel cell stack (internal reaction) have
to be regulated and achieved by a certain degree of reformation [28]. The methane present in
the system cannot electrochemically oxidize, but only reform into hydrogen, carbon monoxide
or carbon dioxide. The reactions occurring in the steam reformer are steam reforming (1),
water–shift reaction (2) and reaction occurring in the fuel cell is electrochemical reaction (3)
[25]:

\[
\begin{align*}
CH_4 + H_2O & \leftrightarrow CO + 3H_2 & (\Delta H=-206 \text{ kJ/mol}) \\
CO + H_2O & \leftrightarrow CO_2 + H_2 & (\Delta H= 41 \text{ kJ/mol}) \\
H_2 + 1/2O_2 & \leftrightarrow CO + 3H_2 
\end{align*}
\]

Compressed air is heated before supplied to a fuel cell A fuel cell comprises two ribbed
bipolar plates, an anode, an electrolyte, and a cathode [29]. The bipolar plates are used to
separate as many individual cells as possible, and to electrically stack them in series, at the
same time bringing gas to the anode and air to the cathode. The ribbed bipolar plates have a flat
surface in between the catalyst layers, so that they adhere better to the surface. An electrical
potential evolves on the anode and cathode, i.e. a voltage by means of which we generate
electrical energy [27].

The temperature of air upon being fed into a fuel cell (cathode) is about 500 °C, while it
rises to about 800 °C when leaving the cell (cathode). Upon leaving the anode of the cell, the
temperature of the propulsion fuel is about 800 °C. Following the electrochemical reaction, a
part of the anode exhaust gases travel further to the combustion chamber of the steam generator,
where it combusts completely with the help of additional quantities of air and fuel.

According to [30], such hybrid systems with fuel cells can generate the output power of
up to 29.6 MW, by consuming 1.19 kg/s of natural gas, with the efficiency of up to 60%. Also,
5% losses due to electrical power conversion should be taken into account [30].

The SOFC generates direct current. For driving a ship’s electromotor, alternating current
of 440 voltages is needed. The conversion of direct current to alternating current was not taken
into consideration by this study due to the fact that, with the systems where the induced voltage
is higher than 110V, the conversion efficiency exceeds 95% [30].
Fig. 4. The combined system for thermal and electrical energy in combination with a steam plant
5. Energy consumption calculation

Measured data shown in table 1 were collected [26] and used for the energy consumption calculation.

<table>
<thead>
<tr>
<th>INPUT DATA</th>
<th>VALUE</th>
<th>UNITS OF MEASUREMENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power electricity of turbo generator</td>
<td>3.7</td>
<td>MW</td>
</tr>
<tr>
<td>Power electricity of diesel engine</td>
<td>3.5</td>
<td>MW</td>
</tr>
<tr>
<td>Total installed capacity of electricity (3 steam generators + 1 diesel generator)</td>
<td>14.6</td>
<td>MW</td>
</tr>
<tr>
<td>Power electricity at maximum discharge flow of cargo (natural gas)</td>
<td>14.2</td>
<td>MW</td>
</tr>
<tr>
<td>Maximum discharge flow of cargo (natural gas)</td>
<td>893,024</td>
<td>m³/h</td>
</tr>
<tr>
<td>Consumption of natural gas to drive 1 steam generator</td>
<td>730</td>
<td>kg/h</td>
</tr>
<tr>
<td>Density of natural gas (p&lt;sub&gt;atm&lt;/sub&gt; and 15°C)</td>
<td>428.633</td>
<td>kg/m³</td>
</tr>
<tr>
<td>The power produced during the operation of SOFC</td>
<td>29.6</td>
<td>MW</td>
</tr>
<tr>
<td>Consumption of natural gas for generated power of 26.9 MW</td>
<td>1.19</td>
<td>kg/s</td>
</tr>
</tbody>
</table>

5.1. Calculation of natural gas consumption in producing electrical energy by a steam generator

For producing electrical energy on LNG carriers, 3 turbo generators, and a diesel motor, which uses natural gas for fuel, were used [26]. These meet the capacity of generating the required quantity of electrical energy. The greatest amounts of electrical energy are required during a maximum compressed gas discharge flow (893,024 m³/h), at which the electric grid load amounts are about 14.2 MW. The estimation of the natural gas consumption by the diesel engine was not taken into consideration in this study.

The overall natural gas consumption by a turbo generators and a diesel engine at various discharge rates is described at the beginning of the paper, and shown in the graph in Figure 3. For producing 3.7 MW of electrical energy, an AT42CT-B model turbo generator uses about 15,200 kg/h of superheated steam [26], produced by an MB-4EKS-2 model of the main boilers [31]. The value of the gas amount required for producing 15,200 kg/h of superheated steam was measured during a cargo discharge, and totals about 730 kg/h of natural gas, meaning that for 3 turbo generators to operate, about 2,190 kg/h, i.e. 52,560 kg/d of natural gas is needed for generating electrical energy. It is important to note that natural gas densities vary depending on the natural gas composition, i.e. on the gas extraction location [26], but the arranged value of 530 kg/m³ is used for estimations [31]. If we take the value of the natural gas consumption for producing electrical energy (P = 3.7 MW) and divide it with the natural gas density, we get the gas amount required [26] for 3 turbo generators to operate:

\[ g_{3TG} = \frac{g}{\rho} \] (4)
where:

- $g$ – natural gas consumption for producing electrical energy by 3 turbo generators [kg/h],
- $\rho$ – natural gas density [kg/m$^3$] at $p_{\text{atm}}$ and 15°C

\[
g_{\text{STG}} = \frac{52,560}{530} = 99.17 \text{ m}^3/\text{d} \tag{5}
\]

Accordingly, it can be concluded that the overall natural gas consumption for generating electrical energy by means of 3 turbo generators amounts to 99.17 m$^3$/d.

In order to produce 15,200 kg/h of superheated steam to generate required electrical energy, main boilers consume $g_{P1} = 730$ kg/h of natural gas. Higher heating value of the natural gas is $H_g = 55,550$ kJ/kg, by multiplying these two values we get the amount of the input power [24]:

\[
P_1 = g_{P1} \times H_g \tag{6}
\]

\[
P_1 = 730 \times 55,550 = 40,551,500 \text{ kJ/h} = 11,264.31 \text{ kW} = 11,26 \text{ MW} \tag{7}
\]

If we divide the required power $P = 3.7 \text{ MW}$ with the input power $P_1 = 11.26 \text{ MW}$, we get the value of a steam generator energy conversion efficiency:

\[
\eta_{\text{EL1}} = \frac{P}{P_1} \tag{8}
\]

\[
\eta_{\text{EL1}} = \frac{3.7}{11.26} = 0.329 \tag{9}
\]

### 5.2. Calculation of natural gas consumption in producing electrical energy by SOFC

The electrical energy required for all the devices on a ship to operate during a compressed gas discharge by using a regasification device can be generated by means of a SOFC stack and a diesel motor using natural gas for fuel.

By increasing the reformation degree, the hydrogen percentage increases upon entering a fuel cell, compared to the steam percentage. The same thing happens to the fuel cell voltage. In contrast to this, increasing the level of fuel efficiency leads to a greater concentration of steam, compared to the concentration of hydrogen in the exhaust gases from the anode. Therefore, this effect is repeated when the gases enter the anode through the gas recirculation flow from the anode. The recirculation ratio can also be adjusted, just like the ratio between the anode exhaust gases and steam. A high recirculation ratio causes an increase in the amount of fuel entering the fuel cell. The fuel cell voltage slowly increases when an appropriate percentage of natural gas is reformed before entering the fuel cell, due to a high hydrogen percentage and a low steam percentage in the anode exhaust gases. Since the concentration of reactants in the fuel decreases with an increase in the fuel efficiency percentage, and since the fuel cell voltage cannot exceed the lowest value in the fuel cell, the fuel efficiency percentage limits the value of the fuel cell voltage. In fact, if the percentage of hydrogen upon entering the fuel cell increases due to recirculation from the anode, the percentage of steam would still be greater when entering the same fuel cell. Furthermore, the voltage drop resulting from more steam would be evident [27].

The input temperature deviation is not shown as an important deviation from the composition of exhaust gases. In reality, decreasing the fuel cell input temperature leads to decreasing the value of carbon dioxide and monoxide, since the chemical reaction has an
enhanced impact on the temperature decrease. During the system activity, the input temperature changes, while the output temperature remains constant and amounts to 800 °C [28].

Taking into account that power is calculated by multiplying electric current and voltage, power is constant with the input temperature, and increases with rises in the reformation degree. During simultaneous activity, i.e. by increasing current density and decreasing voltage, the fuel efficiency (efficiency of a process that converts chemical potential energy contained in a fuel-Methan into electrical energy and heat) for the same power is about 80%, and the anode recirculation is about 65%. The greatest electrical efficiency percentage is gained when the anode recirculation is somewhere between 50-60%, and the fuel efficiency percentage is 75% [27].

Considering the hybrid system with fuel cells from [30], it can produce the output power of up to \( P_0 = 29.6 \) MW, while the natural gas consumption amounts to 1.19 kg/s. If the same efficiency is assumed and by setting the simple ratio between consumption at maximum power and at part load, the following can be calculated for the natural gas consumption by the SOFC:

\[
g_{2m} = g_0 \times \frac{P}{P_0} \tag{10}
\]

where:

- \( g_0 \) – natural gas consumption for generating \( P_0 = 29.6 \) MW of electrical energy produced with the SOFC [kg/s],
- \( P \) – electrical energy demand [W],
- \( P_0 \) – electrical energy produced with the SOFC [W],

\[
g_{2m} = 1.19 \times \frac{3,700,000}{29,600,000} = 0.14875 \text{ kg/s} = 535.5 \text{ kg/h} = 12,852 \text{ kg/d} \tag{11}
\]

The same data for natural gas density can be applied, and the natural gas consumption can be divided for generating \( P = 3.7 \) MW of electricity with the natural gas density with the following calculation:

\[
g_p = g_{2m} / \rho \tag{12}
\]

where:

- \( g_{2m} \) – natural gas consumption for generating \( P = 3.7 \) MW of electrical energy produced with the SOFCs [kg/s],
- \( \rho \) – natural gas density [kg/m³] at \( p_{\text{atm}} \) and 15°C

\[
g_{2v} = 12,852 / 530 = 24.25 \text{ m}^3/\text{d} \tag{13}
\]

Consequently, it can be concluded that the overall natural gas consumption for generating \( P = 3.7 \) MW of electrical energy produced with the SOFC amounts to 24.25 m³/d.

For generating electrical energy produced with the SOFC, we consume \( g_{2m} = 535.5 \) kg/h of natural gas, and the calorific value of natural gas is \( H_g = 55,550 \text{ kJ/kg} \) [26]. Therefore, by multiplying these two values, we get the amount of the input power:

\[
P_2 = g_{2m} \times H_g = 535.5 \times 55,550 = 29,747,025 \text{ kJ/h} = 8,263.1 \text{ kW} = 8.26 \text{ MW} \tag{14}
\]

If we divide the required power \( P = 3.7 \) MW with the input power \( P_2 = 8.26 \) MW, we get a solid oxide fuel cell energy conversion efficiency:

\[
\eta_{EL2} = P / P_2 = 3.7 / 8.26 = 0.448 \tag{15}
\]
5.3. Using of thermal energy generated during a SOFC activity

As mentioned earlier, the heat generated during the activity of a SOFC would be partly used for preheating the air before entering the fuel cell, and partly for generating a lower pressure steam in a somewhat smaller heat exchanger (low pressure steam generator), by which a certain amount of energy generated by the steam generator for its own purposes would be saved. The lower pressure steam from this generator is used for heating fuel tanks, oil tanks and crew areas, for heating fuel, water, glycol and separating oil, and for heating air before entering the steam generator.

Based on [31], the graph in Figure 5 shows the steam amount that needs to be brought to a steam generator for generating lower pressure steam, and the overall quantity of steam produced in main boilers during various compressed air discharge flows through a regasification unit.

Fig. 5. The amount of steam required for steam production in a low pressure steam generator and the total amount of steam generated in the main boilers as a function of discharging flow of natural gas

5.4. Calculation of the thermal energy generated during a SOFC activity

The thermal energy generated during an SOFC activity with the power of \( P = 3.7 \text{ MW} \) and the working temperature of 800 °C, with 50% steam reformation of the natural gas and 50% recirculation of the anode exhaust gases totals [30]:

\[
P_{HE} = P \left( (1.25 / V_e) - 1 \right)
\]

where:

\( P \) – electrical energy from the SOFC [kW],
\( V_e \) – voltage of the single SOFC [V],

\[
P_{HE} = 3,700,000 \left( (1.25 / 0.98) - 1 \right) = 1,019,387.75 = 1.02 \text{ MW}
\]
From this, it can be concluded that, during the activity of an SOFC the power of \( P = 3.7 \) MW, the thermal energy of \( P_{\text{HE}} = 1.02 \) MW is generated. This energy will be used partly for preheating air before entering the fuel cell, and partly for generating lower pressure steam. On the other side, the total quantity of the thermal energy generated increases during the activity of 3 turbo generators, and amounts to \( P_{\text{HE3}} = 3.06 \) MW.

For the purpose of generating low pressure steam, it is necessary to generate 5,835 kg/h of superheated steam in the main boilers. For generating 5,835 kg/h of superheated steam, we use \( g_{P2} = 350 \) kg h\(^{-1}\) of natural gas, and the heat value of natural gas is \( H_g = 55,550 \) kJ/kg. Therefore, by multiplying these two values, we get the value of the input power \([26]\):

\[
P_{\text{HEAT1}} = g_{P2} \times H_g
\]

\[
P_{\text{HEAT1}} = 350 \times 55,550 = 19,442,500 \text{ kJ/h} = 5,400.69 \text{ kW} = 5.4 \text{ MW}
\]

The heat necessary for preheating air before entering an SOFC was not taken into consideration in this study due to its insignificance \([30]\). If we divide the thermal energy gained by the activity of 3 SOFCs, each of the power of \( P = 3.7 \) MW at the working temperature of 800 °C, with the thermal energy needed for the low pressure steam generator to operate, we get the overall degree of the thermal energy efficiency:

\[
\eta_{\text{HE}} = P_{\text{HE3}} / P_{\text{HE1}}
\]

\[
\eta_{\text{HE}} = 3.06 / 5.4 = 0.567
\]

Part of this heat energy is used to preheat the inlet air into the main boilers and the rest is used to heat the low pressure steam boiler. A low pressure steam boiler produces steam to be used for heating fuel and lubricating oil, fuel tanks, oil tanks, and heating air for accommodation on board.

6. Conclusion

The aim of this paper was to analyze energy efficiency when replacing the existing turbo generators by the Solid Oxide Fuel Cells (SOFCs). Instead of three turbo generators (each of the power of 3,700 kW), three SOFCs (each of the power of 3,700 kW) would be installed in the steam plant for producing electrical energy to improve efficiency.

When using the SOFC system, the energy conversion efficiency amounts to 44.8%, while it totals 32.9% when using a steam generator.

Furthermore, by using the SOFC system, the generated thermal energy can be used, as described in this article, and the savings would amount to about 2.6% of the overall natural gas consumption, depending on the natural gas discharge flow.

It is important to point out that the estimation of electrical energy generation was performed for one steam generator and for the SOFC system with the power of 3.7 MW, while the estimation of thermal energy generation was based on the usage of three steam generators.

This research provides an innovative solution to convert natural gas from liquid to gas for a new generation of LNG carriers with the regasification unit. These days the LNG carriers, that are being built, have great needs for electrical energy.

Further research should include different main propulsion technologies and an open, combined or closed cycle together with steam heaters during the regasification operation.

References:


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NUMERICAL SIMULATION OF THE OBLIQUE WATER ENTRY OF WEDGES WITH VORTEX SHEDDING

UDC 629.5.016.8
Original scientific paper

Summary

The hydrodynamic problem of the water entry of wedges with oblique velocity has been investigated numerically. The simulations of the interaction of the wedge and initial calm surface are carried out by solving the Reynolds-averaged Navier-Stokes equations (RANS). The overset mesh technique is introduced and the air-water interface is tracked using the method of Volume of Fluid (VOF). The water entries of wedges with various deadrise angles at different oblique velocities are simulated. A noticeable vortex flow and low pressure area on the leeward side of the wedge have been observed. The evolution of vortex and the pressure distribution over the wedge surface are analysed. The local vortex flow affects the local hydrodynamic pressure significantly.

Key words: Numerical simulation; oblique water entry; vortex shedding; overset mesh technique;

1. Introduction

The hydrodynamic impact, including the slamming of ships with V-shaped bow or planning vessels, the ditching of seaplane, would cause the damage of structures. The impact usually last a short period and the problem can be treated as the water entry of wedges. The Mach number of the fluid flow during impact of these marine structures is usually negligible, and the compressibility of the fluid can be ignored (Newman [1]). The analytical solution was firstly proposed by Von Karman [2] based on momentum conservation. Wagner [3] took account the linear free surface effect; the modification was effective in predicting the overall impact loads at initial stage. The matched asymptotic method was further developed to by Armand and Cointe [4], Korobkin and Puknachov [5], Korobkin [6], Howison, Ockendon and Oliver [7], and the water jet was included. Mei, Liu and Yue [8] developed an analytic solution based on the generalized Wagner theory through boundary-integral equation method; the conformal mapping technique was adopted.

Efforts have been made to simulate the water entry using nonlinear free surface theory. For wedge water entry problem at constant velocity, the time and spatial variables can be combined and the flow becomes self-similar when the gravity effect and the viscosity have
been ignored. The nonlinear similarity solution of vertical water entry of symmetric wedge had been obtained by Dobrovolskaya [9] through conformal mapping. Zhao and Faltinsen [10], Battistin and Iafrati [11] studied the vertical water entry of wedges through time stepping schemes, where the nonlinear free surface boundary conditions had been satisfied. Wu, Sun and He [12] and Xu, Duan and Wu [13] further investigated the free fall motion of wedges; the auxiliary function had been introduced to decouple the interactive motions of the fluid and structures. Except inviscid theory, numerical simulations of water entry problems were carried out through solving Navier-Stokes (N-S) Equations. Fairlie-Clarke and Tveitnes [14] simulated the water entry of wedges using the finite volume method based on commercial computational fluid dynamics (CFD) code Fluent. Shademani and Ghadimi [15] introduced a two dimensional two-phase Finite-Element method (FEM) to study the hydrodynamic impact force of a wedge, and the cavity and the secondary impact of the returning free surface had been investigated.

It happens that the wedge-shaped structure impacts water surface with inclined angle and/or oblique velocity. Xu, Troesch and Vorus [16] studied the impact of asymmetric planning vessel. Garabedian [17] and Chekin [18] had developed similarity solution for the asymmetric water entry problem. Semenov and Iafrati [19] analysed the similarity solution of asymmetric wedge water entry through conformal mapping of complex flow. Xu, Duan and Wu [20] found the similarity solution through iterations based on boundary element method. Negative pressure at the apex on the leeward side of the wedge, which may cause the cavitation-ventilation, had been observed. The typical experimental and theoretical study on the oblique water entry of asymmetric wedge had been carried out by Judge, Troesch and Perlin [21]. Two types of impact, with and without separation-ventilation at the wedge apex, had been investigated. They attempted to address the criteria for the ventilation at the wedge apex when the inclined angle or ratio of horizontal and vertical velocity increases. Agreement of the experiment and the predicted separation-ventilation at the wedge apex had been found. Gu et al [22] had analysed the oblique water entry of wedges based on two phase flow theory and the level set method. The ventilation during water entry had been analysed. Javanmardi, Ghadimi and Tavakoli [23] studied the water entry of a propeller blade impacting on the water surface, the cavitation and ventilation effects on the hydrodynamic characteristic were investigated through Finite volume method. Except the separation-ventilation during the wedge water entry, vortex shedding would develop on the leeward side of the wedge apex in the case of attached flow impact. Low pressure does not necessarily result the ventilation-cavitation since more physical conditions are required. The flow pattern and the pressure distribution require further investigations.

For attached flow, vortex shedding occurs at the sharp edge due to large velocity gradient when there exists transverse flow (Bactheor [24]). Pullin [25], Graham [26] and Pullin and Perry [27] have investigated the vortex shedding of wedges theoretically and/or experimentally, although there is reasonable discrepancy. For the oblique water entry of a wedge, the vortex shedding is developed due to large gradient of pressure on the leeward side. The vortex flow will affect the flow pattern and pressure distribution significantly in return. Riccardi and Iafrati [28] introduced point vortex to circumvent the singularity at wedge apex. However, the free surface was assumed quiescent. Xu and Wu[29] studied the water entry of a wedge at oblique velocity using a time stepping scheme, the vortex shedding is approximated by using point vortex. The Kutta condition was satisfied when solving the boundary integral equation, and the pressure jump at the wedge apex had been avoided. Semenov and Wu [30] considered the water entry of wedges with vortex wake in the framework of potential flow. The vortex flow was obtained through the integral hodograph method combined with Birkhoff-Rott method. Besides the inviscid theory, the hydrodynamic problem can be
modeling through CFD by solving the N-S equations. The effects of viscosity and the dissipation of the vortex flow can be included.

Present study concerns the oblique water entry of wedges with various deadrise angles. In physics, the breath of a wedge is usually finite. If the mass of whole structure is large, like a seaplane or a lifeboat, the variation of velocity due to local impact would be less significant at the initial stage of the impact. It becomes practical to study the water entry of a finite wedge with constant velocity, which makes the problem simpler. The interactions of the wedge and flow are simulated through solving the Reynolds Averaged Navier-Stokes equations (RANS). The realizable \( k-\epsilon \) model [31] with high \( y^+ \) wall treatment is adopted. The overset mesh technique is introduced. Volume of fluid (VOF) method [32] is used to capture the interface of water and air. It is observed that the vortices have been developed as the wedge entering initial quiescent water surface. As the vortex rolls up, the low pressure area close to the wedge apex expands. The pressure distributions with various deadrise angles and oblique velocity are analysed.

2. Numerical methods

The hydrodynamic problem of the oblique water entry of a wedge is considered. The Cartesian coordinate system \( o-x-y-z \) is defined with \( x \) axis along the undisturbed water surface and \( y \) axis pointing upwards. As shown in Fig.1(a), the breath of the wedge is denoted as \( B \), and the deadrise angles is \( \beta \). The wedge impacts on the initial calm water surface with prescribed vertical velocity \( V \) and horizontal velocity \( U \), and we have \( \epsilon = U/V \). A moving coordinate system \( o'-x'-y'-z' \) which is fixed with the wedge is defined; its origin is located at the apex of the wedge and its horizontal axis and vertical axis are parallel with the axes of the \( o-x-y-z \) coordinate system at beginning of the impact. The wedge enters water with constant velocity during the whole process, there is no angular velocity. It moves only in the horizontal direction and vertical direction.

The concerned velocity of the wedge is much lower than the speed of sound, the compressibility of the fluid is therefore ignored. The fluid is treated as incompressible and the density is constant. The conversation equations of mass, momentum and energy are satisfied. The partial differential equation for the mass conservation is expressed as

\[
\frac{\partial u_i}{\partial x_i} = 0
\]  

(1)

where \( i=1,2,3 \), \( x_1 = x \), \( x_2 = y \), \( x_3 = z \), and \( u_i \) is the velocity components.

The conservation of the momentum equation is written as

\[
\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\mu}{\rho} \frac{\partial^2 u_i}{\partial x_j \partial x_j} + f_b
\]

(2)

where \( \mu \) is the coefficient of viscosity, \( \frac{\partial p}{\partial x_i} \) is the surface force, \( f_b \) is the body force acting on the fluid volume and \( f_b = g \).

The VOF is used to track the interface of the two phase flow, we have

\[
\frac{\partial}{\partial t} (\alpha) + \nabla \cdot (\alpha \mathbf{v}) = 0
\]

(3)
where \( \alpha \) is the volume fraction of the water phase in each cell. If \( \alpha = 0 \), the cell will be filled with air; the cell is filled with water when \( \alpha = 1 \); if \( 0 < \alpha < 1 \) the cell will be filled with both water and air as the interface of the two phases.

### 3. Convergence study and validation

The numerical simulations of the water entry of wedges are conducted using commercial CFD code STAR-CCM+(version 11.06). The interactions of the fluid and wedges are simulated by solving the RANS equations through Finite Volume Method (FVM). The VOF is adopted to capture the free surface flow. The turbulent flow is modelled by the realizable \( k-\varepsilon \) model. The overset mesh method is introduced. There are two sets of meshes, one is the overset mesh block moving with the wedge, and the other is the background mesh which contains the regions of air and water. The interpolation boundary exchanges data between the region of overset mesh block and the region of background, as shown in Fig. 1(b). The near wall prism layer is employed on the wedge surface in the overset region, as shown in Fig. 1(c). A local mesh refinement is conducted, and large meshes far away from the wedge are used. The wedge water entry is treated as a two-dimensional problem; the background fluid domain contains only two mesh layers in the direction of \( z \) axis. The requirements of memory and CPU have been reduced significantly.

![Diagram](attachment:image.png)

**Fig.1** (a) The sketch of the wedge water-entry problem, (b) the overset mesh and (c) refined mesh near the wedge surface

The convergence study on mesh size and time resolution is carried out firstly. The vertical water entry of a symmetric wedge with deadrise angle \( \beta = 45^\circ \) is simulated. The breath of the wedge \( B = 0.05m \). The vertical velocity \( V = 5m/s \). Two sets of meshes and two different time step intervals for each set of meshes have been used. The parameters of fundamental mesh size \( \Delta x \) near the wall of wedge and time resolution \( \Delta t \) are listed in table 1. The thickness of near wall prism layer is \( 1.0 \times 10^{-5} m \) and there are 10 mesh layers parallel the wall of the wedge. As shown in Fig.2, the pressure distributions on the wedge surface are
in good agreement except case 1. The pressure distribution and the vertical force are converged when mesh size \( \Delta x < 1.0 \times 10^{-4} \) m and time resolution \( \Delta t < 1.0 \times 10^{-5} \) s. Comparing the pressure distribution with the similarity solution of Xu, Duan and Wu [20], good agreement has been found, as shown in Fig.2(b). In table 1, the vertical impact forces at the depth \( s = Vt = 0.6B \) are in good consistence. We note that the results of Xu, Duan and Wu [20] ignored the effects of viscosity and the gravity. Present numerical results indicate that the effects of gravity and viscosity are less significant during the initial stage of the impact.

**Table 1** The mesh size and time resolution of the simulations, and the corresponding vertical force at \( Vt / B = 0.6 \)

<table>
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<td>( \Delta t ) (s)</td>
<td>1618324</td>
<td>1618324</td>
<td>2551573</td>
<td>2551573</td>
</tr>
<tr>
<td>( \Delta x ) (m)</td>
<td>2.0 \times 10^{-5}</td>
<td>1.0 \times 10^{-5}</td>
<td>1.0 \times 10^{-5}</td>
<td>5.0 \times 10^{-5}</td>
</tr>
<tr>
<td>( F_y / 0.5 \rho V^2 (Vt) )</td>
<td>6.943</td>
<td>6.975</td>
<td>6.962</td>
<td>6.971</td>
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</table>

![Fig.2](image1.png)

**Fig.2** Pressure distribution of vertical water entry of a wedge with \( \beta = 45^\circ \), \( Vt / B = 0.6 \). (a) results of different meshes and time resolutions, (b) comparison with similarity solution.

![Fig.3](image2.png)

**Fig.3** (a) The hydrodynamic force on the wedge, (b) velocity.

The free fall water entry of a wedge with \( \beta = 30^\circ \) is simulated. The wedge is 1.0 meter in length and 0.5m in breath, the mass \( m = 255.5 \) kg. The vertical velocity \( V = 6.15 m/s \). The results are validated through comparing with the experimental data of Zhao and Faltinsen [33]. The length of the measured section is 0.2m. As shown in Fig.3, the vertical force on the
measured section increases as the wedge entering water; it decreases after the wedge is below the initial calm waterline. The vertical force and velocity agree well with the experimental results.

Table 2 The typical characteristics of mesh size and time resolution in the simulations with $\beta = 45^\circ$ and $\varepsilon = 0.3$.

<table>
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<td>Δx(m)</td>
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<td>1.0×10^{-4}</td>
<td>5.0×10^{-4}</td>
<td>5.0×10^{-4}</td>
<td>2.5×10^{-4}</td>
</tr>
<tr>
<td>Δt(s)</td>
<td>2.0×10^{-5}</td>
<td>1.0×10^{-5}</td>
<td>1.0×10^{-5}</td>
<td>5.0×10^{-6}</td>
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</tr>
</tbody>
</table>

We further study the independence of the mesh size and the time resolution in the case of oblique water entry. The vertical velocity $V = 5m/s$ and $\varepsilon = U/V = 0.3$ are prescribed. The typical mesh sizes and time resolutions are shown in table 2. Simulations with three sets of meshes and three different time resolutions are conducted. The horizontal and vertical forces against time are shown in Figs. 4(a) and 4(b) respectively. The forces increase linearly at the beginning of the impact and they drop when the flow separates at the knuckles. In Fig. 5(a) the free surface profile at $Vt/B = 0.6$ is presented. As shown in Fig. 5(b), the pressure distributions found their agreement when different mesh sizes and time resolutions are adopted. The numerical results are converged. The velocity field of case 3 at $Vt/B = 0.72$ are further shown in Fig.6. A stagnation point on the windward side (right wedge surface) can be observed. The vortex develops on the leeward side (left wedge surface). Another stagnation above the vortex core on the leeward side is found. A higher pressure at these stagnations can be found. The pressure close to the vortex core drops significantly. This low pressure area will further expand as the evolution of the vortex. This will be further studied in the following section. The typical mesh size and time resolution of case 3 in table 2 will be adopted.

![Fig.4](image1.png) The time history of (a) horizontal force and $F_1$, and (b) vertical force $F_3$.

![Fig.5](image2.png) water entry of wedge with deadrise angle $\beta = 45^\circ$, (a) The free surface profile and (b) pressure distribution at $Vt/B = 0.6$. 

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Fig. 6 (a) The velocity field and (b) enlargement of the local flow field near the wedge apex when $Vt/B = 0.6$, the deadrise angle wedge $\beta = 45^\circ$.

The oblique water entry of wedge is further validated. In the work of Russo et al [34], the wedge’s deadrise angle $\beta = 37^\circ$, $v = \sqrt{U^2 + V^2} = 2.75m/s$ and $\varepsilon = 0.364$. The mass of the wedge is 0.89 kg, the length is 0.19 m and the width is 0.2 m. The wedge slides along an oblique bar and impact on the water surface. A pressure sensor (PS) is placed on the right hand side of wedge surface at point $n$, which has a distance 0.078 m from the wedge apex.

We define the oblique displacement $r = \sqrt{q^2 + s^2}$, where $q$ is transverse displacement, and $s$ is the vertical displacement. The numerical results are compared with the experimental results of Russo et al [34]. As shown in Fig. 7, the oblique displacement, velocity and acceleration are in good agreement. Fig. 7d shows the time history of the pressure at point $n$. The calculated pressure and the pressure obtained from pressure sensor and the pressure reconstructed from the PIV velocity field agree well with each other. The difference between numerical results and experiments data are mainly due to the friction of experimental apparatus and the resulting discrepancy of velocity.

Fig. 7 Results of oblique water entry of a wedge with $\beta = 37^\circ$ and $\varepsilon = 0.364$, (a) displacement, (b) oblique velocity, (c) oblique acceleration, (d) time history of pressure at point $n$.
4. Numerical results and discussions

Fig. 8 Free surface elevation during the water entry of a wedge with $\beta = 45^\circ$, $\varepsilon = 0.4$ at (a) $Vt/B = 0.24$, (b) $Vt/B = 0.48$, (c) $Vt/B = 0.72$, (d) $Vt/B = 0.96$ (e) $Vt/B = 1.20$ (f) $Vt/B = 1.68$.

The oblique water entry of a wedge with small to medium $\varepsilon = U/V$ would not necessarily result flow detachment-ventilation at the wedge apex. In the case of attached flow water entry, the vortex will shed from the wedge apex and roll up due to the transverse flow. This phenomenon has been omitted in the experiment. The water entry of a finite wedge with deadrise angle $\beta = 45^\circ$ and breath $B = 0.05m$ has been simulated. The entry speed $V = 5m/s$ and $\varepsilon = 0.4$. The free surface profile at various entry distances are shown in Fig.8. The water piles up and the jet flow moves along the wedge surface in initial stage. The flow detachment at the wedge knuckles occurs and a large cavity forms as the wedge entering water further. Fig.9 presents the pressure contour and the local flow velocity at $Vt/B = 0.24$, $Vt/B = 0.48$, $Vt/B = 0.72$, $Vt/B = 0.96$, $Vt/B = 1.20$ and $Vt/B = 1.68$. As shown in Fig. 8(a), the highest pressure at $Vt/B = 0.24$ appears at the jet root on the windward side and the low pressure zone is found close to the apex on the leeward side of the wedge. After the jet flow separates at the knuckles, the high pressure area moves from the jet root to the zone approaching the wedge apex. The transverse flow at the wedge apex induces the vortex flow. The vortex evolves and expands. The low pressure area on the leeward side expands as the vortex develops, see Fig.9. It develops into an elliptic-shaped vortex bolt which elongated along the leeward side of the wedge surface. Similar to Fig.6, the stagnations of the velocity on the two sides of the apex can be observed. Fig.10 shows the pressure distribution on the wedge surface at various entry distances. The pressure is higher at the initial stage of the water entry. Comparing with similarity solution of Xu, Duan and Wu[20], most part of pressure distribution at $Vt/B = 0.24$ agree each other, but discrepancy near the wedge apex can be observed, as shown in Fig.10(b). The low pressure area is larger than that of the inviscid theory. After the jet flow detaches the knuckles, the pressure at the knuckle decreases significantly and equals the ambient air pressure. The peak pressure moves to the wedge apex where the flow stagnation can be found. The lowest pressure due to vortex flow is found on the leeward side of the wedge. The low pressure area expands as the development of vortex. The highest pressure on the leeward side, which is corresponding to the stagnation point, is moving away
Numerical simulation of the oblique water entry of wedges

from the apex as the wedge entering water. We further compare the pressure distribution obtained from the inviscid theory by Bao, Wu and Xu [35], the discrepancy caused by the vortex flow on the leeward side are significant at $\frac{Vt}{B} = 0.72$ and $\frac{Vt}{B} = 1.20$, as shown in Figs.10(c) and 10(d). The pressure distribution in Fig.10 indicates that vortex flow affects the local pressure significantly during oblique water entry.
Fig.9 Wedge water entry with $\beta = 45^\circ$, $\varepsilon = 0.4$, the pressure contour and velocity vector at (a)(b) $Vt / B = 0.24$, (c)(d) $Vt / B = 0.48$, (e)(f) $Vt / B = 0.72$, (g)(h) $Vt / B = 0.96$, (i)(j) $Vt / B = 1.20$, (k)(l) $Vt / B = 1.68$.

Fig.10 (a) Pressure distribution at various water entry distances with $\beta = 45^\circ$ and $\varepsilon = 0.4$, comparison with (b) the similarity solution at $Vt / B = 0.24$ and (c)(d) the numerical pressure with $Vt / B = 0.48$ and $Vt / B = 1.2$. 

![Numerical simulation of the oblique water entry of wedges with vortex shedding](image-url)
Numerical simulation of the oblique water entry of wedges with vortex shedding

X.B. Yang, G.D. Xu

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Fig. 11 Pressure distribution at various water entry depth with deadrise angle $\beta = 45^\circ$, (a) $\varepsilon = 0.3$ and (b) $\varepsilon = 0.5$

Further simulations of the water entry of the wedge with deadrise angle $\beta = 45^\circ$ at oblique velocity $\varepsilon = 0.3$ and $\varepsilon = 0.5$ are carried out. The pressure distributions on the wedge surface are shown in Fig. 11. As the wedge entering water, the low pressure area due to vortex shedding can be found on the leeward side of the wedge. Similar with those in Fig. 10, the low pressure zone expands as the wedge entering water further. We note that the lowest pressure coefficient $c_p$ is positive when $\varepsilon = 0.3$, while the lowest pressure $c_p = -2.5$ when $\varepsilon = 0.5$. This suggests that the cavitation and ventilation would take place if the pressure is lower than the cavitation pressure. The interactions of cavity and vortex flow would make the problem extremely complicated which is beyond the scope of present study.

The oblique water entries of wedges with $\beta = 30^\circ$, $\beta = 40^\circ$, $\beta = 50^\circ$, $\beta = 60^\circ$ are simulated. The entry speed $V = 5m/s$ and $\varepsilon = 0.4$. Fig. 12 shows the pressure distributions at various entry distances. The low pressure area is observed and expands on the leeward side as the wedge entering water. The highest pressure on the windward side decreases as $\varepsilon V_t / B$ increases; it becomes stable as the entry distance increases further. It is interesting to see that the highest pressure coefficient on the windward side is close to 2.0, although the deadrise angles are different. We note that the negative pressure coefficient of wedges with $\beta = 50^\circ$ and $\beta = 60^\circ$ is considerably low; while the pressure coefficient on the leeward side of wedge with $\beta = 30^\circ$ is positive. The typical pressure contour and the local velocity vectors at $\varepsilon V_t / B = 0.6$ are shown in Fig. 13. The pressure on the windward side is much higher than that on the leeward side. The vortex shed from the wedge apex and developed into vortex bolt. We note that the induced vortex flow velocity with $\beta = 60^\circ$ is much higher than that of wedge with $\beta = 30^\circ$. The vortex flow is circular when $\beta = 60^\circ$ while it is oblate when $\beta = 30^\circ$, as shown in Fig. 13. The sharper wedge apex induces stronger transverse flow and lower pressure can be found on the leeward side. The negative pressure and the development of vortex flow depend on the shape of the apex.
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Numerical simulation of the oblique water entry of wedges with vortex shedding

Fig. 12 Pressure distribution of the wedges at various entry distance. (a) $\beta = 30^\circ$, $\varepsilon = 0.4$, (b) $\beta = 40^\circ$, $\varepsilon = 0.4$, (c) $\beta = 50^\circ$, $\varepsilon = 0.4$, (d) $\beta = 60^\circ$, $\varepsilon = 0.4$

Fig. 13 The pressure contour and velocity vectors of wedge water entry with $\varepsilon = 0.4$, $Vt/B = 0.6$. (a)(b) $\beta = 60^\circ$, (c)(d) $\beta = 30^\circ$. 

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Numerical simulation of the oblique water entry of wedges with vortex shedding

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Fig. 14 The typical pressure distribution with various oblique velocity, (a) $\beta = 30^\circ$, $Vt / B = 0.30$ (b) $\beta = 40^\circ$, $Vt / B = 0.6$, (c) $\beta = 50^\circ$, $Vt / B = 0.6$ and (d) $\beta = 60^\circ$, $Vt / B = 0.6$.

The effects of oblique velocity are further investigated. The pressure distributions on the wedges surface with various $\varepsilon$ are shown in Fig.14. The positive pressure on the windward side increases while the pressure on the leeward side decreases significantly as $\varepsilon$ increases. Comparing the pressure of the wedges with $\beta = 30^\circ$, $\beta = 40^\circ$, $\beta = 50^\circ$ and $\beta = 60^\circ$, the sharper wedge apex results lower pressure on the leeward side of the wedge. The lowest pressure coefficient $c_p$ will decrease significantly as $\beta$ increases. The results suggest that the cavitation and ventilation will occur with larger $\varepsilon$.

5. Conclusion

The oblique water entries of wedges have been investigated numerically through solving the RANS equations. The overset mesh technique has been adopted in the simulations. The vortex shedding has been observed near the leeward side of wedge surface. When a sharp wedge with oblique velocity is considered, high pressure on the windward side and the low pressure area near the vortex core on the leeward side can be observed. There exists two stagnation points on the windward side and leeward side of the wedge. The highest pressure coefficient $c_p$ can be found at these two stagnations, one is close to the wedge apex on the windward side and the other is right above the vortex core on the leeward side. As the wedge entering water, the highest $c_p$ decreases on the windward side, and the low pressure area on the leeward side of the wedge expands. The numerical results show that the vortex induced flow and the lowest pressure is significantly affected by the oblique velocity and the deadrise angle. The wedge with sharp apex will induce stronger vortex flow and lower pressure, where the ventilation and separation would become an issue.
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Reference


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Numerical simulation of the oblique water entry of wedges with vortex shedding

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